IN SITU TESTING PROCEDURE

FOR OBTAINING DYNAMIC AND

CYCLIC SOIL PROPERTIES

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Studies are presented which indicate the feasibility of an in situ testing procedure for determining in situ cyclic and dynamic soil properties. The properties are to be described in the detail and with the accuracy appropriate for later stages of the analysis and design of important, earthquake-resistant, soil-structure-equipment systems. The procedure is intended to allow more effective use of the costly and potentially powerful analysis procedures used to predict system behavior during earthquakes. One study in which tests were simulated analytically indicates theoretical feasibility. A second study indicates operational and economic feasibility. Items were identified which could create added uncertainty in inferring soil properties, but the added uncertainty was judged to be only modest.

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TECHNICAL SUMMARY

Introduction

This document reports on feasibility studies, conducted for the National Science Foundation, which indicate the feasibility of a proposed in situ geotechnical testing procedure for obtaining in situ cyclic and dynamic engineering soil properties. The studies represent Phase I of a planned three phase project. The project falls under the domain of the Earthquake Hazard Mitigation program.

Purpose of Project

The purpose of the project is to develop an in situ geotechnical testing procedure for obtaining accurate and detailed descriptions of the in situ liquefaction and cyclic degradation characteristics of a soil, the low amplitude dynamic shear modulus, and the variation in the dynamic shear modulus with shear strain. This information is required for analyses which predict the behavior of soil-structure-equipment systems during earthquakes. The level of accuracy and detail to be provided is intended to be appropriate for the intermediate and final stages of the analysis and design of important structures located in seismically active areas.

Existing Methods

While a number of important advances have been made in the area of determining in situ cyclic and dynamic soil properties, further worthwhile advancement is possible. We feel that there are no existing procedures for determining in situ cyclic degradation and liquefaction characteristics in the detail and to the accuracy that is needed for the effective use of the costly and potentially powerful analysis procedures appropriate for the intermediate and final stages ot analysis and design. Additionally, there are significant limitations cr drawbacks in existing procedures for determining dynamic shear moduli.

Currently, in situ cyclic and dynamic soil properties are obtained by laboratory and in situ testing. Laboratory testing of samples is plagued by the serious problem of disturbance to in situ conditions. Generally, with in situ testing, in situ conditions are preserved to a greater degree; however, it is difficult in situ to apply earthquake-type cyclic shear loads to a well-defined element of soil, to induce important phenomena observed during earthquakes, and to obtain information in appropriate detail. The problems with laboratory and in situ testing procedures may cause considerable uncertainty and potential for error in defining in situ soil properties.

Uncertainty and error in defining in situ soil properties will create uncertainty and error in the results of analyses relying on these properties. In fact, the limitations in our ability to accurately and descriptively define in situ soil properties can severely limit the effectiveness of the costly and potentially powerful analysis procedures which have been developed during recent years. This can lead to either costly, excessively conservative designs or unconservative designs.

Proposed Method

The proposed testing procedure is intended to provide more accurate and detailed descriptions of in situ cyclic and dynamic soil properties than can currently be provided. It is to do so by effectively combining attractive features of in situ testing and laboratory testing while minimizing shortcomings.

The method will be direct. Cyclic earthquake-type shear loads will be applied to a well-defined element of soil in a simple but effective manner. The behavior of the test soil is expected to correspond closely to behavior expected during earthquakes. Thus, uncertainties associated with the loading of the soil and its mode of failure will be reduced. Tests will be conducted in situ and a number of steps will be taken to minimize disturbances to in situ conditions. Thus, the very important effects of in situ factors are expected to be captured in measurements. Several features are to be provided to help induce the phenomena of interest and also to simplify the interpretation of test results. The detailed information needed by earthquake analyses will be directly provided, minimizing intermediate interpretation. Also, the procedure is expected to apply to most soils of interest. Finally, the method will require only a single borehole so that it may be used in confined and harsh environments.

The probe of the testing system will consist of two concentric thin-walled cylinders. In practice, the cylinders will be carefully penetrated below the base of a borehole. The test soil will be the well-defined annular zone of soil between the two cylinders. A cyclic or impulsive torsional loading will be applied to the inner cylinder. In response, the inner cylinder is expected to rotate in a manner dependent on the shear properties of the test soil. Both torques and rotations will be measured by transducers in an instrumented head. Soil properties will be inferred by modeling tests analytically using descriptive procedures.

Potential Benefits

The proposed in situ testing procedure is intended to provide descriptions of in situ cyclic and dynamic soil properties to the level of accuracy and detail appropriate for the intermediate and final stages of the analysis and design of important structures located in seismically active areas. This would allow more effective use of the costly and potentially powerful analysis procedures used for this application. Defining in situ cyclic and dynamic soil properties more accurately and in greater detail will reduce error and uncertainty in predicted site-specific earthquake excitations and predicted behaviors of soil-structure-equipment systems (dams, buildings, offshore structures, embankments, etc.) during earthquakes. In turn, this would advance our ability to economically insure the integrity and reliability of soil-structure-equipment systems during earthquakes.

Potential Commercial Applications and Users

There are several potential commercial applications of the proposed testing procedure. The main application is intended to be the design,

construction, and maintenance of soil-structure-equipment systems to insure resistance to earthquakes. Additionally, the procedure is expected to apply to the design, construction, and maintenance of 1) soil-structure-equipment systems to insure resistance to water wave and blast loadings, 2) machine foundations, and 3) delicate soil-structure-equipment systems to insure isolation from vibrations. Potential users of the testing system or the information provided by the testing system include agencies of governments, universities, geotechnical engineering firms, oil companies, and power companies.

Phase I Objectives, Procedures, and Accomplishments

Phase I work included several studies: a theoretical feasibility study, a supplementary theoretical feasibility study, and an operational feasibility study.

We conducted a theoretical feasibility study to determine the theoretical feasibility of the proposed testing system. Tests were simulated analytically considering expected ranges of selected soil properties. Studies were carried out to determine the theoretical feasibility of the testing system for determining 1) the low amplitude dynamic shear modulus and the variation in the dynamic shear modulus with shear strain, 2) the degradation characteristics of clays, and 3) the degradation and liquefaction characteristics of sands and silts.

Results from our study indicate that the proposed testing system is theoretically feasible. The behavior of the testing system was predicted to be sensitive to each soil property considered in a clear, physically reasonable manner.

During our study, we identified one area requiring further study. The configuration of the inner cylinder, specified for the preliminary design of a laboratory research prototype testing system, was thought to be torsionally more flexible, relative to the test soil, than we had originally considered. As a result, we carried out a cursory supplementary study in addition to our proposed theoretical feasibility study. The purpose of the supplementary study was to estimate roughly the rotational flexibility of the inner cylinder, as configured in the preliminary design of the testing system, relative to the rotational flexibility of the test soil. The estimate was made to help determine whether special attention would be required as a result of such relative flexibility. The results of the study indicate that the inner cylinder may be torsionally flexible relative to the test soil under certain conditions. This may increase somewhat uncertainty in inferring soil properties from test results. However, we concluded that because of the many effective steps which could be taken, if necessary, to mitigate this concern, the proposed testing system should serve its stated purpose well and advance our ability to determine in situ cyclic and dynamic soil properties.

An operational feasibility study was conducted to determine the operational feasibility of a laboratory research prototype testing system. This system will be functionally similar to the field system scheduled for development during Phase III. The mechanical engineering design firm of Sweet & Aiken, Inc., (Sweet & Aiken) conducted the study. Studies were carried out to determine the operational feasibility of a laboratory testing system for determining 1) the low amplitude dynamic shear modulus and the

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variation in the dynamic shear modulus with shear strain, 2) the cyclic
degradation characteristics of clays, and 3) the cyclic degradation and
liquefaction characteristics of sands and silts.

Results from the operational feasibility study indicate that the
laboratory research prototype testing system is operationally feasible. The
main components of the system, which satisfy reasonably well design criteria
specified by our firm, were found to be either available or readily
producible and it was determined that these components could be assembled
into a convenient, workable arrangement. The operational feasibility study
resulted in a preliminary design for a laboratory research prototype testing
system.

Not all specified criteria were satisfied; however, the resulting
consequences are not expected to be severe. For example, Sweet & Aiken was
unable to find a transducer capable of accurately measuring the smallest
specified angles of rotation of the inner cylinder. However, it appears that
reasonably small angles will be able to be measured. Thus, the potential
increase in uncertainty in inferring soil properties from test results was
judged to be only modest. Also, advances in appropriate technologies are
expected to advance our ability to infer soil properties using the proposed
testing system.

Sweet & Aiken estimated a cost of $78,260 for the detailed design and
construction of a complete laboratory research prototype testing system.
This is a conservative estimate and such a cost is consistent with that of
the design and construction of comparable testing systems. The benefits of a
working system are expected to greatly exceed the overall cost. The cost
saving alone, resulting from basing the design of a single major structure
on appropriately accurate and detailed information on in situ dynamic and
cyclic soil properties, is expected to greatly exceed the cost of the
development of the testing system.

Estimate of Feasibility

Based on the results of the feasibility studies discussed in this
report, we concluded that the proposed in situ testing system is feasible.
Ultimately, the testing system is expected to effectively provide information
on in situ cyclic and dynamic soil properties to the level of accuracy and
detail appropriate for the intermediate and final stages of the analysis and
design of important structures located in seismically active areas. The
Phase I studies indicate that a laboratory research prototype testing system
is theoretically, operationally, and economically feasible. Since the
additional equipment required for a field system is reasonably conventional,
and since the additional problems that could be encountered in the field
appear to be either small or surmountable, we believe that an in situ field
testing system is feasible.

Much research and development will be required before the full potential
of the testing system can be realized. This is consistent with comparable
testing systems. Much effort will be needed because the problem of
determining in situ cyclic and dynamic soil properties is not a simple
problem, the testing system is somewhat complex, and the testing system has
considerable safety and economic implications.

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NOMENCLATURE

In this section, we define the nomenclature used in the main body of this report and in Appendices A and B.

A = Surface area of active (unshielded) portion of inner cylinder

C_T = Coefficient of viscous damping

C_1 = Parameter of Ramberg-Osgood equations; Parameter of liquefaction and degradation submodel

C_2, C_3, C_4 = Parameters of liquefaction and degradation submodel

DE = Energy lost in test soil due to viscous damping

D_i = Inner diameter of inner cylinder

D_o = Outer diameter of inner cylinder

D_r = Relative density

$\bar{D}_i$ = Inner diameter of circular shaft

$\bar{D}_o$ = Outer diameter of circular shaft

E_R = Energy ratio

e = Void ratio

F_i = Rotational flexibility of inner cylinder relative to rotational flexibility of test soil during initial cycle of loading

f = Frequency in cycles per second

G = Tangent shear modulus of test soil

G_i = Shear modulus of inner cylinder

G_{mo} = Undegraded low amplitude shear modulus

G_o = Low amplitude dynamic shear modulus

$\bar{G}$ = Shear modulus of circular shaft

g = Acceleration due to gravity

h_c = Thickness of wall of inner cylinder (mass moment of inertia calculations)

h_s = Thickness of segment of test soil (mass moment of inertia calculations)

I = Total mass moment of inertia of effectively rotating mass

$\bar{J}$ = Polar moment of inertia of cross section

KE = Instantaneous kinetic energy of rotating mass

K_o = Coefficient of earth pressure at rest

K_T = Tangent torsional spring constant

K_{To} = Low amplitude torsional spring constant

k_2 = Parameter of liquefaction and degradation submodel

$\bar{L}$ = Length of circular shaft

l = Length of active (unshielded) portion of inner cylinder

l_{IH} = Height of instrumented head (mass moment of inertia calculations)

l_c = Length of inner cylinder (mass moment of inertia calculations)

l_s = Length of segment of test soil (mass moment of inertia calculations)

m = Parameter of liquefaction and degradation submodel

N_c = Number of cycles

N_q = Bearing capacity factor

N_γ = Bearing capacity factor

n = Parameter of liquefaction and degradation submodel

R = Parameter of Ramberg-Osgood equations

R_{IH} = Radius of instrumented head (mass moment of inertia calculations)

R_c = Radius of inner cylinder (mass moment of inertia calculations)

R_s = Radius of segment of test soil (mass moment of inertia calculations)

r = Radius

r_i = Outer radius of inner cylinder

r_o = Inner radius of outer cylinder

SE = Sum of energy lost in test soil due to hysteresis and instantaneous elastic strain energy stored within test soil

S_u = Rotational stiffness of inner cylinder relative to rotational stiffness of test soil during ultimate cycle of loading

T = Torque applied to inner cylinder

T_A = Amplitude of cyclic torque applied to inner cylinder

T_D = Damping torque

T_s = Spring torque

T_{sA} = Amplitude of cyclic spring torque

$\bar{T}$ = Static torque acting throughout circular shaft

t = Time; Degradation parameter

u = Horizontal displacement

$\bar{u}$ = excess porewater pressure

W = Work done by applied load

z = Depth below surface of ground

α = Parameter of Ramberg-Osgood equations

γ = Shear strain; Amplitude of cyclic shear strain

γ_A = Amplitude of cyclic shear strain

γ_{IH} = Weight per unit volume of instrumented head

γ_c = Weight per unit volume of inner cylinder

γ_r = Shear strain developed in test soil in horizontal planes

γ_{rA} = Amplitude of cyclic shear strain developed in test soil in horizontal planes

γ_{ruA} = Peak shear strain developed in test soil in horizontal planes during ultimate cycle of loading

γ_s = Total weight per unit volume of test soil

$\bar{\gamma}$ = Effective weight per unit volume of soil

ζ = ratio of damping to critical damping

$\Theta, \dot{\Theta}$ = Rotation and rotational velocity of rigid inner cylinder

Θ_A = Amplitude of cyclic rotation of rigid inner cylinder

Θ_{AL} = Limiting amplitude of cyclic rotation of rigid inner cylinder

Θ_i = Static twist of active portion of flexible inner cylinder

Θ_{si} = Static rotation of rigid inner cylinder embedded in undegraded test soil due to static torque, T

Θ_{suA} = Amplitude of low frequency (inertia and viscous damping forces negligible), cyclic rotation of rigid inner cylinder embedded in fully degraded test soil caused by low frequency, cyclic torque having amplitude, T

$\bar{\Theta}$ = Static twist between ends of circular shaft

ρ = Mass density of soil

$\bar{\sigma}_o$ = Average effective confining pressure

σ_o' = Initial effective vertical stress

$\bar{\sigma}_v$ = Effective vertical stress

$\bar{\sigma}_{vi}, \sigma_{vi}'$ = Initial effective vertical stress

τ = Shear stress; Amplitude of cyclic shear stress

τ_A = Amplitude of cyclic shear stress

τ_h = Amplitude of cyclic shear stress applied to horizontal planes

τ_m = Average of shear strengths in horizontal and vertical planes

τ_{mo} = Undegraded shear strength

τ_r = Shear stress developed in test soil in horizontal planes

τ_{rA} = Amplitude of cyclic shear stress developed in test soil in horizontal planes

τ_{rs} = Shear stress developed in test soil, in horizontal planes, as a result of shear strain

$\bar{\phi}$ = Effective angle of internal friction

ω = Circular frequency in radians per second

ω_n = Damped natural circular frequency in radians per second

INTRODUCTION

This report, to the National Science Foundation, covers work performed during Phase I of a planned three phase project. The purpose of the project is to develop an in situ testing procedure for determining in situ cyclic and dynamic soil properties. Phase I was a feasibility study, Phase II will be a research phase, and Phase III will be a commercialization phase. Phase I work was supported by a Small Business Innovation Research (SBIR) grant awarded to our firm, Dynamic In Situ Geotechnical Testing, Inc. The project falls under the domain of the Earthquake Hazard Mitigation program.

In the remainder of this report, we present and discuss, in detail, the purpose of the project, existing methods, the proposed method and its potential benefits, Phase I objectives, procedures, and accomplishments, and our estimate of feasibility. We also present technical definitions and terminology, discuss potential commercial applications and users, identify recipients of this report, provide references, and provide supplementary information in the appendices.

PURPOSE OF PROJECT

In this section, we state the purpose of our project. We also summarize potential benefits, relevant definitions, and background information.

The purpose of our project is to develop an in situ geotechnical testing procedure for obtaining accurate and detailed descriptions of in situ cyclic and dynamic engineering soil properties. This information is required for earthquake analyses involving foundation soils. The level of accuracy and detail to be provided is intended to be appropriate for the intermediate and final stages of the analysis and design of important structures located in seismically active areas. Based on our experiences, we feel that there are no existing procedures for determining most of the soil properties of interest in the detail and to the accuracy that is needed for the effective use of the costly and potentially powerful analysis procedures used at the later stages of analysis and design.

If successfully completed, the project should advance earthquake hazard mitigation technology by advancing our ability to economically insure the integrity and reliability of soil-structure-equipment systems during earthquakes. These systems include earth structures (dams, embankments, etc.) as well as concrete and steel structures (buildings, offshore platforms, etc.). The testing procedure is intended to reduce potential for error and uncertainty in predicting site-specific earthquake ground motions and system response and stability. This should result in greater safety, reliability, and economy.

Specifically, we are developing an in situ testing procedure for defining accurately and in detail the in situ liquefaction and cyclic degradation characteristics of a soil, the low amplitude dynamic shear modulus, and the variation in the dynamic shear modulus with shear strain.

Herein, liquefaction is defined as the almost total loss of the stiffness and strength of a sand or silt. Liquefaction is caused by the buildup of excess porewater pressure in the soil. Degradation, a related phenomenon, which may occur in sands, silts, and clays, is the reduction in the stiffness of a soil. Both phenomena may be caused by cyclic loads resulting from earthquakes. The dynamic shear modulus is the shear stiffness of a material under dynamic loading. In soils, this property varies considerably with shear strain (24).

These soil properties are important because they significantly affect the excitation, response, and stability of soil-structure-equipment systems during earthquakes. Liquefaction primarily affects stability. It does so, for example, by reducing the bearing capacity of foundation soils, the integrity of earth structures, and the resistance of pile-founded structures to lateral loads. Liquefaction has caused considerable damage to soil-structure-equipment systems during recent earthquakes, notably in Alaska, 1964, and in Niigata, 1964 (28) (38). The low amplitude dynamic shear modulus of a soil deposit and the variation in the dynamic shear modulus with shear strain affect the natures of both the excitation to and the response of systems. The response of an earth structure is additionally affected by the dynamic shear modulus of the structure itself. Cyclic degradation of a soil during earthquakes affects excitation, response, and stability. Degradation

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changes the stiffness of a soil deposit during an earthquake, and thus, changes the nature of the excitation to and the response of a system. Degradation changes the stiffness of an earth structure itself, and thus, additionally affects the nature of its response. Degradation, in a manner similar to liquefaction, but to a lesser degree, also affects the stability of soil-structure-equipment systems.

Because of the importance of these soil properties, they are required for earthquake analyses which predict the excitation to, response of, and stability of soil-structure-equipment systems. Such analyses serve as a basis for earthquake resistant design and include: 1) earthquake site response analyses used to estimate liquefaction potential and to obtain site-specific ground motion histories and design response spectra, 2) earthquake soil-structure interaction analyses used to obtain the response of soil-structure system or the excitation to supported equipment, and 3) earthquake stability analyses used to predict the stability of natural slopes or the slopes of earth structures. Examples of earthquake analysis procedures are DESRA (12) and CHARSOIL (35).

Uncertainty and error in defining in situ soil properties will create uncertainty and error in the results of analyses relying on these properties. In fact, the limitations in our ability to accurately and descriptively define in situ soil properties can severely limit the effectiveness of the costly and potentially powerful analysis procedures which have been developed during recent years. To be effective, such analysis procedures, which are most appropriate for the intermediate and final stages of the analysis and design of important structures located in seismically active areas, require detailed and accurate descriptions of the in situ properties of interest.

By improving our ability to define the in situ soil properties of interest, in both accuracy and detail, we will reduce uncertainty and potential for error in the results of earthquake analyses. This will allow, in particular, the potential benefits of costly and potentially powerful analysis procedures to be realized to a far greater degree. This will result in greater safety, reliability, and economy of soil-structure-equipment systems constructed in seismically active areas and may eliminate earthquake resistance as a barrier to the feasibility of construction at a site.

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TECHNICAL DEFINITIONS AND TERMINOLOGY

In this section, we define and explain important technical terms used in this report. These terms include the low amplitude dynamic shear modulus, the variation in the dynamic shear modulus with shear strain, relative density, the degradation and liquefaction characteristics of sands and silts, and the degradation characteristics of clays.

Low Amplitude Dynamic Shear Modulus, G_o.--Referring to Fig. 1, the low amplitude dynamic shear modulus is the slope, at low levels of shear strain (<0.001%), of the nonlinear shear stress-strain curve for a soil loaded dynamically. Generally, stress-strain behavior of soils at these levels of shear strain is linear. Reference 25 provides more detail.

Variation in Dynamic Shear Modulus with Shear Strain--As shown in Fig. 1, high levels of shear strain induce highly nonlinear shear stress-strain behavior in soils. As a result, the tangent shear modulus, G, may vary considerably with shear strain. Reference 24 provides more detail.

Relative Density, D_r, of a Sand or Silt--The relative density of a sand or cohesionless silt is a measure of the compactness of the arrangement of its grains. Relative density is indicated with a scale of 0 to 100%. A sand with a relative density of 0% is a structurally "loose" sand in its least compact state. A sand with a relative density of 100% is a structurally "dense" sand in its most compact state. The relative density of a sand or silt has a strong influence on its cyclic degradation and liquefaction characteristics. Generally, sands or silts with lower relative densities (<50%) have little resistance to degradation and liquefaction while sands or silts with higher relative densities (>70%) have high resistance. Reference 28 provides more detail.

Degradation and Liquefaction Characteristics of Sands and Silts--Under large, earthquake-type, cyclic shear loads, because of buildups in excess porewater pressure, the shear stiffness (slope of straight line through peaks of shear stress-strain curve) of a sand or silt may decrease with an increase in the number of cycles of loading. This cyclic decrease in shear stiffness is referred to as cyclic degradation. As shown in Fig. 2, the degradation of an element of soil, under a cyclic shear load having a uniform amplitude, may be observed as an increase in the amplitude of the resulting cyclic shear strain of the element with an increase in the number of cycles of loading. Under loadings corresponding to large earthquakes, loose sands will generally show severe degradation while dense sands will show only mild degradation. After a sufficient number of cycles of loading, the loose sands may undergo unrestrained deformation, which is termed liquefaction. Dense sands will not liquefy because of the restraining effects of dilation, the expansion of the volume of the structure of a soil due to shear load. Rather, a dense sand will show only limited deformations regardless of the number of cycles of loading. The frequently used term, initial liquefaction (28), identifies the instant, during cyclic loading, when the excess porewater pressure of an element of soil first rises to the level of the initial effective confining pressure. Initial liquefaction may occur in loose and in dense sands. References 28 and 29 provide more detail.

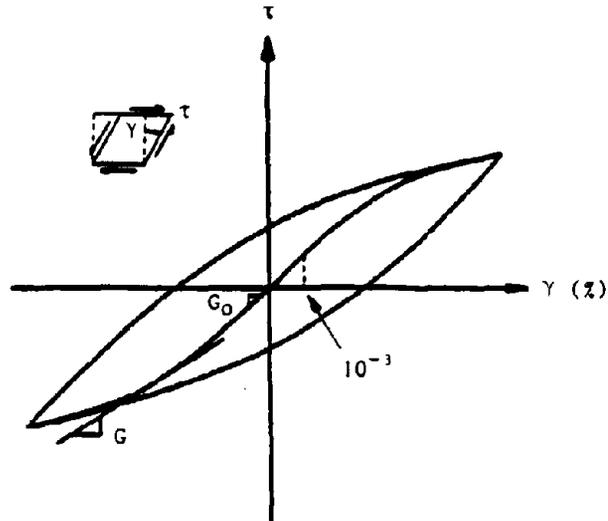**Degradation Characteristics of Clays**--Under large, earthquake-type,

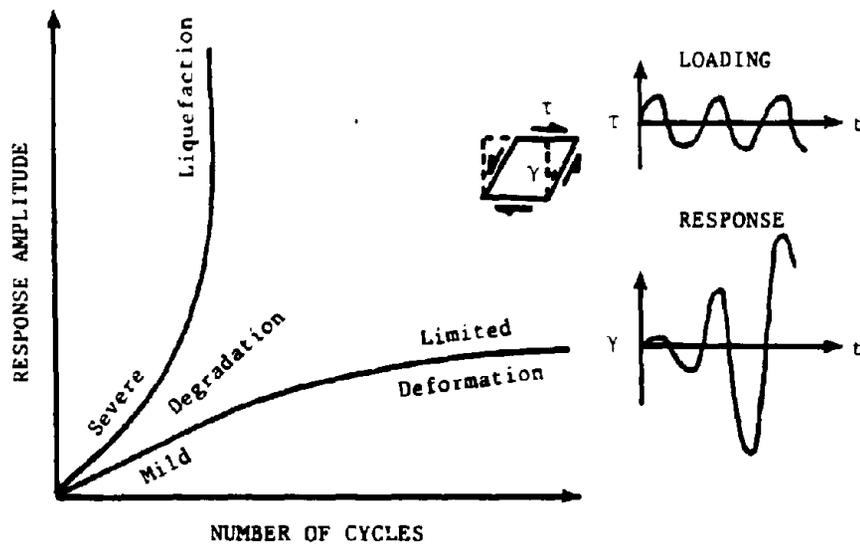Figure 1:   Nonlinear Shear Stress-Strain Curve for Soils

Figure 2:   Degradation and Liquefaction

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cyclic shear loads, because of buildups in excess porewater pressure and structural breakdown (21), the shear stiffnesses of clays may degrade similarly to those of sands and silts. Clays, however, generally do not liquefy (29). Instead, like dense sands, clays usually develop limited deformations after a large number of cycles of loading (see Fig. 2). References 20, 21, and 29 provide more detail.

EXISTING METHODS

In this section, we critically review the existing methods for performing the functions of the proposed testing system. These functions include determining the cyclic degradation and liquefaction characteristics of a soil, the low amplitude dynamic shear modulus, and the variation in the dynamic shear modulus with shear strain.

We mainly discuss methods for determining degradation and liquefaction characteristics of soils. Providing in situ cyclic degradation and liquefaction characteristics, to a level of accuracy and detail appropriate for the intermediate and final stages of the analysis and design of important structures located in seismically active areas, is expected to be the main capability of the proposed testing system. Methods for determining in situ liquefaction and degradation characteristics are at an early stage of development and thus, involve considerable uncertainty and potential for error. Also, the consequences of uncertainty and error in estimates of in situ degradation and liquefaction characteristics are generally expected to be greater than those in estimates of dynamic shear moduli.

A number of significant advances have been made in the very active field of dynamic and cyclic soil testing (37); however, because of the need for safe', reliability, and economy in the massive or delicate soil-structure-equipment systems of the future, there is still considerable motivation for further advancement. The proposed testing procedure is intended to advance our ability to determine in situ cyclic and dynamic soil properties by overcoming some of the shortcomings of existing methods. These shortcomings can create error and uncertainty which can severely limit the effectiveness of costly and potentially powerful analysis procedures. Currently, the properties of interest are obtained either by laboratory testing of soil samples recovered from a site, in situ testing at the site, or a combination of these methods.

Laboratory testing is attractive because the soil properties of interest can be determined directly. That is, cyclic shear loads, representative of earthquakes, can be applied to samples to induce appropriate responses. Detailed information (for example, cyclic strain history), required at the intermediate and final stages of analysis and design, can be readily obtained, and laboratory testing is applicable to all soils. Additionally, laboratory testing offers the means to create conditions different from those existing in situ. However, laboratory testing suffers from the problem of disturbance to in situ conditions. This problem can lead to considerable uncertainty and potential for error in interpreted soil properties. We feel that this uncertainty and potential for error can be excessive for the intermediate and final stages of analysis and design and inconsistent with that introduced by analysis procedures appropriate for these stages.

Our experiences indicate that in situ conditions can be greatly disturbed and difficult to recreate. For example, during the recovery, transport, and test preparation of samples, loose sand samples may be densified significantly (28) leading to considerable uncertainty and potential for error in estimates of liquefaction resistance. In contrast, dense sand samples may be significantly loosened. Additionally, it is often difficult to recreate the in situ state of stress. This is because it is

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often difficult and costly to determine this state of stress. Even when the state of stress has been determined, commonly used laboratory tests frequently cannot adequately recreate the defined state of stress. This difficulty leads to even further uncertainty and potential for error in estimates of liquefaction resistance. While freezing of samples has been offered as a means to minimize disturbances, based on our experiences, this process is complicated and can itself introduce considerable uncertainty and potential for error. Also, the in situ state of stress and degree of saturation may not be preserved. Arulmoli et al. (2) suggest the use of electrical measurements to reconstruct a sample to its original in situ state. This procedure may be difficult for two reasons: 1) we would expect considerable uncertainty in relating electrical measurements taken in laboratory samples to those taken in situ when in situ conditions are complex (i.e., many factors such as age and cementation act on soil, Ref. 28), and 2) it would be very difficult to recreate complex in situ conditions in laboratory samples. In addition to uncertainty and potential for error in estimates of liquefaction resistance, disturbance to in situ conditions may also cause considerable uncertainty and potential for error in interpreted degradation characteristics and shear moduli.

In situ testing overcomes, to a degree, some of the problems with laboratory testing since in situ conditions may be better preserved. However, our experiences with in situ testing have given rise to concerns which are unique to each property of interest.

Several concerns arise in determining liquefaction and cyclic degradation characteristics of sands and silts by in situ testing, and as pointed out by Woods (37), "...in situ evaluation of liquefaction potential remains elusive." We feel that there are no in situ methods which can provide the information on in situ cyclic degradation and liquefaction characteristics to the level of accuracy and detail appropriate for the intermediate and final stages of the analysis and design of important structures located in seismically active areas. The basic problem of most methods is that they are indirect methods. That is, loads corresponding to earthquakes are not applied to elements of soil and appropriate responses are not induced. Cyclic degradation and liquefaction are very complex phenomena which are functions of many parameters. Thus, measuring these phenomena indirectly may result in considerable uncertainty in deriving single parameters to describe liquefaction and degradation characteristics (for example, amplitude of cyclic shear stress needed for initial liquefaction in 10 cycles of loading, Ref. 29). As a result of additional assumptions, even greater uncertainty may result in deriving detailed information regarding cyclic degradation and liquefaction characteristics (for example, increase in cyclic shear strain amplitude with an increase in the number of cycles of loading). In situ methods for determining liquefaction and degradation characteristics are discussed in the following paragraphs.

Currently, tests for determining in situ liquefaction resistance involve penetrating devices into the ground. Liquefaction resistance is estimated from penetration resistance (29). Penetration tests are useful for the preliminary stage of analysis or design. Additionally, penetration tests may be used to extrapolate detailed and accurate information over a site and to

identify locations where further detailed and accurate information should be obtained.

However, penetration tests would not be expected to provide information on cyclic degradation and liquefaction characteristics to a high level of certainty. With penetration testing, one concern, believed to cause considerable scatter in data relating penetration and liquefaction resistances, is that the loading of the soil and its mode of failure are considerably different from those induced by earthquakes. In the cone penetrometer test (CPT), a cone is slowly pushed into the soil, and in the standard penetration test (SPT), an open cylinder is violently driven into the soil. In each case, the failure of the soil is immediate and severe. In contrast, earthquake-induced loads are cyclic and generally of lower intensity. The soil fails by gradually losing its resistance to load over a number of cycles of loading. A second concern, resulting from the fact that penetration resistance is an indirect measure of liquefaction resistance and also believed to cause considerable scatter in data relating liquefaction and penetration resistances, is the effect of the numerous factors such as age, past seismic history, and cementation which can affect the properties of a soil (28). Although it is argued that these factors affect the two resistances in directionally the same manner (28), because of the differences in loading and failure discussed above, the factors are unlikely to affect both resistances to the same degree. Thus, it is easily conceived that two soils having a given penetration resistance may have two different liquefaction resistances. As a result of scatter and the need for safety, design curves relating liquefaction and penetration resistances are very conservative. Use of such curves may lead to excessively, and possibly prohibitively, costly designs. Furthermore, correlations are not available for all soils which could liquefy (for example, calcareous soils).

New in situ approaches for inferring in situ degradation and liquefaction characteristics are being advanced. Sasaki and Koga (27) propose the use of a vibrating cone penetrometer to estimate liquefaction resistance. The force required to advance a vibrated cone penetrometer at a controlled rate or the difference between the forces required to advance the cone penetrometer with and without vibrations is taken as an indicator of liquefaction resistance. The static component of this test (penetration without vibrations) is expected to share the advantages and disadvantages of the CPT discussed in the previous paragraph. However, the vibratory component is not expected to offer a strong advantage over the conventional CPT in evaluating liquefaction resistance. While vibratory loads are more appropriate for estimating the potential for earthquake-induced liquefaction, the results from the vibratory component of the vibratory cone penetrometer test would be expected to be dominated by the cyclic properties of the zone of soil immediately adjacent to the tip and sides of the cone. This zone would be highly disturbed as a result of displacement by the cone. This displacement would be expected to alter greatly the most important factors affecting liquefaction resistance: relative density, structure, the in situ state of stress, cementation, effects of stress history, etc. (28).

Dobry et al. (10) and Stokoe and Nazarian (34) propose the use of in situ shear wave velocity as an indicator of in situ liquefaction susceptibility. Dobry et al. (10) combine shear wave velocity measurements

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with the concept of the strain approach. This results in a convenient, simplified procedure for predicting the threshold ground acceleration below which excess porewater pressures will not develop. Shear wave velocity, used with or without the strain approach, is potentially a useful indicator of liquefaction susceptibility appropriate for the preliminary stage of analysis or design. Additionally, shear wave velocity measurements, which may be conveniently made (at least for onshore sites) (34), are potentially useful for extrapolating detailed and accurate information over a site and identifying locations where further detailed and accurate information should be obtained.

However, approaches based on shear wave velocity would not be expected to provide detailed information on cyclic degradation and liquefaction characteristics to a high level of accuracy. Shear wave velocity, like penetration resistance, is an indirect measure of liquefaction resistance during earthquakes. The low levels of loading generally induced in a soil by shear waves and the responses developed do not generally correspond to the loads and responses developed during large earthquakes. Thus, like the use of penetration resistance, the use of shear wave velocity as a measure of liquefaction resistance during earthquakes would be expected to involve excessive uncertainty for higher stages of analysis and design. This is supported by the work of DeAlba et al. (8). They observed, from laboratory tests, that samples may show similar shear wave velocities but very different resistances to liquefaction. Additionally, it is difficult to determine shear wave velocity offshore; however, considerable effort is being directed toward this problem (33) (26).

Castro et al. (5) propose the use of an in situ vane shear test to provide an indication of in situ liquefaction resistance. The difference between the in situ shear resistance of a soil at high strain, and the shear resistance of the soil at high strain under a constant confining pressure corresponding to the initial in situ pressure, is taken as an indicator of liquefaction resistance. This difference provides a measure of the in situ potential for the contraction or dilation of a cohesionless deposit under shear loads. This procedure, although slightly more complicated than other in situ index tests because of the need for additional information, is potentially a useful indicator of liquefaction resistance appropriate for the preliminary stage of analysis or design. The procedure is also potentially useful for extrapolating detailed and accurate information over a site and identifying locations where further detailed and accurate information should be obtained.

However, this approach would not be expected to provide detailed information on cyclic degradation and liquefaction characteristics to a high level of accuracy. Like the other index tests discussed, this method is indirect. The high levels of noncyclic shear loads applied to the soil and the noncyclic responses developed do not correspond to the loads and responses developed during earthquakes. Thus, this method would be expected to involve excessive uncertainty for advanced stages of analysis and design. While Castro et al. did apply cyclic, earthquake-type loads with the vane, they apparently did this only to compare static behavior before and after the application of the cyclic loads. Since this method has much in common with our proposed method, we mention further aspects of this method in the sections entitled **PROPOSED METHOD**, pg. 21, and **ESTIMATE OF FEASIBILITY**, pg. 83.

Arulmoli et al. (2) propose the use of electrical measurements to determine liquefaction and degradation characteristics. Detailed information concerning the in situ liquefaction and degradation characteristics of sandy soils may be derived from correlations between liquefaction and degradation characteristics and combinations of various electrical parameters. The electrical parameters are obtained from the in situ measurement of electrical quantities. The electrical parameters are potentially useful indicators of liquefaction resistance appropriate for the preliminary stage of analysis or design. Additionally, as with other index tests, the electrical method is potentially useful for extrapolating detailed and accurate information over a site and identifying locations where further detailed and accurate information should be obtained.

However, electrical measurements would not be expected to provide detailed information on cyclic degradation and liquefaction characteristics to a high level of accuracy. The electrical method is clearly an indirect method. Additionally, it is not apparent how in situ factors such as the in situ state of stress and cementation would be taken into account. For example, Arulmoli et al. (2) appear to indicate that electrical measurements are fairly insensitive to the state of stress. Thus, like other indirect methods, the electrical method would be expected to involve considerable uncertainty in estimating liquefaction and degradation characteristics. The level of uncertainty would be expected to be excessive for the intermediate and final stages of analysis and design.

Esashi et al. (11) and Briaud and Meyer (4) propose the use of cyclic pressuremeters for determining the cyclic degradation characteristics of soils. Using this procedure, cyclic lateral pressures are applied through a cylindrical rubber membrane to the cylindrical wall of the soil in which the pressuremeter is embedded. The cyclic lateral movement of the membrane is estimated. The procedure has two important features: 1) the procedure is direct, and 2) the procedure is conducted in situ and when using the self boring capability (1), would not be expected to introduce excessive disturbance to the test soil. Thus, assuming adequate analytical procedures exist for interpreting test results, the procedure would be expected to provide information appropriate for advanced stages of analysis and design. Because of the nature of the loading induced by the cyclic pressuremeter, it would be expected to be effective for predicting the behavior of foundations subjected to cyclic lateral loads generated from above the foundation (for example, behavior of offshore piles subjected to wave loading).

However, the cyclic pressuremeter is expected to have several significant drawbacks with respect to determining the behavior of soils during earthquakes. These drawbacks would be expected to create excessive uncertainty in deriving in situ cyclic degradation and liquefaction characteristics to the level of accuracy and detail appropriate for the intermediate and final stages of the analysis and design of important structures located in seismically active areas. The drawbacks include 1) apparent limitations on the intensity of the applied cyclic loads, 2) potentially excessive drainage from the test soil during testing, and 3) poorly defined boundaries for the test soil.

The intensity of the cyclic load which can be applied appears to be limited because the pressuremeter seems to be able to load the soil

effectively only after exceeding the original in situ lateral stress. Unloading below this level appears to result in unrepresentative stress-strain behavior (3). Thus, it would appear that the peak to peak cyclic shear stress which may be induced in the test soil by the pressuremeter may be somewhat less than 1/2 of that which could be induced by large earthquakes. This would be expected to limit considerably the ability of the pressuremeter to induce cyclic degradation and liquefaction and may be one reason why material degradation (see Degradation and Liquefaction Characteristics of Sands and Silts, pg. 4) does not seem to be particularly significant during a cyclic pressuremeter test in a highly plastic clay (Fig. 2, Ref. 3). This may also be one reason why results from cyclic pressuremeter tests presented by Esashi et al. (11) do not appear to show the level of cyclic degradation expected in soils subjected to earthquake loadings. Their data indicates a decrease in the modulus of deformation of only about 30% during the first 20 cycles of loading when testing a "soft silt". After this the modulus stabilized. Their data also shows that the modulus of a "comparatively hard" soil remained unchanged over about 200 cycles of loading.

Excessive drainage would be expected during th cyclic testing of permeable sandy soils because the pressuremeter does not provide barriers to drainage. When using a vane shear device, Castro et al. (5) observed substantial drainage during somewhat comparable cyclic tests in sandy soils. The vane shear device also does not provide barriers to drainage. Generally, earthquakes can induce high levels of degradation and liquefaction because significant drainage does not occur leading to increased excess porewater pressures and decreased effective stresses. Thus, when cyclically testing sandy soils with the pressuremeter, the levels of cyclic degradation expected during earthquakes may not be observed. Additionally, the effects of volumetric changes of the soil structure permitted by drainage are expected to be complex and difficult to estimate. It could be easily visualized that under cyclic loading a freely draining soil could densify, and in the absence of high porewater pressure, actually increase in rigidity with an increase in the number of cycles of loading. Excessive drainage may partially explain the low levels of degradation which seem to have taken place during the tests conducted by Esashi et al. (11).

Finally, the poorly defined boundaries of the test soil may present some difficulty. Drainage, stresses, and strains may be distributed over a relatively large region in a complex manner. This may create uncertainties in analytical modeling. As a result, interpretation of test results in terms of the cyclic degradation and liquefaction characteristics of the test soil would be expected to be relatively difficult.

To our knowledge, there are no commonly used in situ testing procedures for determining the in situ cyclic degradation characteristics of clays to the level of accuracy and detail appropriate for the intermediate and final stages of the analysis and design of important structures located in seismically active areas. The level of effort that has been and still is being directed at determining the in situ behavior of sands and silts during earthquakes does not appear to have been directed at determining the in situ behavior of clays.

An example of recent work in determining the cyclic degradation characteristics of clays is the work of Briaud et al. (3). They discuss the

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testing of clays using a cyclic pressuremeter. As indicated previously, the cyclic pressuremeter offers considerable potential for certain important problems, but is expected to have drawbacks with regards to determining the behavior of soils during earthquakes. We feel the most serious drawback with regard to testing clayey soils would be the apparent limitation on the intensity of the applied cyclic stress and thus, on the potential to induce cyclic degradation. Drainage would not be expected to be a problem when testing clays using the pressuremeter. As a result, the difficulties presented by a poorly defined sample would not be expected to be as large when testing clays as when testing sands.

To our knowledge, there are no commonly used in situ testing procedures for determining the in situ variation in the dynamic shear modulus with shear strain. However, considerable effort has been directed toward developing this capability. References 1 and 37 discuss in detail past efforts, pointing out strengths and drawbacks of various proposed procedures.

Examples of recent work toward providing in situ testing procedures for determining the in situ variation in the shear modulus with shear strain include pressuremeter studies conducted by Esashi et al. (11) and Briaud et al. (3) and developmental work undertaken by Sidey et al. (31).

The pressuremeter appears to be a potential means for determining the variation in the shear modulus with shear strain. The greatest source of uncertainty in interpreting shear moduli from test results would be expected to be poorly defined sample boundaries. Since, in general, volume change, porewater pressure buildup, and drainage would not be expected to be dominant factors   in tests for determining shear moduli, this uncertainty may not be excessive.

Sidey et al. (31) presented feasibility and preliminary studies for a borehole shear device. This device also appears to be a potential means for determining the variation in the shear modulus with shear strain. With this system, which is conceptually similar in some respects to our proposed system, a single cylinder will be embedded into the test soil below the base of a borehole. The cylinder will be excited cyclically in torsion and the in situ variation in the shear modulus with shear strain will be inferred from the response of the system. Uncertainty would be expected in interpreting soil properties since the boundaries of the test soil will not be particularly well defined. However, as with the pressuremeter, this uncertainty may not be excessive. The system does not appear to be compact and portable; thus, the system would be expected to be difficult to use in harsh (for example, offshore) or confined (for example, indoors) environments.

There are a number of in situ methods in use for determining the in situ low amplitude dynamic shear modulus; however, each has drawbacks or limitations as discussed in detail in Ref. 37. For example, the widely accepted seismic crosshole test, in which the wave propagation velocity in a soil is measured, requires at least two boreholes; thus, this test may be relatively expensive and difficult to use in confined or harsh environments.

Recently, testing procedures have been advanced for determining the low amplitude dynamic shear modulus within a single borehole. These procedures

include wave propagation procedures such as one developed by the Oyo Corporation in Japan (no known reference) and one developed by K. H. Stokoe et al. (33). With the Oyo system, the wavespeed of waves propagated along the wall of a borehole is measured. Since these walls are generally quite disturbed from the drilling process, measured wavespeeds are not expected to correspond to wavespeeds in undisturbed soils. To our knowledge, the system developed by K. H. Stokoe et al. has not yet been applied in practice. An alternative procedure developed by Hardin (14) involves a harmonically rotating, cone-ended, torsional sleeve which may be either penetrated below the base of a borehole or lodged against the walls of the borehole. In either mode, the test soil is expected to be disturbed. The borehole wall would have been disturbed by the drilling process, and, in penetrating the cone-ended device below the base of a borehole, much of the soil in the test zone would have been displaced by the device.

PROPOSED METHOD

In this section, we describe the proposed testing procedure in some detail and discuss the major features of the procedure and various potential modes of operation.

The proposed testing procedure is intended to provide accurate and detailed descriptions of in situ cyclic and dynamic soil properties. This information is needed for earthquake analyses involving foundation soils. The level of accuracy and detail to be provided is intended to be appropriate for the intermediate and final stages of the analysis and design of important structures located in seismically active areas. Alternatively, the procedure could be used as an index test.

The testing procedure is intended to effectively combine attractive features of laboratory and in situ testing while minimizing shortcomings. The method will be direct. Cyclic earthquake-type shear loads will be applied to a well-defined element of soil in a simple but effective manner. The response of the test soil is expected to correspond closely to behavior during earthquakes. Thus, uncertainties associated with the loading of the soil and its mode of failure will be reduced. Tests will be conducted in situ and a number of steps will be taken to minimize disturbances to in situ conditions. Thus, the very important effects of in situ factors are expected to be captured in measurements. Several features are to be provided to help induce the phenomena of interest and to simplify the interpretation of test results. The information needed by earthquake analyses will be directly provided, minimizing intermediate interpretation. Also, the procedure is expected to apply to most soils of interest. Finally, the method will require only a single borehole so that it may be used in confined and harsh environments.

Figures 3 and 4 show, schematically, the main components of the probe of the testing system. Equipment above the probe is not shown. Using conventional equipment and procedures, two concentric, thin-walled cylinders will be carefully penetrated below the base of the borehole. The test soil will be the well-defined annular zone of soil between the two cylinders. A cyclic or impulsive torque or cyclic rotation of selected amplitude will be applied to the inner cylinder about its vertical axis to induce simple, earthquake-type shear stresses and strains in the test soil. In response, the cylinder is expected to rotate or develop torque in a manner dependent on the shear properties of the test soil. Both the torque and rotation will be measured by transducers in the instrumented head[1]. Secondary responses such as the buildup in excess porewater pressure could also be measured. The torsional shear stress-strain distributions that are expected to develop within the test soil, while not complex, will be nonuniform; thus, soil properties will be inferred by modeling tests analytically (soil-probe

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[1]De Domenico (9) was awarded a patent in 1982 for a two cylinder testing system for determining the static strength of soils. This system is configured similarly to ours but functions differently than ours. After penetration, the inner cylinder is removed leaving a pressurized membrane in its place. Thus, lateral pressure is imposed on the test soil. An annular piston located between the two cylinders applies a vertical load to the test soil causing the soil to fail statically.
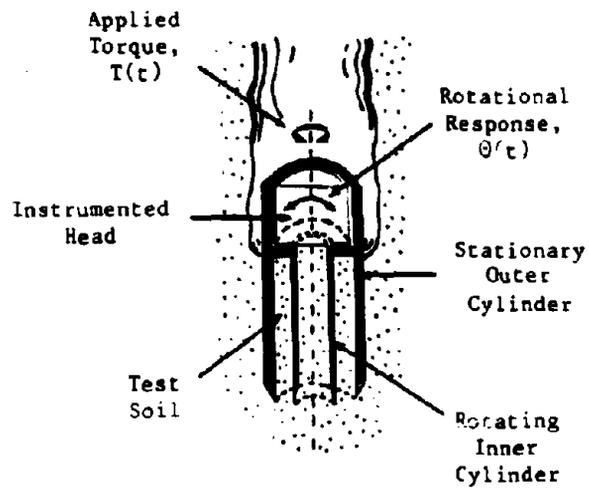
Figure 3:   Probe in Borehole Setting
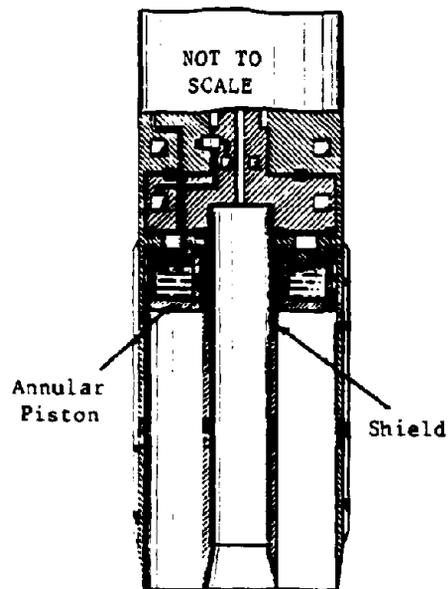(Main Elements Only)

Figure 4:   Detailed Schematic Diagram of Probe
(Patent Pending)

interaction analysis) using available, well-tested, and easily adaptable
analysis procedures. Soil properties will be iteratively assumed until
computed and measured results agree acceptably. The final assumed properties
are expected to closely represent the in situ properties.

For demonstrative purposes, Fig. 5 presents, schematically, what we
would expect to be the most commonly used testing sequence. First, an
impulse test would be conducted to determine dynamic shear moduli. An
impulsive torque would be applied to the inner cylinder. As shown in Fig.
5(a), the amplitude, frequency, and rate of decay of the oscillating
rotational response of the inner cylinder would be expected to be strongly
related to the shear stiffness of the test soil. For example, when testing
stiffer soils with higher shear moduli, we would expect the inner cylinder to
oscillate at lower amplitudes and higher frequencies.

After the impulse test, a low frequency, cyclic test would be conducted
to determine degradation and liquefaction characteristics. A cyclic torque
having a uniform amplitude would be applied to the inner cylinder. As shown
in Fig. 5(b), the rate of increase in the amplitude of the cyclic rotation of
the inner cylinder and the ultimate value of this amplitude would be expected
to be strongly related to soil characteristics. When testing highly
degradable, liquefiable soils (loose sands), under appropriate levels of
loading, we would expect rapid increases in the amplitude of rotation of the
inner cylinder with an increase in the number of cycles of loading. When
testing such soils, if the excess porewater pressure were to rise to the
level of the confining pressure, liquefaction, resulting in virtually
unrestrained rotation of the inner cylinder, would be expected. When testing
more easily liquefiable soils, we would expect liquefaction in fewer cycles.
As shown in Fig. 5(b), when testing moderately degradable, nonliquefiable
soils (dense sands, clays), we would expect only gradual increases in the
amplitude of the rotation of the inner cylinder with an increase in the
number of cycles of loading. When testing dense sands, because of the
restraining effect of dilation, the inner cylinder would not be expected to
undergo unrestrained rotation regardless of the level of excess porewater
pressure. Generally, unrestrained rotations would not be expected when
testing clays either.

Currently, the testing procedure is to consist of a number of important
features and to involve important steps to preserve in situ conditions. Both
inner and outer cylinders are to have thin walls. As shown in Figs. 3 and 4,
the penetrating edges of the cylinders will be shaped to minimize
disturbance to the test soil during penetration. The inside of the outer
cylinder and the outside of the inner cylinder will be coated with a low
friction material to minimize disturbance to the test soil caused by shear
during penetration and to minimize the development of residual shear stresses
in the test soil prior to testing. These surfaces are to be grooved
vertically to minimize slip during testing without excessively disturbing the
test soil during penetration. Additionally, we will relieve the penetration
force prior to testing to minimize static stresses developed in the test soil
as a result of penetration. We will also incorporate features to minimize
the influence of the soil within the inner cylinder on the motion of this
cylinder. The smooth inner wall of this cylinder will be coated with a low
friction material, soil will be diverted away from the inner wall by jutted
penetrating edges, shown in Fig. 4, and confining pressures acting on the
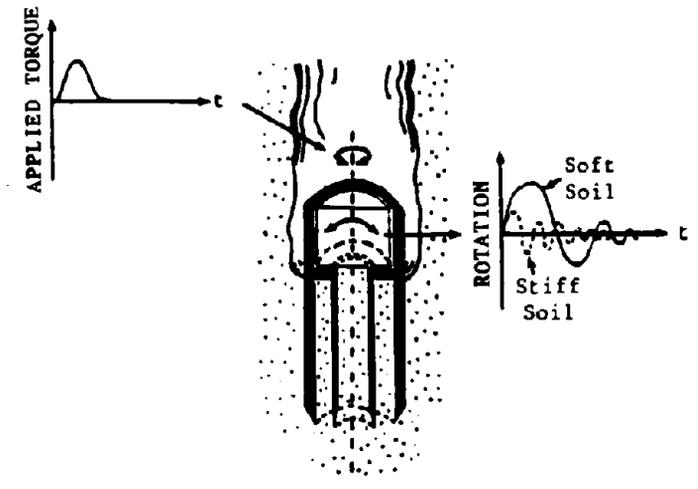
(a) Impulse Test
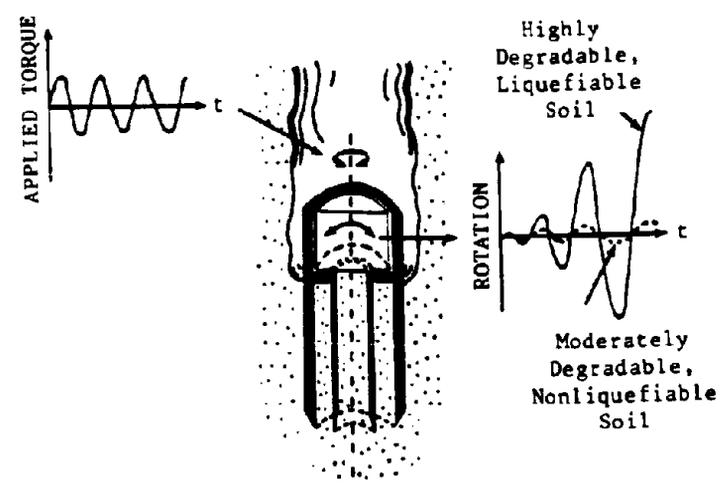
(b) Cyclic Test

Figure 5: Possible Testing Sequence

soil within the inner cylinder will be minimized by providing excess volume. The upper portion of the rotating inner cylinder will be shielded as shown in Fig. 4. Thus, only the soil some distance below the base of the borehole will be excited and effects of the disturbances near the base of the borehole will be reduced. An annular piston, shown in Fig. 4 and located between the two cylinders, will apply an appropriate vertical pressure to the top of the test soil to recreate, as closely as possible, the vertical state of stress existing prior to removal of soil from the borehole.
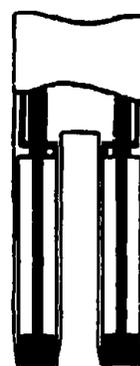
Two vertical pressure systems have been under consideration. The simplest system would use a short stroke piston as shown in Fig. 6(a). Using this system, a selected vertical pressure would be applied after penetration. Because of the low friction of the walls of the inner and outer cylinders, prior to the application of the vertical stress the test soil would be expected to be almost unloaded. After the application of the selected vertical pressure, because of the low friction and the rigidity of the walls of the cylinders, the lateral stress developed in the test soil would be expected to correspond closely to that developed in a normally consolidated level deposit. This is a very common practical condition and simple shear tests have been configured to simulate this condition. In many practical cases, the in situ lateral stress would be expected to either equal or exceed the lateral stress corresponding to this condition. Thus, information on liquefaction and degradation characteristics related to stability, obtained under this condition, would generally be conservative. Test results could be corrected, if necessary, for different lateral pressures as is commonly done in laboratory testing (28). Clearly, this would introduce uncertainty.

The second somewhat more complex system, shown in Fig. 6(b), will consist of a long stroke piston which will move along the entire length of the two cylinders. Similar piston systems have been used in samplers to prevent excessive deformations during sampling (36). With this system, the unloading of the test soil during penetration should be minimized and the original in situ stresses acting on the test soil should be preserved reasonably well. A selected vertical pressure would be applied at an early stage during penetration. Because of the low friction of the cylinder walls, elements of soil immediately ahead of the penetrating cylinders would be expected to be subjected to approximately the vertical stress existing prior to removal of soil from the borehole. The lateral stresses acting on these elements and the lateral deformations of these elements in response to the applied vertical stress would be expected to be reasonably close to those existing prior to the removal of soil from the borehole. The passage of the cylinders over an element would not be expected to alter the state of stress and deformation of the element greatly because of the geometry, low friction, and rigidity of the cylinders and the presence of the vertical pressure. Thus, with this system we would expect to preserve the in situ state of stress reasonably well. Additionally, pressurizing the test soil early during penetration is expected to reduce penetration-induced changes in void ratio.

We plan to use the simpler system shown in Fig. 6(a) during the initial stages of research. During these stages, identifying the torque-rotation characteristics of the testing system is of primary concern. This can be done more effectively using the simpler vertical pressure system. Should the testing system prove to be effective with the simpler system, we would then

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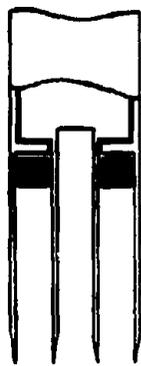
(a) Short Stroke Piston   (b) Long Stroke Piston
        System   System

Figure 6: Vertical Pressure Systems

develop the second, more refined system.

Several features of the proposed testing system are expected to help induce the main phenomena of interest, cyclic degradation and liquefaction. We expect to be able to apply shear loads to the test soil equally effectively in either direction. Thus, there should not be test-imposed limitations on the peak to peak amplitudes of the cyclic excitations which can be applied. Therefore, we should be able to induce reasonably well the cyclic degradation expected during earthquakes. The outer cylinder and the piston are intended to impede the flow of water from the test soil during cyclic tests carried out on permeable sands and silts. This should promote the buildup in excess porewater pressure and thus, the development of cyclic degradation and liquefaction. Alternatively, the outer and inner cylinders, and the piston may function to impose approximately constant volume conditions on the test soil. In this mode, instead of relying on excess porewater pressure, we would be relying on constant volume conditions (13) to bring about the cyclic changes in effective confining pressure which give rise to the cyclic degradation and liquefaction of sands and silts. Finally, the outer cylinder would serve to confine the excitation energy in a relatively small zone of soil. Thus, relatively high stresses should be obtainable with a minimum of power.

Also, several features of the proposed testing procedure are intended to simplify the interpretation of test results bringing the test within the descriptive capabilities of state-of-the-art analyses. The configuration of the probe is intended to promote the development of a relatively simple state of axisymmetric torsional shear stress and strain in the test soil. This state is expected to be relatively easy to model with respect to stresses, strains, and porewater flow. Additionally, the inner and outer cylinders, and the piston will provide relatively well-defined boundaries which are expected to be relatively easy to describe analytically. These boundaries will also help isolate the test soil from its surroundings. Thus, complex interactions between the test soil and its surroundings should be reduced. Such interactions were identified by Castro et al. (5) as possible explanations for certain trends in the results from vane shear tests in sands. Finally, the testing system will be designed to test relatively long samples so that effects of end conditions will be reduced.

We plan to use a descriptive soil-probe interaction analysis to accurately infer in situ soil properties from test results. References 17, 18, and 19, describe an applicable analysis procedure which was conceived and developed by the technical advisor of our firm. This multidimensional, dynamic, nonlinear, continuum analysis procedure has been proved capable of accurately simulating the torsional, dynamic behavior of an axisymmetric, soil-rigid body system. The analysis treats the soil as a nonlinear, inelastic continuum and permits slip between the continuum and the rigid body. This analysis will allow us to determine the low amplitude dynamic shear modulus and the variation in the dynamic shear modulus with shear strain. To be able to determine cyclic degradation and liquefaction characteristics, we will extend the soil modeling to describe the degradation of soil stiffness, liquefaction, and the flow of porewater within and out of the test soil during testing. Appropriate modeling for these extensions is discussed in Refs. 12, 20, and 22.

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Until appropriate models are well-developed, we will be unable to model precisely the behavior of dense sands and silts at high levels of excess porewater pressure. Current, commonly used, state-of-the-art models, such as the ones we plan to use, do not model dilation. At this time, we do not consider this limitation severely restrictive. The sands and silts that pose the greatest problems in providing earthquake resistant structures are looser sands and silts.

The proposed procedure should provide, with a minimum of intermediate interpretation, the information required by commonly applied earthquake analysis procedures such as DESRA (12) and CHARSOIL (35). This is because the soil modeling to be used in the soil-probe interaction analysis will be almost identical to that used in such analysis procedures.

Variations on tests are expected to be possible. Rather than an impulsive torque, a high frequency, cyclic torque could be applied to the inner cylinder. This would be intended to give resonant response from which shear moduli could be inferred. Similarly, low frequency, cyclic, controlled-rotation tests could be used to determine degradation and liquefaction characteristics of sands and silts, rather than low frequency, cyclic, controlled-torque tests.

Additionally, low frequency, cyclic tests involving sands and silts could be conducted in two different modes, a constant pressure mode and a constant volume mode. In the constant pressure mode, we would rely on excess porewater pressure to cause the changes in effective confining pressure which lead to the cyclic degradation and liquefaction of the test soil. In this mode, the pressure initially applied by the vertical pressure system would be maintained throughout a test. Tests would be carried out at frequencies high enough to permit the development of significant excess porewater pressure, yet low enough to allow the effective simulation of earthquakes. In the constant volume mode, we would rely on approximately constant volume conditions (13) imposed on the test soil to bring about the changes in effective confining pressure which lead to cyclic degradation and liquefaction. In this mode, after the application of the selected vertical pressure, the position of the piston would be maintained throughout a test. Some boundary flexibility would, however, exist at the upper and lower ends of the test soil. Tests would be carried out at frequencies low enough to avoid the development of significant excess porewater pressure.

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POTENTIAL BENEFITS

In this section, we discuss the main potential benefits of the proposed testing system. These benefits, which include technical, safety, and economic benefits, are first discussed in general and then in detail.

The proposed in situ testing procedure is intended to advance our ability to economically insure the integrity and reliability of soil-structure-equipment systems during earthquakes. It is to do so by providing more accurate and detailed descriptions of in situ cyclic and dynamic soil properties than can currently be provided. This would allow more effective use of the costly and potentially powerful analysis procedures used at the intermediate and final stages of the analysis and design of important structures located in seismically active areas.

Currently, the effectiveness of such analysis procedures is limited by our inability to define cyclic and dynamic soil properties accurately and in detail. Thus, defining these properties more accurately and in greater detail will reduce uncertainty and the potential for error in predicted site-specific earthquake excitations and predicted responses and stabilities of soil-structure-equipment systems during earthquakes. In turn, this will lead to greater safety, reliability, and economy of soil-structure-equipment systems.

Technically, the proposed testing procedure should provide more accurate and detailed descriptions of cyclic and dynamic soil properties by effectively combining attractive features of in situ testing and laboratory testing while minimizing shortcomings. The procedure will be direct. Cyclic earthquake-type shear loads will be applied to a well-defined element of soil in a simple but effective manner. The response of the test soil is expected to correspond closely to behavior during earthquakes. Thus, uncertainties associated with the loading of the soil and its mode of failure will be reduced. Tests will be conducted in situ and a number of steps will be taken to preserve in situ conditions. Thus, the very important effects of in situ factors are expected to be captured in measurements. Several features are to be provided to help induce the phenomena of interest and to simplify the interpretation of test results. The method should apply to most soils of interest. Also, by simulating tests analytically using models very similar to those used in earthquake analysis procedures, the method will provide, with a minimum of intermediate interpretation, the soil properties required for earthquake analyses. Finally, the method will require only a single borehole, so that it may be used in confined and harsh environments.

The technical benefits of the proposed testing procedure are expected to produce significant safety benefits. For example, a loose sand sample may densify an unknown amount because of disturbances during recovery, transport, and test preparation. As a result, the sample may show greater resistance to liquefaction in a laboratory test than the sample had in the field. The same sample may also lead to overestimates in dynamic shear moduli or degradation resistance. This may lead to unconservative estimates of site-specific earthquake ground motions for soil-structure-equipment systems with low natural frequencies (offshore structures, for example). Thus, laboratory testing may lead to unconservative estimates of soil behavior. In contrast, the proposed testing procedure, which is expected to minimize disturbances,

would be expected to reduce the possibility of unconservative estimates of soil behavior.

The technical benefits of the proposed testing procedure are also expected to produce significant economic benefits without a reduction in the targeted level of safety. For example, a dense sand sample may be loosened during recovery, transport, and test preparation. As a result, in a laboratory test, the sample may show less resistance to liquefaction than the sample had in the field. The same sample may also lead to underestimates in dynamic shear moduli and degradation resistance. This may lead to excessively conservative and costly predictions of site-specific earthquake ground motions for soil-structure-equipment systems with low natural frequencies (offshore structures, for example). Thus, laboratory testing may lead to excessively conservative and costly estimates of soil behavior. In contrast, the proposed testing procedure, which is expected to minimize disturbances, would be expected to reduce the possibility of excessively conservative estimates of soil behavior.

Additionally, the proposed testing procedure may affect feasibility decisions and design considerations. For example, using this procedure to more fully and accurately account for in situ factors may prove a liquefaction resistant structure to be economically feasible at a marginal site. Similarly, for a superior site, use of the procedure may eliminate the costly need to design and construct a structure to resist liquefaction. (The additional cost for designing and constructing a large offshore platform to resist a modest amount of liquefaction is conservatively estimated to be several million dollars.) These possibilities arise as a result of the very significant effects of in situ factors (age, stress history, etc.) on liquefaction resistance (28). For example, from Fig. 7 it may be inferred that in situ factors can increase the resistance of a soil to initial liquefaction by a factor of 2 to 3.5. Based on our experiences, we believe that such and even lesser factors can have dramatic effects on estimates of the potential for liquefaction and its extent. Thus, using a procedure such as the proposed procedure to preserve in situ conditions can be important to feasibility decisions and design considerations.
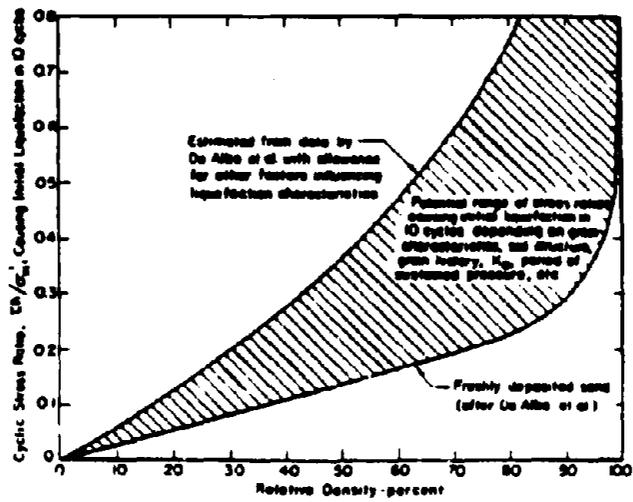
Fig 20 ESTIMATED RANGE OF LIQUEFACTION CHARACTERISTICS
FOR IN-SITU DEPOSITS

Figure 7. Effects of In Situ Factors on Resistance of Soil to Initial
Liquefaction (after Seed, 1976)

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POTENTIAL COMMERCIAL APPLICATIONS AND USERS

In this section, we discuss the potential commercial applications of the proposed testing procedure. We also identify potential users of the procedure or of the information provided by the procedure.

There are several important potential worldwide commercial applications of the proposed testing procedure. Generally, the procedure is expected to be useful for determining in situ cyclic and dynamic engineering soil properties before, during, and after the construction of soil-structure-equipment systems. The main application of the testing procedure is intended to be the design, construction, and maintenance of soil-structure-equipment systems to insure resistance to earthquakes. Additionally, the procedure is expected to apply to the design, construction, and maintenance of 1) soil-structure-equipment systems to insure resistance to water wave and blast loads, 2) machine foundations, and 3) delicate soil-structure-equipment systems to insure isolation from vibrations.

Also, there are a number of potential users of the proposed testing procedure or of the information provided by the procedure. Potential users include organizations involved in construction related activities, for example, agencies of governments, universities, geotechnical engineering firms, oil companies, and power companies.

RECIPIENTS OF REPORT

The Office of Energy-Related Inventions (OERI) of the National Bureau of Standards (NBS) is the only other organization, outside the NSF, which will receive copies of this report. Currently, the OERI is evaluating the proposed testing procedure for possible support from the Department of Energy (DOE). The proposed testing procedure has been advanced to the second stage of evaluation by the OERI.

PHASE I OBJECTIVES, PROCEDURES, AND ACCOMPLISHMENTS

In this section, we discuss, in detail, the Phase I research objectives, procedures, and accomplishments. The accomplishments cover the performances of the proposed theoretical feasibility study, a supplementary theoretical feasibility study, and the proposed operational feasibility study. First, the theoretical feasibility study is presented, then, the supplementary theoretical feasibility study is presented, and finally, the operational feasibility study is presented.

Theoretical Feasibility Study

Introduction and Summary

We conducted a theoretical feasibility study to determine the theoretical feasibility of the proposed testing system. Tests were simulated analytically considering expected ranges of selected soil properties. Studies were carried out to determine the theoretical feasibility of the testing system for obtaining 1) the low amplitude dynamic shear modulus, 2) the variation in the dynamic shear modulus with shear strain, 3) the degradation characteristics of clays, and 4) the degradation and liquefaction characteristics of sands and silts.

Results from our study indicate that the proposed testing system is theoretically feasible. The behavior of the testing system was predicted to be sensitive to each soil property considered in a clear, physically reasonable manner.

There were also secondary objectives of the theoretical feasibility study. These objectives were generally satisfied.

During our study, we identified one area requiring further study. The configuration of the inner cylinder, specified for the preliminary design of a laboratory research prototype testing system, was thought to be torsionally more flexible, relative to the test soil, than we had originally considered. As a result, we carried out a supplementary cursory study presented in the subsection entitled Supplementary Theoretical Feasibility Study, pg. 63. The study indicates that the inner cylinder may be torsionally flexible under certain conditions. This may increase somewhat uncertainty in inferring soil properties from test results. However, we concluded that because of the number of effective steps which could be taken, if necessary, to mitigate this concern, the proposed testing system should serve its stated purpose well.

Details are provided in the remaining subsections. We state the objectives and general procedures of the theoretical feasibility study, present and discuss analysis procedures used in the study, present and discuss results from the study, and present conclusions.

Objectives

The main objective of the theoretical feasibility study was to determine the theoretical feasibility of the proposed testing system for determining 1) the low amplitude dynamic shear modulus, 2) the variation in the dynamic shear modulus with shear strain, 3) the degradation characteristics of clays,

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and 4) the degradation and liquefaction characteristics of sands and silts.

There were also several secondary objectives of the theoretical feasibility study. These objectives were 1) to provide the design criteria needed for the detailed design of the laboratory research prototype testing system scheduled for Phase II, 2) to develop an initial testing methodology for Phase II laboratory research testing, and 3) to develop a simple but effective analysis procedure for qualitatively checking, during Phase II, the results from laboratory research tests and developmental results from descriptive soil-probe interaction analyses, to be constructed as part of Phase II research.

General Procedures

The theoretical feasibility of the proposed testing system was determined by simulating tests analytically considering expected ranges of selected soil properties. If we predicted the behavior of the testing system to be sensitive to a soil property in a clear, physically reasonable manner, then we concluded that we could reasonably infer this property from the behavior of the testing system and therefore, that the proposed testing procedure is a theoretically feasible means for determining this property.

The secondary objectives of the theoretical feasibility study were pursued in the following manners. We developed preliminary design criteria needed for the preliminary design of the laboratory research prototype testing system. We provided these criteria to the mechanical engineering firm designing the system. Using these criteria, the firm developed a preliminary design of the system. Using the preliminary design as a basis, we had planned to revise our criteria to final design criteria, but we did not carry out this last task. Therefore, this task will be proposed for Phase II research. To satisfy the objective of developing an initial testing methodology for Phase II testing, we reviewed the results of our analytical simulations. Based on this review, we identified the most promising modes of testing. Finally, the performance of the theoretical feasibility study automatically satisfied the objective of developing a simple, but effective, analysis procedure for use in Phase II research.

Analysis Procedures

In the following subsections, we present and discuss the analysis procedures used for the theoretical feasibility study. Specifically, we present and discuss our analytical modeling, solution procedures, a solution check procedure, assumptions, and validations of solution procedures.

Analytical Modeling--As shown in Fig. 8, the analytical model developed for the theoretical feasibility study was a simple single-degree-of-freedom, torsional spring-dashpot-mass moment of inertia system. The spring represented the nonlinear, inelastic, degrading, torsional reaction of the test soil to rotation of the inner cylinder. The dashpot represented viscous damping of the test soil. The mass moment of inertia represented the rotational inertia of the inner cylinder, its instrumented head, and an appropriate portion of the test soil.

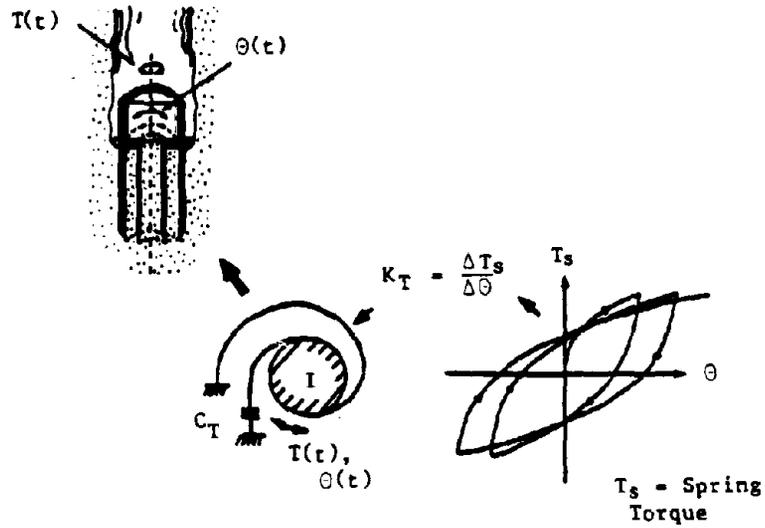The modeling involved the following elements: 1) a relationship between

Figure 8: Simple Nonlinear, Inelastic, Degrading Model of Soil-Probe System
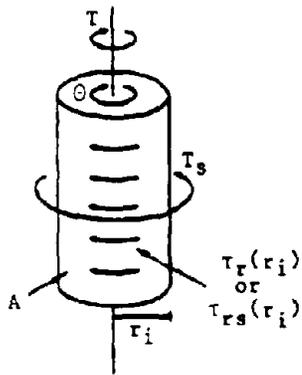

Figure 9: Schematic Diagram of Active (Unshielded) Portion of Inner Cylinder Showing Assumed Uniform Distribution of Shear Stress and Related Terms in Positive Sense
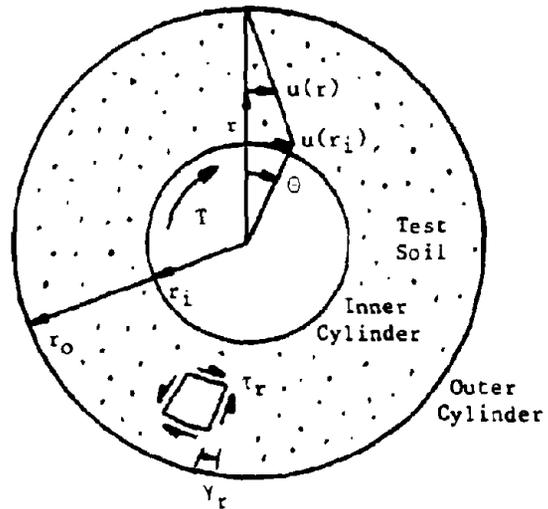
Figure 10: Horizontal Cut Through Probe Showing Assumed Linear, Horizontal Displacement Distribution and General Nomenclature

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the spring torque and the corresponding shear stress developed within the test soil along the wall of the active (unshielded) portion of the inner cylinder, 2) a relationship between the rotation of the rigid inner cylinder and the shear strain developed within the test soil along the wall of the active portion of the inner cylinder, 3) a relationship giving the tangent spring stiffness of the single-degree-of-freedom system, 4) relationships describing the nonlinear, inelastic, shear stress-strain behavior of the test soil, 5) relationships describing the cyclic degradation of shear stress-strain behavior observed in clays, 6) relationships describing the cyclic degradation of shear stress-strain behavior observed in sands and silts and the liquefaction of such soils, and 7) a relationship for the mass moment of inertia of the effectively rotating mass. These elements are summarized below. Detailed derivations for elements 1), 2), and 3) are provided in Appendix B.

Assuming a uniform distribution of shear stress in the vertical direction within the test soil along the wall of the inner cylinder (see Fig. 9), the following relationship was derived relating the change in the spring torque, ΔT_s, to the corresponding change in the shear stress in the test soil along the wall of the active portion of the inner cylinder, $\Delta \tau_{rs}(r_i)$:

$$\Delta \tau_{rs}(r_i) = - \frac{\Delta T_s}{2 \pi r_i^2 l} \tag{1}$$

where r_i = outer radius of inner cylinder
         l = active length of inner cylinder

A form of Eq. 1, valid when damping and inertia forces are negligible, gives the amplitude of the cyclic torque applied to the inner cylinder, T_A, as a function of the corresponding cyclic shear stress ratio, $\tau_{rA}(r_i)/\bar{\sigma}_{vi}$:

$$T_A = -2 \pi r_i^2 l \bar{\sigma}_{vi} \left(\frac{\tau_{rA}(r_i)}{\bar{\sigma}_{vi}} \right) \tag{2}$$

where $\bar{\sigma}_{vi}$ = initial effective vertical stress
         $\tau_{rA}(r_i)$ = amplitude of cyclic shear stress

We derived a relationship between the change in the rotation of the rigid inner cylinder, $\Delta \Theta$, and the change in the shear strain developed in the test soil along the wall of the inner cylinder, $\Delta \gamma_r(r_i)$. For this, we assumed an axisymmetric, linear, horizontal displacement distribution in the test soil, as indicated in Fig. 10, and no slip between the cylinders and the test soil. The derived relationship is given as:

$$\Delta \gamma_r(r_i) = -\left(\frac{r_o}{r_o - r_i} \right) \Delta \Theta \tag{3}$$

where r_o = inner radius of outer cylinder

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An expression for the tangent spring stiffness, K_T, was derived and is given as:

$$K_T = \frac{2\pi r_i^2 r_o l}{r_o - r_i} G(r_i)$$ (4)

where $G(r_i)$ = tangent shear modulus of test soil along wall of inner cylinder

The tangent shear modulus, $G(r_i)$, needed to define the spring constant, K_T, using Eq. 4, was derived from the shear stress-strain model describing the behavior of the test soil. We used two models to represent the undegraded, nonlinear, inelastic shear stress-strain behavior of the test soil: a Ramberg-Osgood model (24) and a hyperbolic model (12). The hyperbolic model was used in simulating tests for determining the degradation characteristics of clays and the degradation and liquefaction characteristics of sands and silts, while the Ramberg-Osgood model was used in simulating all other tests. The Ramberg-Osgood model requires the definition of the low amplitude dynamic shear modulus of the test soil, the shear strength, and parameters describing the shape of the shear stress-strain curve. The hyperbolic model requires only the definition of the low amplitude dynamic shear modulus and the shear strength.

To represent the cyclically degrading shear stress-strain behavior expected when conducting low frequency, cyclic tests in degradable clays, we constructed a clay degradation submodel. This submodel corresponds to the model presented in Ref. 20. Using this model, the cyclic degradation of shear stiffness and strength was computed as a function of cyclic strain amplitude and a degradation parameter describing the degradation characteristics of the test soil.

We used two models to represent the cyclically degrading shear stress-strain behavior expected when conducting low frequency, cyclic tests in degradable sands and silts which could possibly liquefy. One model was analytically-based and one was based on published test data.

The analytically-based method involved a sand and silt degradation and liquefaction submodel corresponding to the model presented in Ref. 22. This model is expected to predict, reasonably accurately, the behavior of loose, liquefiable sands and the behavior of dense, nonliquefiable sands at lower levels of excess porewater pressure. Using this model, excess porewater pressure was computed primarily as a function of cyclic strain amplitude and parameters describing the densification and rebound characteristics of the test soil. Shear moduli and strength were cyclically degraded in accordance with the computed excess porewater pressure. Liquefaction, the almost complete loss of the stiffness and strength of a soil, was predicted if the excess porewater pressure rose to the level of the initial effective overburden pressure. Alternatively, a shear failure could be predicted if the applied cyclic shear stress exceeded the cyclically degraded shear strength.

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We constructed the second model based upon published test data to more
realistically predict the behavior of dense sands at higher levels of excess
porewater pressure. One limitation with currently available sand and silt degra-
dation and liquefaction models, with which we are familiar, is that these models
do not fully take into account the restraining effects of dilation. These effects
are significant for dense sands and prevent the unrestrained deformation
which may develop in loose, liquefiable sands (see Fig. 2). The published
test data, which includes effects of dilation, relates the cyclic simple
shear stress applied to a commonly used test sand, the resulting cyclic shear
strain, and the relative density of the sand. Figure 11 taken from Ref. 28,
shows the test data. The figure gives the amplitudes of the cyclic shear
strains developed in large, freshly prepared samples of Monterey No. 0 sand
subjected to cyclic simple shear stresses having uniform amplitudes.
Specifically, Fig. 11(a) presents the limiting shear strain amplitude
developed as a function of relative density. In this special case of freshly
deposited and identically prepared samples tested in the same pressure
environment, relative density essentially determines degradation and
liquefaction characteristics. Figure 11(b) presents the shear strain
amplitude, developed after 10 cycles of loading, as a function of relative
density and shear stress ratio, τ_h/σ_o'. The quantities σ_o' and τ_h are the
initial effective vertical stress and the amplitude of the applied cyclic
horizontal shear stress, respectively.

The total mass moment of inertia was calculated assuming parameters for
the inner cylinder, the instrumented head, and the portion of the test soil
which we assumed would effectively rotate with the inner cylinder. Figure 12
presents assumed geometries and includes nomenclature. For our study, we
assumed a solid disk for the instrumented head and thin-walled cylinders for
the inner cylinder and the zone of soil assumed to be effectively rotating.
Thus, using basic principles of mechanics, the total mass moment of inertia,
I, was estimated using the following expression:

$$I = \frac{\pi \gamma_{IH} R_{IH}^4 l_{IH}}{2g} + \frac{2\pi \gamma_c R_c^3 h_c l_c}{g} + \frac{2\pi \gamma_s R_s^3 h_s l_s}{g} \qquad (5)$$

where
g = acceleration due to gravity
γ_{IH} = equivalent unit weight of instrumented head
γ_c = unit weight of inner cylinder
γ_s = total unit weight of test soil

The remaining symbols are defined in Fig. 12.

Solution Procedures—We used two procedures for obtaining solutions:
a dynamic procedure and a static procedure. The dynamic procedure was used
to simulate tests for determining dynamic shear moduli and initially to
simulate tests for determining cyclic degradation and liquefaction
characteristics of sands and silts using the analytical model. The static
procedure was used later to simulate tests for determining cyclic degradation
characteristics of clays and cyclic degradation and liquefaction
characteristics of sands and silts after it was established that dynamic
effects were small (see Appendix A, pg. A-14). The dynamic effects were small
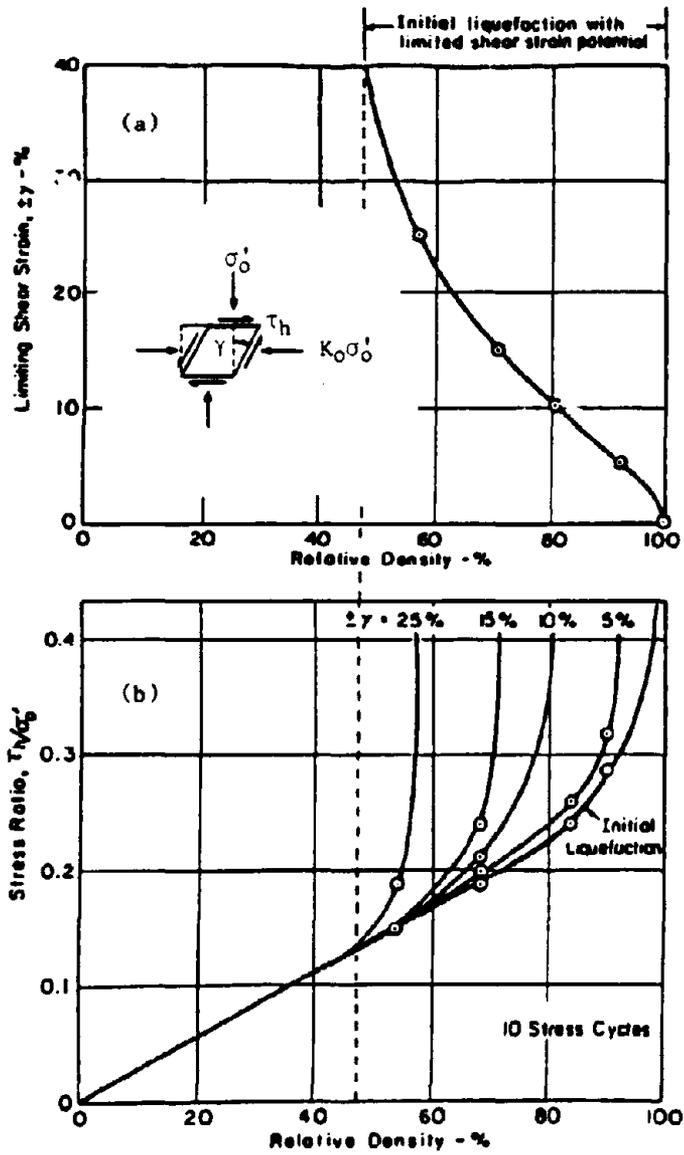
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Figure 11: Results from Large Scale, Cyclic, Simple Shear Tests Conducted on Freshly Prepared Samples of Monterey No. 0 Sand (after Seed, 1976)

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Figure 12: Vertical Cut Through Probe Showing Nomenclature and Geometries Assumed for Elements Involved in Mass Moment of Inertia Calculations

Figure 13: Relationships Between Elements of Simple Model

because of the low excitation and response frequencies considered (1cps).

With the dynamic procedure, we simulated tests by solving the equation of motion for the single-degree-of-freedom system. Because of the nonlinearity of the spring, we used a numerical integration procedure for solving the equation of motion. Specifically, we used the widely accepted step-by-step procedure described in Ref. 6. This procedure is based on the assumptions of a linear variation in acceleration and constant properties during computational time steps.

The basic steps used in the dynamic solution procedure are summarized with reference to Fig. 13. At the beginning of a computational time step, values are known for T, Θ, $d\Theta/dt$, $d^2\Theta/dt^2$, and K_T. These values are either specified as initial conditions, or are known from calculations for the previous time step. The values for $\gamma_r(r_i)$ and $\tau_{rs}(r_i)$ are obtained from Θ and K_T. Past and current values for $\gamma_r(r_i)$ and $\tau_{rs}(r_i)$ are used to obtain a value for the strain-dependent, possibly degraded, tangent shear modulus of the test soil along the wall of the inner cylinder, $G(r_i)$. The corresponding tangent spring stiffness, K_T, is obtained from this tangent shear modulus using Eq. 4. Values for $\Delta\Theta$ and $\Delta(d\Theta/dt)$ are obtained for the time step using the above-mentioned numerical solution procedure. These values are used to obtain new values for Θ, $d\Theta/dt$, and $d^2\Theta/dt^2$. The process is repeated until a solution has been obtained for the duration of interest.

With the static procedure, dynamic effects were neglected, and static equilibrium was satisfied for selected times during the cyclic loading. With this simplification, the torque applied to the instrumented head was equated to the torque applied to the test soil. When using the analytical models to describe cyclic degradation and liquefaction characteristics, we obtained solutions by satisfying equilibrium after each half-cycle of loading. The shear modulus and strength were degraded after each half-cycle. When using the test data published by Seed (28) to define the degrading shear-stress strain behavior of the test soil, we considered only a single time during any test. As stated previously, this data relates the amplitude of the cyclic simple shear stress applied to a test sand, the amplitude of the resulting cyclic shear strain, and the relative density of the sand. To simulate tests using the proposed testing system, we converted the stress amplitudes to equivalent applied torque amplitudes using Eq. 2 and the corresponding strain amplitudes to equivalent rotation amplitudes using Eq. 3. In the conversions, we assumed the behavior of the soil to be independent of the planes on which shear stresses develop. The converted data provided, for a range of cyclic torque loadings and a given time during a test, the predicted amplitude of the cyclic rotation of the inner cylinder as a function of relative density (degradation and liquefaction characteristics).

Solution Check Procedure—We constructed an energy balance procedure to help validate nonlinear dynamic solutions. The basic concept of the energy balance is presented in Refs. 16 and 39 and is summarized in this subsection.

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The key parameter of the energy balance is the energy ratio, $E_R(t)$, defined as:

$$E_R(t) = \frac{SE(t) + KE(t) + DE(t)}{W(t)} \qquad (6)$$

where $W(t)$ = work done on system by applied load
         $SE(t)$ = sum of energy lost in test soil due to hysteresis and instantaneous elastic strain energy stored within test soil
         $DE(t)$ = energy lost in test soil due to viscous damping
         $KE(t)$ = instantaneous kinetic energy of rotating mass

The individual terms are given as:

$$W(t) = \int_0^{\Theta(t)} T(t)\, d\Theta \qquad (7)$$

$$SE(t) = \int_0^{\Theta(t)} T_s(t)\, d\Theta \qquad (8)$$

$$DE(t) = \int_0^{\Theta(t)} T_D(t)\, d\Theta \qquad (9)$$

$$KE(t) = \frac{1}{2} I \left(\frac{d\Theta}{dt}\right)^2 \qquad (10)$$

where $T(t)$ = applied torque
         $\Theta(t)$ = rotation of inner cylinder
         $T_s(t)$ = spring torque
         $T_D(t)$ = damping torque
         I = mass moment of inertia of rotating mass

Equations 7, 8, and 9 may be effectively integrated by use of the trapezoidal rule.

At any instant of time during a simulation, the value of the energy ratio should equal one. That is, the work done on the system should equal the sum of the energy lost by the system and the instantaneous energy stored

by the system. A value other than one may indicate either an invalid solution or numerical error.

Assumptions--The simulations of tests involved several significant simplifications or assumptions: a lumped parameter system; a linear, axisymmetric, horizontal displacement distribution in the test soil in horizontal planes; uniform behavior in the vertical direction; independence in the behavior of the soil with respect to the planes on which shear stresses develop; no apparatus friction or viscous damping; and no slippage. Also, we did not model the flow of porewater caused by gradients in excess porewater pressure.

Even though the simple model was reasonably descriptive, because of the simplifying assumptions and because not all important phenomena were modeled, the model was expected to give results which were only qualitatively accurate. However, the level of accuracy provided was judged sufficient and appropriate for establishing feasibility and providing design criteria for a prototype testing system.

Validations--In this subsection, we briefly describe the validations which were carried out for the computer procedures used. The validations are presented in detail in Appendix A. The computer procedures validated included those which were used to simulate impulse and high frequency, cyclic tests for determining the low amplitude dynamic shear modulus and the variation in the dynamic shear modulus with shear strain, and low frequency, cyclic tests for determining the degradation characteristics of clays and the degradation and liquefaction characteristics of sands and silts.

Procedures were validated in various manners. Solutions were tested in the linear range by comparison with closed-form solutions. Because appropriate nonlinear, degrading, closed-form solutions were not found to be available, we validated the nonlinear, degradation, and liquefaction procedures by use of an energy balance (see Solution Check Procedure, pg. 36). We also studied solutions judgmentally to insure that they were reasonable.

Presentation and Discussion of Results

In this subsection, we present and discuss the main results of our theoretical feasibility study. To determine the theoretical feasibility of the proposed testing system, we simulated representative tests analytically considering expected ranges of selected soil properties. If the behavior of the testing system was predicted to be sensitive to a soil property in a clear, physically reasonable manner, then it was concluded that, theoretically, the soil property could be inferred from test results and therefore, that the testing system is a theoretically feasible means for determining this property. In the subsections that follow, we first discuss model parameters common to most simulations and then, we discuss, by property, the simulations.

Model Parameters Common to Most Simulations--Several model parameters were common to most simulations. All tests were simulated assuming a probe

having dimensions approximately compatible with commonly used drilling equipment. The outer cylinder was assumed to have an inner diameter of 3 in, and the inner cylinder, an outer diameter of 1 in. The active (unshielded) length of the inner cylinder was assumed to be 8 in and the shielded length, 4 in. Thus, the annular element of test soil was assumed to be 8 in tall and 1 in thick with an inner radius of 1/2 in. We roughly estimated a value of 0.00028 lb-sec^2-ft for the mass moment of inertia of the rotating mass. Referring to Fig. 12, for mass moment of inertia calculations, we assumed a 12 in long steel (weight per unit volume = 487 pcf) inner cylinder having an outer radius of 1/2 in and a wall thickness of 1/16 in, and, arbitrarily, a solid aluminum (weight per unit volume = 169 pcf) instrumented head having a height of 1/2 in and a diameter of 3 in. The test soil considered to be effectively rotating with the inner cylinder was assumed to be a cylindrical volume of soil (total weight per unit volume = 122 pcf) having a length of 8 in, a thickness of 1/2 in, and an inner radius of 1/2 in. We did not consider variations in the total weight per unit volume of the soil in our predictions because variations in this parameter were expected to have little effect on conclusions.

Low Amplitude Dynamic Shear Modulus--To determine the theoretical feasibility of the proposed testing system for determining the low amplitude dynamic shear modulus, we considered soil properties ranging from those expected in a uniform, saturated, loose sand deposit at a shallow depth of 5 ft, to those expected in a uniform, dry, dense sand deposit at a depth of 50 ft. This range of properties is expected to cover reasonably well the properties in deposits of clays and silts at similar depths. The depths considered cover the shallower layers of a deposit. These layers usually experience the greatest activity during earthquakes and therefore, require the most accurate definition for earthquake analyses. The lower bound of the range for the low amplitude dynamic shear modulus was calculated to be 5.14x10^5 psf while the upper bound was calculated to be 5.86x10^6 psf. These bounds were calculated assuming a depth, effective unit weight, void ratio, and coefficient of earth pressure at rest of 5 ft, 56 pcf, 0.85, and 0.6, respectively, for the lower bound, and 50 ft, 122 pcf, 0.35, and 1.0, respectively, for the upper bound. The equation used to calculate the low amplitude dynamic shear modulus is presented in Ref. 15. A value of viscous damping corresponding to 2% of critical damping was assumed. This is representative of sands (25).

We simulated both low amplitude impulse tests and low amplitude, high frequency, cyclic tests. The simulations of the impulse tests are discussed first, after which the simulations of the high frequency, cyclic tests are discussed.

For the simulations of low amplitude impulse tests, we considered a torsional impulsive loading having a rectangular shape and a duration greater than 1/2 the natural period of the system. Thus, the impulsive loading was of long duration (6). The response of the inner cylinder to this type of loading was a decaying rotational oscillation. The amplitude of the loading was 0.015 ft-lb. This induced only linear elastic behavior in the test soil for the range of low amplitude dynamic shear moduli considered. The duration of all loadings was 1.7 ms.

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Results are presented in Fig. 14. Figure 14(a) gives samples of the predicted measured quantities while Fig. 14(b) gives selected parameters of predicted measured quantities. The selected parameters include the peak and the frequency of the cyclic rotation of the inner cylinder. The peak rotation, in all cases, was the initial peak.

The results of the study indicate that, during low amplitude impulse tests, the behavior of the testing system will be sensitive to the low amplitude dynamic shear modulus in a clear, physically reasonable manner. Referring to Fig. 14(b), the frequency of the oscillation of the inner cylinder was predicted to be about 3 times larger when testing the most rigid sand than when testing the least rigid sand while the amplitude of the peak rotation of the inner cylinder was predicted to be about 10 times larger when testing the least rigid sand than when testing the most rigid sand. In fact, the peak rotation was predicted to be significantly more sensitive to the low amplitude dynamic shear modulus than the shear wave velocity (theoretically proportional to square root of low amplitude dynamic shear modulus) measured in widely used seismic wave propagation tests (in our example case, about 3 times more sensitive).

The results of this study are physically reasonable. One would expect, for a given low amplitude loading, the peak rotation of the inner cylinder to be greater and the frequency of oscillation to be less, when testing less rigid sand. Thus, it was concluded that conducting impulse tests using the proposed testing procedure is a theoretically feasible means for determining the low amplitude dynamic shear modulus.

For the simulations of low amplitude, high frequency, cyclic tests, we considered a sinusoidal torque excitation. The response of the inner cylinder to this loading was a cyclic rotation consisting of a decaying transient component and a steady state component. For a test soil of given properties, frequency response curves were developed in the vicinity of the damped natural frequency. The frequency response curves were developed by maintaining the amplitude of the loading while varying the frequency. The frequency response curves were based only on the steady state component of the response. The amplitude of the loadings was 0.001 ft-lb. This induced only linear elastic behavior for the range of low amplitude dynamic shear moduli considered.

Results are presented in Fig. 15. Figure 15(a) gives samples of predicted measured quantities while Fig. 15(b) gives selected parameters of the predicted measured quantities. The selected parameters include the damped natural frequency of the cyclic rotation of the inner cylinder and the amplitude of this rotation at the damped natural frequency.

The results of the study indicate that, during low amplitude, high frequency, cyclic tests, the behavior of the testing system will be sensitive to the low amplitude dynamic shear modulus in a clear, physically reasonable manner. Referring to Fig. 15(b), the damped natural frequency of the oscillation of the inner cylinder was predicted to be about 3 times larger when testing the most rigid sand than when testing the least rigid sand and
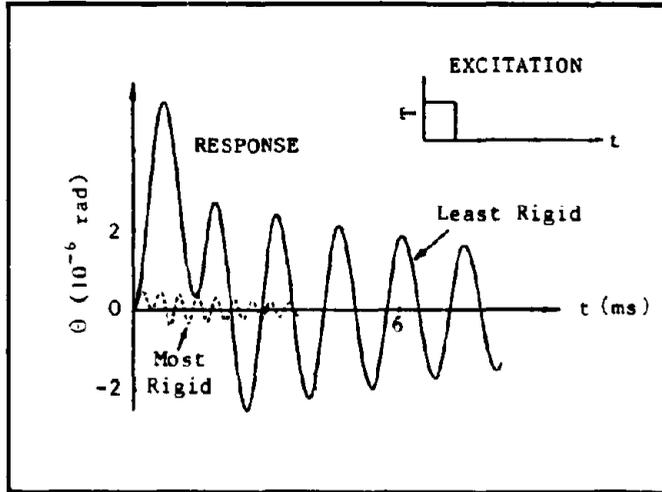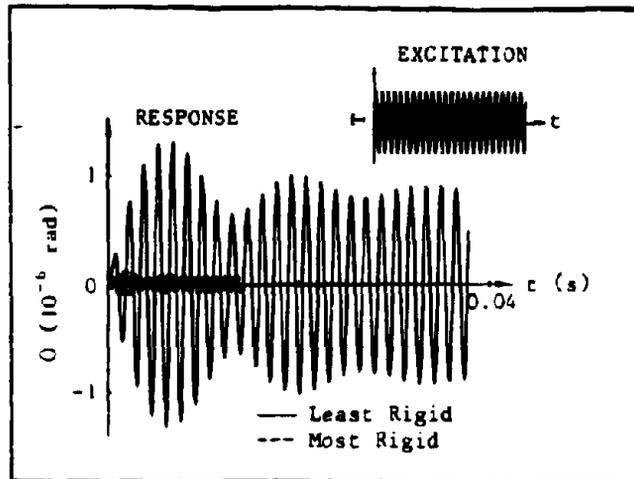
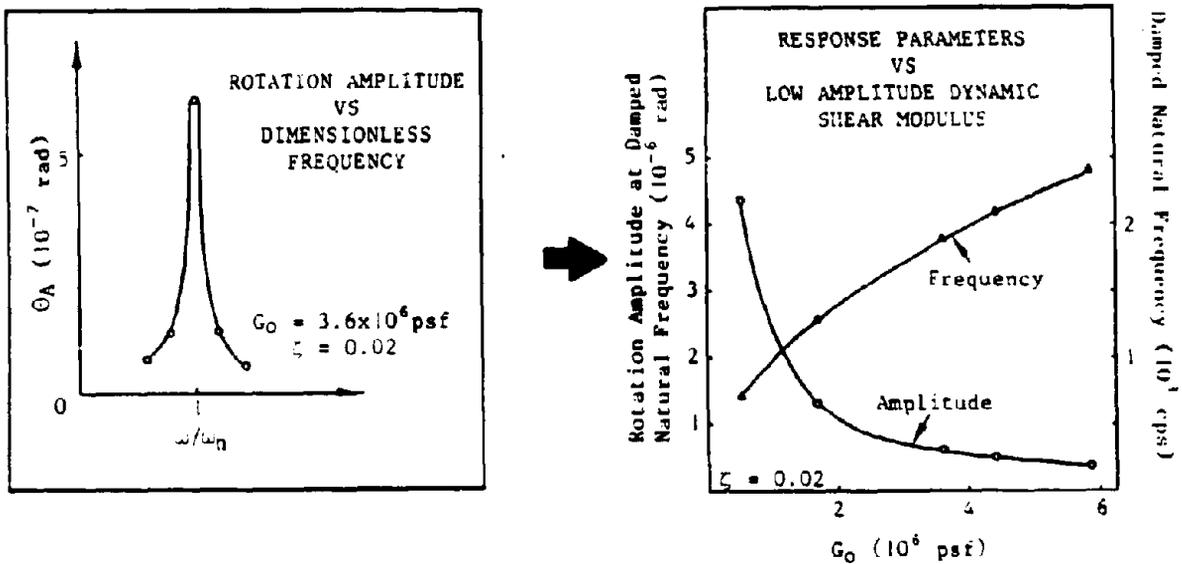(a) Samples of Predicted Measured Quantities
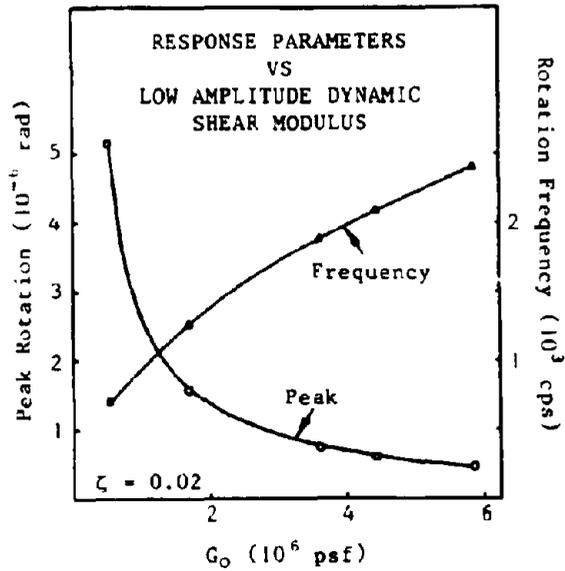
(b) Parameters of Predicted Measured Quantities

Figure 14: Results from Simulations of Low Amplitude Impulse Tests - Low Amplitude Dynamic Shear Modulus Study

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(a) Samples of Predicted Measured Quantities

(b) Parameters of Predicted Measured Quantities

Figure 15: Results from Simulations of Low Amplitude, High Frequency, Cyclic Tests - Low Amplitude Dynamic Shear Modulus Study

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the amplitude of the cyclic rotation of the inner cylinder at the damped natural frequency was predicted to be about 10 times larger when testing the least rigid sand than when testing the most rigid sand.

The results of this study are physically reasonable. One would expect, for a given low amplitude loading, the amplitude of the cyclic rotation of the inner cylinder at the damped natural frequency to be less and the damped natural frequency to be greater when testing more rigid sand. Thus, it was concluded that conducting high frequency, cyclic tests using the proposed testing procedure is a theoretically feasible means for determining the low amplitude dynamic shear modulus.

Based on our experiences in analytically simulating low amplitude impulse tests and low amplitude, high frequency, cyclic tests, we concluded that the impulse test is a preferable test for determining the low amplitude dynamic shear modulus. With the need to carry out a frequency sweep, the high frequency, cyclic test is somewhat more complex than the impulse test. Also, with the impulse test, it would be necessary to carry out simulations only for a very short duration to obtain the information required. In contrast, because of the time needed to reach steady state conditions, with the high frequency, cyclic test it would be necessary to carry out simulations for a relatively long duration.

Variation in the Dynamic Shear Modulus with Shear Strain--To determine the theoretical feasibility of the testing system for determining the variation in the dynamic shear modulus with shear strain, we varied the variation in dynamic shear modulus with shear strain without varying other parameters. The test soil was considered to have the same shear stress-strain behavior for low levels of strain but we varied shear stress-strain behavior for high levels of strain. To do this, we considered the test soil to have a single low amplitude dynamic shear modulus and shear stress-strain curve shape. However, we varied the shear strength over a range that could be realistically expected for the selected low amplitude dynamic shear modulus. Figure 16 presents the upper and lower bounds for the shear stress-strain skeleton curves used for our study.

For the study, we as   ed properties expected in sand deposits. However, results are expected to apply equally to silts and clays. The low amplitude dynamic shear modulus for each stress-strain curve was 2×10^6 psf. The Ramberg-Osgood parameters defining the shape of the curves were $\alpha = 1$, $C_1 = 0.8$, and $R = 5$. The upper bound shear strength was calculated to be 1500 psf while the lower bound was calculated to be 500 psf. The low amplitude dynamic shear modulus and the shear strengths were calculated assuming the following: depth = 6 ft, void ratio = 0.35, coefficient of earth pressure at rest = 1.0, effective unit weight of soil = 122 pcf, and effective angle of internal friction = 42 deg for the lower bound shear stress-strain curve, and depth = 77.5 ft, void ratio = 0.85, coefficient of earth pressure at rest = 0.6, effective unit weight of soil = 56 pcf, and effective angle of internal friction = 32 deg for the upper bound shear stress-strain curve. The equations used to calculate the low amplitude dynamic shear moduli and the shear strengths are presented in Refs. 15 and 16. The shear strengths were averages of the shear strengths estimated separately for horizontal and vertical planes. Intermediate values of strength were also considered. The selected Ramberg-Osgood shape parameters
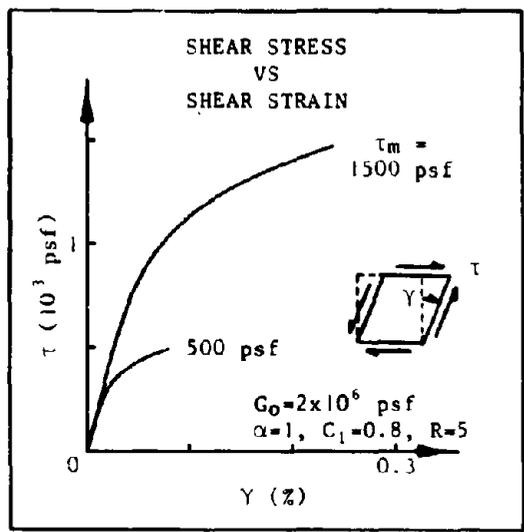
Figure 16: Bounds for Skeleton Shear Stress-Strain Curves Considered - Variation in Dynamic Shear Modulus with Shear Strain Study

give stress-strain curves that closely match experimental data for sands
(24). A value of viscous damping corresponding to 2% of critical damping was
assumed. This value is representative of sands (25).

We simulated both impulse tests and high frequency, cyclic tests. The
simulations of impulse tests are discussed first, and then, the simulations of
high frequency, cyclic tests are discussed.

For the simulations of the impulse tests, we considered an impulsive
loading having a rectangular shape. The response of the inner cylinder to
this loading was a decaying, cyclic rotation. To study the effects of
loading amplitude, we considered two different amplitudes for the torsional
impulsive loading: a medium amplitude of 2.7 ft-lb, and a relatively high
amplitude of 5.2 ft-lb. These amplitudes were intended to induce shear
strains on the order of 0.05% and 0.15%, respectively, in the test soil
having a medium strength of 1000 psf. The duration of all the loadings was
1.1 ms which, in each case, exceeded the time of occurrence of the first peak
of the rotation of the inner cylinder.

Results are presented in Fig. 17. Figure 17(a) gives samples of the
predicted measured quantities while Fig. 17(b) gives selected parameters of
the predicted measured quantities for the two amplitudes of loading. The
selected parameters include the peak rotation of the inner cylinder and the
time to this peak. The peak rotation of the inner cylinder, in all cases,
was the initial peak. The time to this peak was used as a measure of the
apparent natural period of the nonlinear system for the level of strain
induced; the greater the time to this peak, the greater was the apparent
natural period of the cyclic rotation of the inner cylinder. Results were
not plotted for simulations in which the strength of the test soil was
exceeded.

The results of the study indicate that, during impulse tests which
induce nonlinear shear stress-strain behavior in the test soil, the behavior
of the testing system will be sensitive to the variation in the dynamic shear
modulus with shear strain in a clear, physically reasonable manner. For
example, referring to Fig. 17(b), for the medium amplitude of loading, the
peak rotation and the time to this peak were predicted to be about 2 times
larger when testing the weakest sand than when testing the strongest sand.
Similar behavior was also predicted for the high amplitude of loading.
Additionally, an increase in the peak rotation of the inner cylinder and an
increase in the time to this peak were predicted, for each case considered,
as the amplitude of the loading was increased.

The results of this study are physically reasonable. For loadings which
induce nonlinear shear stress-strain behavior, one would expect the peak
rotation of the inner cylinder and the time to this peak to increase with
decreases in the shear rigidity of the test soil at high strains.
Additionally, one would expect the peak rotation of the inner cylinder and
the time to this peak to increase with increases in the amplitude of loading
when testing a given soil at strains for which the soil behavior is
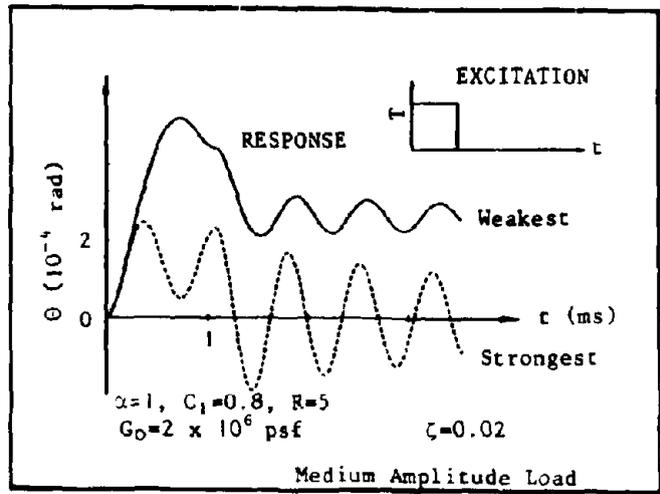
(a) Samples of Predicted Measured Quantities
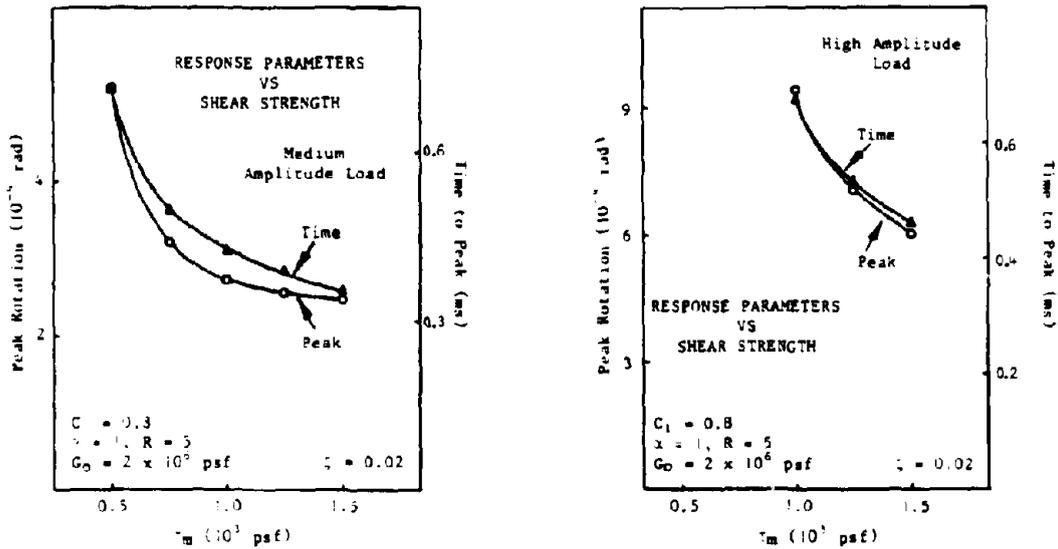
(b) Parameters of Predicted Measured Quantities

Figure 17: Results from Simulations of Impulse Tests - Variation in
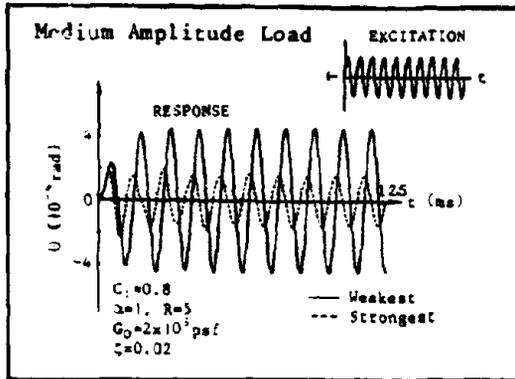Dynamic Shear Modulus with Shear Strain Study

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nonlinear. Thus, it was concluded that conducting impulse tests at higher amplitudes of loading using the proposed testing procedure is a theoretically feasible means for determining the variation in the dynamic shear modulus with shear strain.

For the simulations of high frequency, cyclic tests for determining the variation in the dynamic shear modulus with shear strain, we considered a sinusoidal torque excitation. The response of the inner cylinder to this loading was a cyclic rotation consisting of a decaying transient component and a steady state component. For a test soil of given properties, frequency response curves were developed in the vicinity of the apparent natural frequency to determine useful response parameters. The frequency response curves were developed by maintaining the amplitude of the loading while varying the frequency. The frequency response curves were based only on the steady state component of the response. Two amplitudes of loading were selected: a medium amplitude of 2.2 ft-lb and a relatively high amplitude of 4.8 ft-lb. These amplitudes were intended to induce shear strains on the order of 0.05% and 0.15%, respectively, in the test soil having a medium strength of 1000 psf, at the apparent natural frequency. These strains corresponded approximately to those induced in the simulations of impulse tests discussed previously in this subsection.

Results are presented in Fig. 18. Figure 18(a) gives samples of the predicted measured quantities while Fig. 18(b) gives selected parameters of the predicted measured quantities for the two different levels of loading. The selected parameters include the apparent natural frequency of the cyclic rotation of the inner cylinder and the amplitude of the cyclic rotation of the inner cylinder at the apparent natural frequency.

The results indicate that, during high frequency, cyclic tests which induce nonlinear shear stress-strain behavior in the test soil, some parameters describing the behavior of the testing system will l ⋅ sensitive to the variation in the dynamic shear modulus with shear strain, ι .le, under some conditions, other parameters may not. The behavior of the testing system was physically reasonable. For example, referring to Fig. 18(b), for the medium amplitude of loading, the apparent natural frequency of the cyclic rotation of the inner cylinder was predicted to be about 1.5 times greater while the amplitude of the cyclic rotation of the inner cylinder at the apparent natural frequency was predicted to be slightly larger when testing the strongest soil than when testing the weakest soil. A similar trend was predicted for the apparent natural frequency for the high amplitude of loading; however, the amplitude of the rotation of the inner cylinder was predicted to be about 1.2 times smaller when testing the strongest soil than when testing the weakest soil for which data was plotted. Additionally, a decrease in the apparent natural frequency of the cyclic rotation of the inner cylinder and an increase in the amplitude of the cyclic rotation of the inner cylinder at the apparent natural frequency were predicted as the amplitude of the loading was increased for any given set of soil properties.

The results are physically reasonable. For loadings which induce nonlinear shear stress-strain behavior, one would expect decreases in the apparent natural frequency of the cyclic rotation of the inner cylinder with

(a) Samples of Predicted Measured Quantities
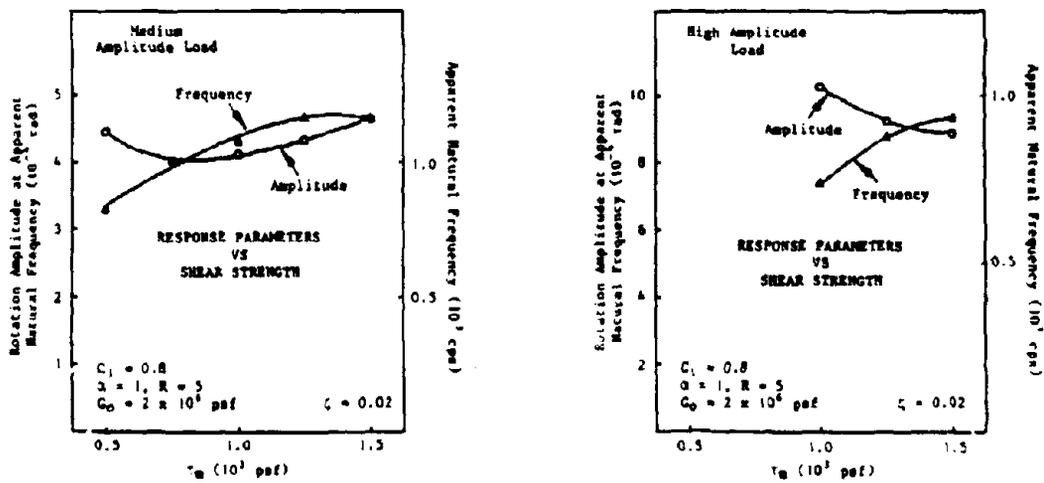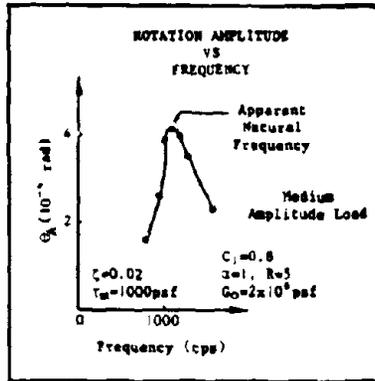
(b) Parameters of Predicted Measured Quantities

Figure 18: Results from Simulations of High Frequency, Cyclic Tests -
Variation in Dynamic Shear Modulus with Shear Strain Study

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decreases in shear strength while holding other parameters describing stress-strain behavior constant. The decrease in shear stiffness and the increase in hysteretic damping accompanying the decrease in shear strength would both contribute to a reduction in the apparent natural frequency. The insensitivity predicted for the amplitude of the cyclic rotation of the inner cylinder at the apparent natural frequency when considering the loading having medium amplitude was the result of the compensating effects of changes in shear stiffness and corresponding changes in hysteretic damping. For sinusoidal torque loads at the apparent natural frequency, decreases in shear stiffness would be expected to result in increases in the amplitude of the cyclic rotation of the inner cylinder while the corresponding increases in hysteretic damping would be expected to result in compensating decreases in this amplitude. The relative magnitudes of these compensating effects would be expected to vary with strain. Thus, the relationship between the amplitude of the cyclic rotation of the inner cylinder and the variation in the dynamic shear modulus with shear strain is predicted to be complex and may not show obvious trends. Finally, one would expect the predicted decreases in the apparent natural frequency of the cyclic rotation of the inner cylinder and increases in the amplitude of the cyclic rotation of the inner cylinder at the apparent natural frequency with increases in the amplitude of the loading, assuming a given set of soil properties. Thus, because at least one parameter of the response of the testing system was predicted to be sensitive to the variation in the dynamic shear modulus with shear strain in a clear, physically reasonable manner, we concluded that conducting high frequency, cyclic tests using the proposed testing procedure is a theoretically feasible means for determining the variation in the dynamic shear modulus with shear strain.

Based on our experiences in simulating impulse tests and high frequency, cyclic tests which induce high levels of shear strain, we concluded that the impulse test would be a preferable test for determining the variation in the dynamic shear modulus with shear strain. With the need to carry out a frequency sweep, the high frequency, cyclic test is somewhat more complex than the impulse test.

Degradation Characteristics of Clays--To determine the theoretical feasibility of the proposed testing system for determining the degradation characteristics of clays, a range of soil conditions was established by varying the degradability of the test soil while maintaining constant the parameters describing the undegraded shear stress-strain behavior (behavior expected during first cycle of loading). Thus, elements of soil at a single depth in uniform, saturated deposits were considered. We did not vary depth since behavior for different depths was expected to be qualitatively similar.

The degradability of the soil was varied by assuming three different degradation parameter functions (20). These functions, which describe cyclic degradation characteristics, are shown in Fig. 19. The functions describe the characteristics of a highly degradable, a moderately degradable, and a nondegradable clay. The average degradation parameter curve presented by Idriss et al. in Fig. 12 of Ref. 20, was used to describe the degradation characteristics of the highly degradable soil while 0.5 times this function was used to describe the degradation characteristics of the moderately
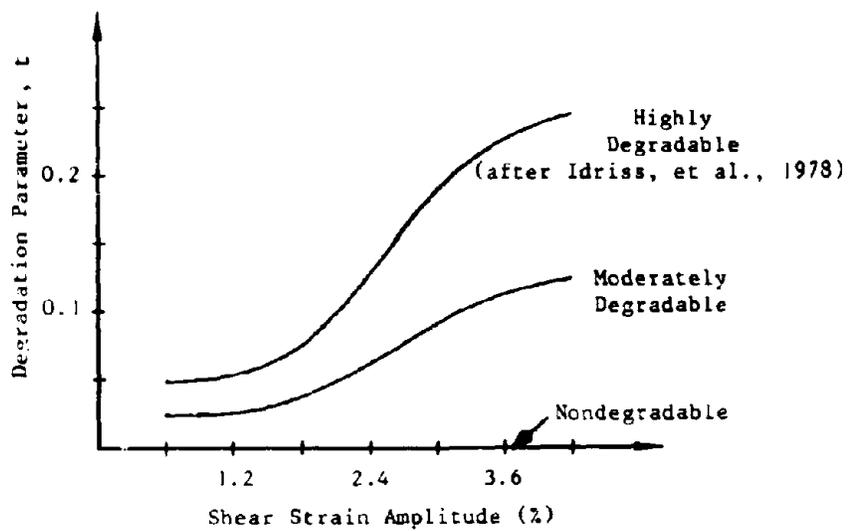
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Figure 19: Range of Clay Degradation Parameter Functions Used -
Degradation Characteristics of Clays Study

degradable soil. The base curve was converted to terms of shear strain from terms of normal strain (20) by assuming Poisson's ratio = 0.5 and linear elastic behavior. The degradation parameter function for the nondegradable clay was zero over the entire range of shear strain.

The low amplitude dynamic shear modulus and shear strength were calculated to be 720,000 psf and 660 psf, respectively. The depth, effective unit weight, void ratio, coefficient of earth pressure at rest, effective angle of internal friction, and cohesion were assumed to be 50 ft, 44 pcf, 1.3, 0.6, 28 deg, and 0 psf, respectively. The equations used to calculate the low amplitude dynamic shear modulus and the shear strength are presented in Ref. 15. The shear strength was the average of the shear strengths estimated separately for horizontal and vertical planes. Since solutions were obtained by the static procedure, we did not assume a value for viscous damping.

Guided by current laboratory practice for determining clay degradation characteristics (20), we considered a low frequency, cyclic excitation having a uniform rotation amplitude. The inner cylinder responded by developing the low frequency, cyclic torque needed to maintain the imposed rotation. Three amplitudes of rotation were considered: a high amplitude equal to 0.02 rad, a medium amplitude equal to 0.01 rad, and a low amplitude equal to 0.004 rad. These amplitudes, intended to induce various levels of degradation, were predicted to cause cyclic shear strains in the test soil along the wall of the inner cylinder having amplitudes of 3%, 1.8%, and 0.6%, respectively.

Results are presented in Fig. 20. Figure 20(a) gives samples of the predicted measured quantities. Figure 20(b) gives a selected parameter of the predicted measured quantities as a function of the number of cycles of loading for the various levels of degradability and the various amplitudes of excitation considered. The selected parameter was the amplitude of the cyclic torque developed by the inner cylinder, averaged over each cycle.

The results of the study indicate that the behavior of the testing system will be sensitive to the degradation characteristics of a clay in a clear, physically reasonable manner. For example, referring to Fig. 20(b), with the high rotation amplitude imposed on the inner cylinder, the amplitude of the cyclic torque developed by the inner cylinder was predicted to be about 50% less after 50 cycles of loading when testing the highly degradable clay than when testing the nondegradable clay, and about 30% less when testing the moderately degradable clay. Qualitatively similar behavior was predicted for the other amplitudes of loading. Also, as shown in Fig. 20(b), it was predicted that, when testing the highly degradable clay, the amplitude of the torque developed by the inner cylinder would decrease by about 50% in 50 cycles at the high amplitude of rotation, by about 25% at the medium amplitude, and by about 20% at the low amplitude. Qualitatively similar behavior was predicted for the clay having medium degradability while degradation was not predicted for the nondegradable clay.

The results of the study are physically reasonable. Because the test soil will be more susceptible to degradation when it is more degradable and subjected to larger loads, one would expect the amplitude of the cyclic torque developed by the inner cylinder to be less after a given number of
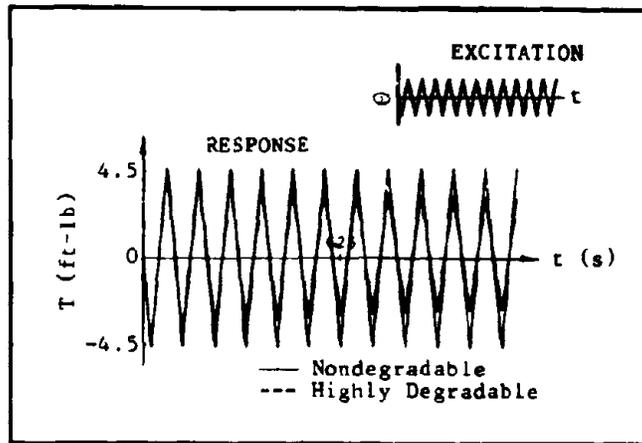
(a) Samples of Predicted Measured Quantities
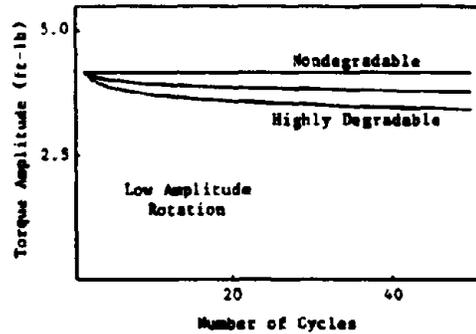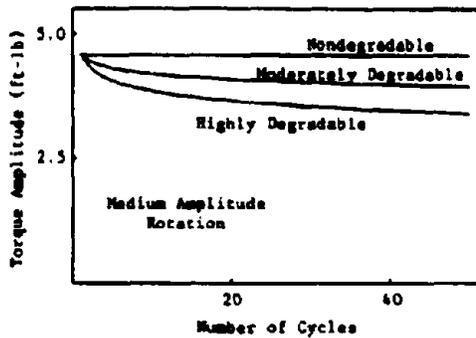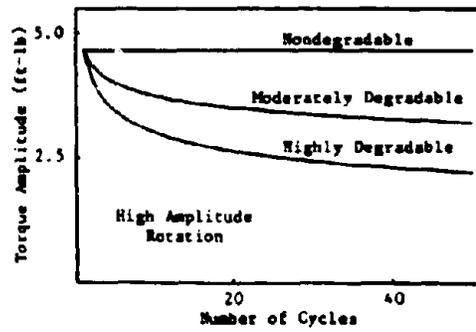
(b) Parameter of Predicted Measured Quantities

Figure 20: Results from Simulations of Low Frequency, Cyclic,
Controlled-Rotation Tests - Degradation Characteristics of Clays Study

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cycles of rotation when testing a more degradable clay at a given amplitude
of rotation and when testing a clay of given degradability at higher
amplitudes of rotation. Thus, it was concluded that the proposed testing
procedure is a theoretically feasible means for determining the degradation
characteristics of clays.

Degradation and Liquefaction Characteristics of Sands and Silts--To
determine the feasibility of the proposed testing system for obtaining the
degradation and liquefaction characteristics of sands and silts, we modeled
the degradation and liquefaction characteristics of the test soil
analytically and by the use of published test data (see Solution Procedures,
pg. 36). The published test data was used to more fully take into account
the restraining effect of dilation.

The range of soil conditions considered for the study based on the
analytical model was established by varying the degradation and liquefaction
characteristics of the test soil. Also, we varied the undegraded
shear stress-strain behavior (behavior expected during first cycle of
loading) to be consistent with the degradation and liquefaction
characteristics of the test soil. Elements of sand at a single depth
in uniform, saturated deposits were considered. We did not vary depth since
behavior for different depths was expected to be qualitatively
similar. Also, silts were expected to behave similarly to the sands.

The test soil was assumed to have high, medium, and low resistance to
degradation and liquefaction. The parameters defining the three models are
presented in Table 1. Values for the parameters defining the degradation and
liquefaction characteristics, C_1, C_2, C_3, C_4, k_2, m, and n, were obtained
from Ref. 23. The low amplitude dynamic shear moduli were calculated using
the equation presented in Ref. 15. Strengths were estimated iteratively such
that the sands having low, medium, and high resistance to degradation and
liquefaction showed approximately the same resistance to initial
liquefaction, under cyclic simple shear loading, as freshly deposited
Monterey No. 0 test sand (see Ref. 28) at relative densities of 54%, 75%, and
90%, respectively. For the strength estimates, we constructed a simple
model to simulate cyclic simple shear tests. We iteratively assumed shear
strengths until the behavior predicted with this model agreed reasonably well
with the data presented in Ref. 28. Figure 21 shows comparisons between the
behavior predicted by our simple shear model at the final stage of iteration
and that observed during the tests reported by Seed. Because of the
approximate nature of our study, we did not distinguish between initial
liquefaction and shear failure (see Analytical Modeling, pg. 32).

The range of soil conditions considered for the study based on the
published test data was also established by varying the degradation and
liquefaction characteristics of the test soil. Elements of sand at a
single depth in uniform, saturated deposits were considered. We did not vary
depth since behavior for different depths was expected to be
qualitatively similar. Silts were expected to behave similarly to the sands.
The degradation and liquefaction characteristics of the test soil
were varied by varying the relative density. Figure 11 presents the
published test data (see Solution Procedures, pg. 36). In this special case
of freshly deposited and identically prepared samples of sand tested in the
same pressure environment, relative density essentially determines
degradation and liquefaction characteristics. Broad ranges of degradation

Parameters	Low Resistance Sand	Medium Resistance Sand	High Resistance Sand
z (ft)	25	25	25
G_{mo} (psf)	1,550,000	1,700,000	1,810,000
τ_{mo} (psf)	800	1200	1400
$\bar{\gamma}$ (pcf)	63	66	68
e	0.66	0.59	0.54
D_r (%)	54	75	90
K_0	0.50	0.45	0.40
ζ	0.02	0.02	0.02
c_1[1]	0.666	0.52	0.346
c_2[1]	1.968	2.518	3.536
c_3[1]	4.761	4.569	3.707
c_4[1]	3.865	3.706	3.025
m[1]	0.43	0.43	0.43
n[1]	0.62	0.62	0.62
k_2[1]	0.0025	0.0025	0.0025

[1]After Martin et al., (1981)

Table 1: Parameters Defining Analytically-Based Models - Degradation and Liquefaction Characteristics of Sands and Silts Study
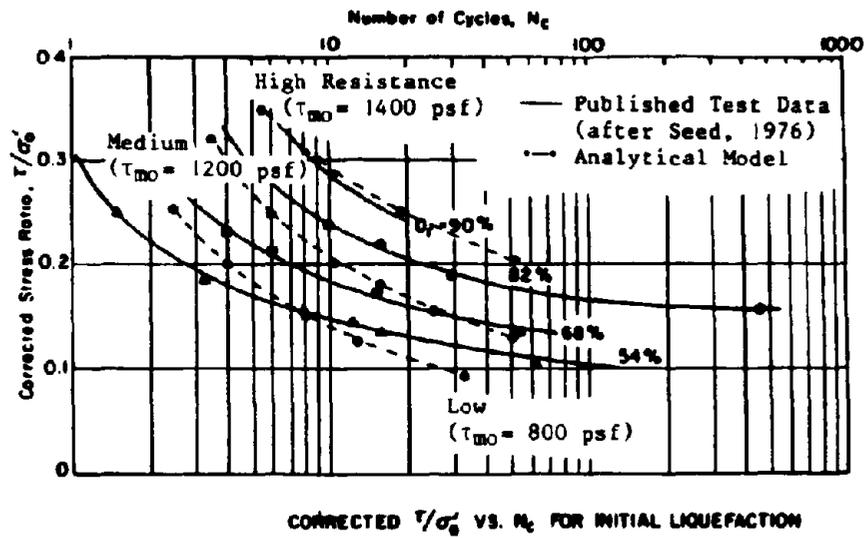
Number of Cycles, N_c

CORRECTED τ/σ_e' VS. N_c FOR INITIAL LIQUEFACTION

Figure 21: Resistance to Initial Liquefaction Under Cyclic Simple Shear Loading

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and liquefaction characteristics were assumed. The sand was considered to have relative densities ranging from 40% (very little resistance to liquefaction and degradation) to 100% (highly resistant to liquefaction and degradation). Consistent with the study based on the analytical model, we considered an element of soil at a depth of 25 ft. A single effective unit weight of 60 pcf was assumed.

Guided by current laboratory practice for determining the cyclic degradation and liquefaction characteristics of sands and silts, we considered a low frequency, cyclic, controlled-torque excitation having a uniform amplitude. The response of the inner cylinder was a low frequency, cyclic rotation. When using the analytically-based model, tests were simulated assuming amplitudes of applied torque which induced shear stress ratios ranging in value from 0.075 to 0.40. When using the model based on the published test data, we simulated tests assuming amplitudes of applied torque which induced shear stress ratios ranging in value from 0.15 to 0.40.

Figures 22 and 23 present results. Figure 22(a) presents samples of predicted measured quantities obtained using the analytically-based model. Figures 22(b) and (c) present a selected parameter of the predicted measured quantities as a function of the number of cycles of loading for the various soil conditions and amplitudes of loading (indicated in Fig. 22 as high, medium, and low stress ratios). The selected parameter was the amplitude of the rotation of the inner cylinder. Expected responses when taking dilation fully into account are sketched in Figs. 22(b) and (c). Figure 23 presents parameters obtained using the model based on the published test data. Figure 23(a) gives the estimated amplitude of the rotation of the inner cylinder after 10 cycles of loading as a function of the amplitude of the applied cyclic torque and relative density. Figure 23(b) gives the estimated limiting amplitude of the cyclic rotation of the inner cylinder as a function of relative density.

The results of the study involving the analytically-based model indicate that the behavior of the testing system will be sensitive to the degradation and liquefaction characteristics of a sand. Referring to Fig. 22, it was predicted that for a medium torque amplitude (medium stress ratio) it will take about 15 times as many cycles of loading for the inner cylinder to reach a rotation amplitude of 5×10^{-4} rad when testing the sand that is highly resistant to degradation and liquefaction than when testing the sand having low resistance. Also, it was predicted that when testing the sand having medium resistance, it will take about 15 times as many cycles for the inner cylinder to develop a rotation amplitude of 5×10^{-4} rad when applying a torque having a low amplitude of 1.4 ft-lb (low stress ratio) as when applying a torque having a high amplitude of 2.9 ft-lb (high stress ratio). Finally, it was predicted that the inner cylinder would develop unrestrained rotations in about 16 times as many cycles of loading when testing the sand having high resistance to liquefaction and degradation at the medium amplitude of loading (medium stress ratio) than when testing the sand having low resistance. Unrestrained rotation of the inner cylinder indicated either liquefaction of the test soil or shear failure.
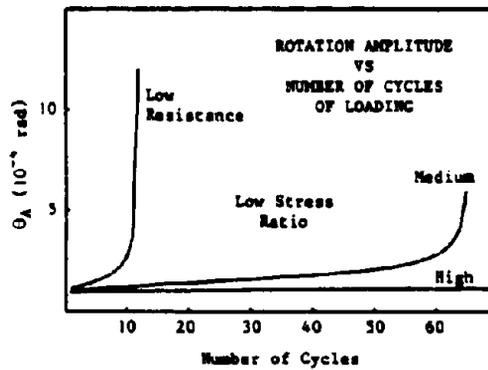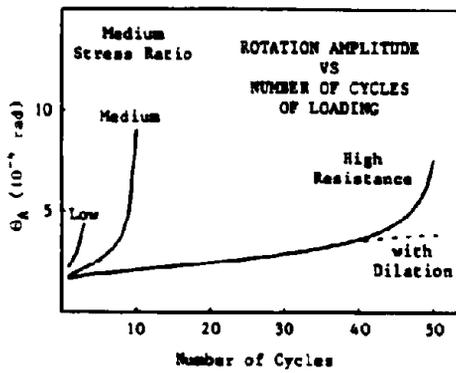
The results of the study are physically reasonable for loose sands and for dense sands at lower levels of excess porewater pressure. In these cases, the amplitude of the cyclic rotation of the inner cylinder would be expected to increase at a
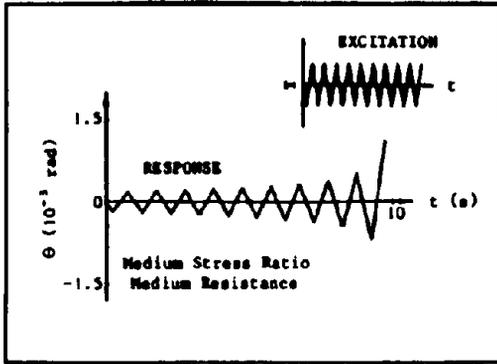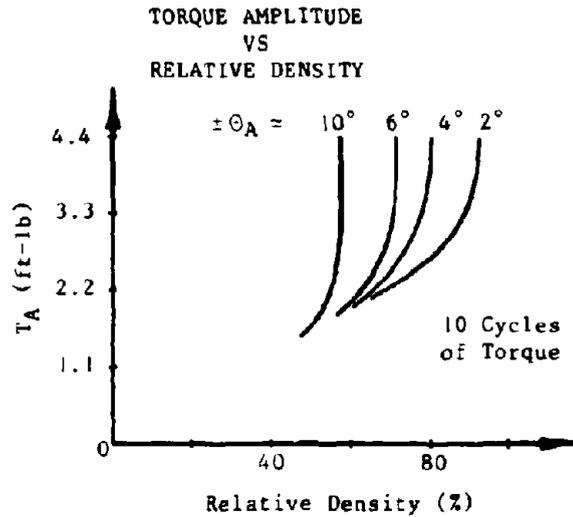
(a) Sample of Predicted Measured
Quantities
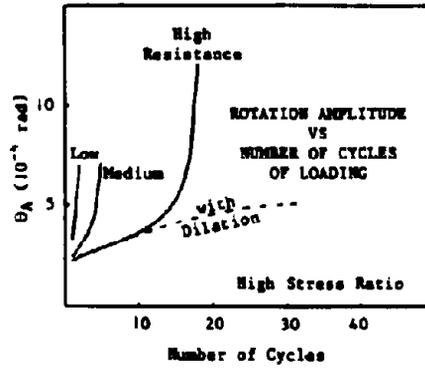
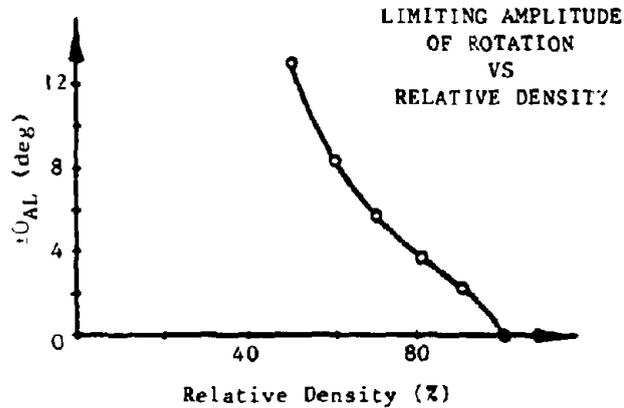(b) Parameter of Predicted Measured
Quantities

(c) Parameter of Predicted Measured Quantities

Figure 22: Results from Simulations of Low Frequency, Cyclic,
Controlled-Torque Tests Using Analytically-Based Model - Degradation and
Liquefaction Characteristics of Sands and Silts Study

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TORQUE AMPLITUDE
VS
RELATIVE DENSITY

$\pm \Theta_A \simeq$ 10° 6° 4° 2°

4.4

3.3

2.2

T_A (ft-lb)

1.1

10 Cycles
of Torque

0

40   80

Relative Density (%)

(a) Estimated Rotation Amplitudes of Inner Cylinder after Applying 10 Cycles of Torque When Testing Freshly Deposited Monterey No. 0 Sand

LIMITING AMPLITUDE
OF ROTATION
VS
RELATIVE DENSITY

12

8

$\pm \Theta_{AL}$ (deg)

4

0

40   80

Relative Density (%)

(b) Estimated Limiting Amplitude of Rotation of Inner Cylinder When Testing Freshly Deposited Monterey No. 0 Sand

Figure 23: Results from Simulations of Low Frequency, Cyclic, Controlled-Torque Tests Using Model Based on Published Test Data - Degradation and Liquefaction Characteristics of Sands and Silts Study

greater rate with an increase in the number of cycles of loading when testing sands less resistant to liquefaction and degradation and when applying higher amplitudes of torque to the inner cylinder. Additionally, when testing loose sands the inner cylinder would be expected to undergo unrestrained rotations if loaded sufficiently.

However, the results are not physically reasonable for dense sands at higher levels of excess porewater pressure. Because of the restraining effect of dilation, the inner cylinder would not be expected to undergo unrestrained rotation when testing a dense sand regardless of the level of excess porewater pressure. Rather, as sketched in Figs. 22(b) and (c), the inner cylinder would be expected to develop a limited amplitude of rotation. As explained in the section entitled **PROPOSED METHOD**, pg. 22, current, commonly used, state-of-the-art analytical models, such as the one we used, do not model dilation. As a result, we were unable to model effectively the behavior of dense sands at high levels of excess porewater pressure using analytical models. Thus, from the results obtained using the analytically-based model, we were only able to conclude that the proposed testing procedure is a theoretically feasible means for determining the degradation and liquefaction characteristics of looser sands and silts and the degradation characteristics of denser sands and silts at lower levels of excess porewater pressure.

The results of the study involving the published test data indicate that the behavior of the testing system will be sensitive to the degradation and liquefaction characteristics of all sands at higher levels of excess porewater pressure. For example, referring to Fig. 23(a), when applying a cyclic torque having an amplitude of 3.3 ft-lb, it was predicted that, after 10 cycles of loading, the inner cylinder will develop a rotation amplitude when testing a sand having low resistance to liquefaction and degradation (relative density = 55%) about 5 times larger than that developed when testing a highly resistant sand (relative density = 90%). Additionally, referring to Fig. 23(b), the limiting amplitude of the rotation of the inner cylinder was predicted to be about 4 times larger when testing the sand having low resistance to liquefaction and degradation than when testing the highly resistant sand. Also, as shown in Fig. 23(b), with sufficiently large amplitudes of applied torque, virtually unrestrained rotations of the inner cylinder were predicted when testing the sand at relative densities less than about 45%. This indicated liquefaction of the test soil.

These results are physically reasonable. One would expect both the amplitude of the cyclic rotation of the inner cylinder after an arbitrary number of cycles of loading and the limiting amplitude of this rotation to be less when testing sands with greater resistance to liquefaction and degradation. Also, one would expect liquefaction after a sufficient number of cycles of loading at an appropriate amplitude when testing sands having little resistance to liquefaction. Thus, it was concluded that the proposed testing system is a theoretically feasible means for determining the degradation and liquefaction characteristics of sands and silts.

Figures 24 and 25 are presented for demonstrative purposes. These figures show plots of other information that was computed, using the analytically-based model. This type of information could be derived from measured data by soil-probe interaction analysis. Figure 24(a) gives the buildup in the excess porewater pressure within the test soil as a function of the number of cycles of loading at a high shear stress ratio, while Fig.
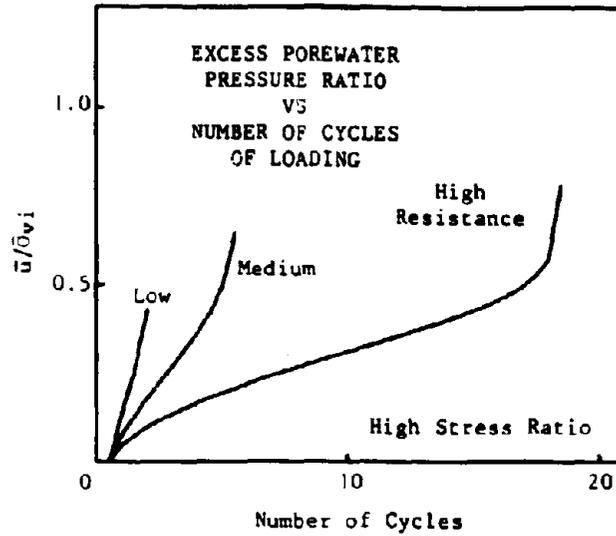
(a) Predicted Derived Information
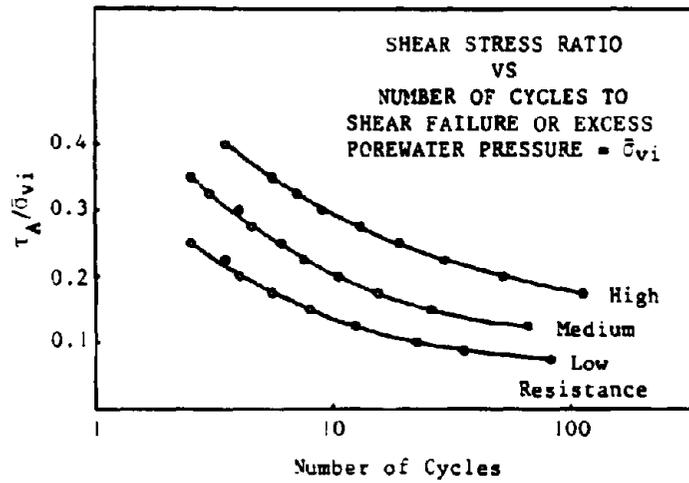
(b) Predicted Derived Information

Figure 24: Results from Simulations of Low Frequency, Cyclic, Controlled-Torque Tests Using Analytically-Based Model - Degradation and Liquefaction Characteristics of Sands and Silts Study
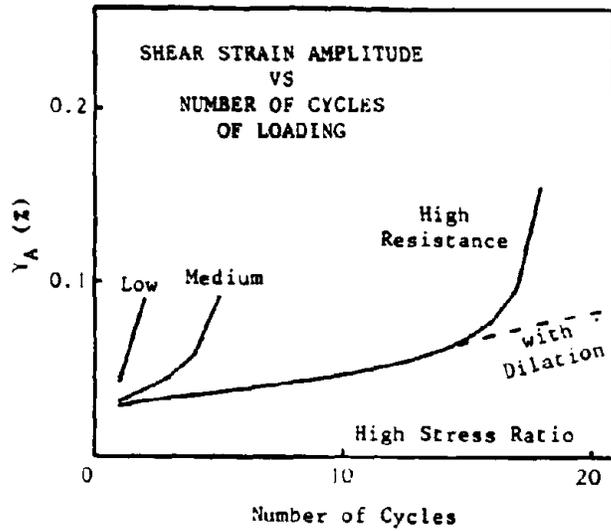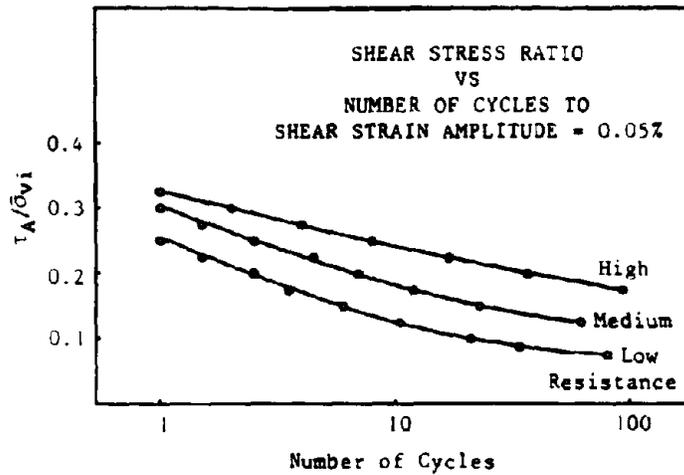
60

(a) Predicted Derived Information

(b) Predicted Derived Information

Figure 25: Results from Simulations of Low Frequency, Cyclic,
Controlled-Torque Tests Using Analytically-Based Model - Degradation and
Liquefaction Characteristics of Sands and Silts Study

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25(a) similarly gives the amplitude of the cyclic shear strain within the
test soil. Such information may be used to develop typical presentations of
degradation and liquefaction characteristics such as those shown in Figs.
24(b) and 25(b). Figure 24(b) presents the shear stress ratio as a function of
the number of cycles of loading required to cause the excess porewater pressure
to rise to the level of the initial effective vertical stress (liquefaction
in looser sands) or to cause shear failures for the various soil conditions.
Figure 25(b) presents similar information in terms of a
specified level of the amplitude of the cyclic shear strain rather than a
specified level of excess porewater pressure.

Conclusions

       In this subsection, we itemize the main conclusions resulting from the
theoretical feasibility study. These conclusions directly relate to the
objectives presented in the subsection entitled **Objectives**, pg. 28.

       1) The proposed testing system is a theoretically feasible means for
determining the low amplitude dynamic shear modulus, the variation in the
dynamic shear modulus with shear strain, the degradation characteristics of
clays, and the degradation and liquefaction characteristics of sands and
silts.

       2) The theoretical feasibility study served as a basis for the
preliminary design criteria needed to conduct an operational feasibility
study for determining the operational feasibility of a laboratory research
prototype testing system and to develop a preliminary design for such a system.

       3) Based on the results of the theoretical feasibility study, an
initial testing methodology was established for Phase II research involving
the laboratory testing of a research prototype testing system. The impulse
test was selected as the preferable test for determining dynamic shear
moduli. Using conventional practice as a guide, the low frequency, cyclic,
controlled-rotation test was selected as the initial testing mode for
determining the degradation characteristics of clays, while the low
frequency, cyclic, controlled-torque test was selected as the initial testing
mode for determining the degradation and liquefaction characteristics of
sands and silts.

       4) The analytical procedures, developed as part of the theoretical
feasibility study, will serve as simple, but effective means
for qualitatively checking, during Phase II research, the results from
laboratory tests involving a research prototype testing system, and
developmental results from descriptive soil-probe interaction analysis
procedures, to be constructed as part of Phase II research.

       5) We found two unexpected results in the theoretical feasibility study.
The peak rotation of the inner cylinder during long duration impulse tests for
determining the low amplitude dynamic shear modulus was predicted to be
significantly more sensitive to the low amplitude dynamic shear modulus than
the shear wave velocity. Laboratory testing would be required to confirm
this finding. Also, the results of our study indicate that the impulse test
may be preferable to the high frequency, cyclic test for determining dynamic
shear moduli.

Supplementary Theoretical Feasibility Study

Introduction and Summary

We conducted a cursory supplementary theoretical feasibility study in addition to our proposed theoretical feasibility study. The purpose of this supplementary study was to estimate roughly the rotational flexibility of the inner cylinder, as configured in the preliminary design of the testing system, relative to the rotational flexibility of the test soil. The estimate was made to determine whether special attention would be required as a result of such relative flexibility. For our theoretical feasibility study, we assumed rigid behavior for the inner cylinder. Rigid behavior is desired mainly so that the measured rotational motions will be due entirely to the rotation of the test soil. Because of project limits, we estimated the rotational flexibility of the inner cylinder relative to that of the test soil only during low frequency, cyclic, controlled-torque tests for determining the degradation and liquefaction characteristics of sands and silts. This mode of testing is expected to be the main capability of the proposed testing system.

The results of our cursory study indicate that the inner cylinder, as configured in the preliminary design of the testing system, may be rotationally flexible relative to the test soil during the initial cycles of loading. However, the inner cylinder was predicted to be rotationally rigid in comparison to the degraded test soil during the later cycles of loading.

Based on the results of the study, we concluded that the estimated rotational flexibility of the inner cylinder would increase uncertainty in inferring soil properties from test results. Nevertheless, because of the many promising mitigating steps which could be taken, if necessary, to reduce the uncertainty caused by flexibility of the inner cylinder, we feel that the proposed testing system will serve its stated purpose well and will advance our ability to determine in situ cyclic and dynamic soil properties.

We provide details in the remaining subsections. We state the objective and procedures of the supplementary theoretical feasibility study and present and discuss results, mitigating steps, and conclusions.

Objective

The objective of this supplementary study was to provide a rough estimate of the rotational flexibility of the inner cylinder, as configured in the preliminary design of the testing system (see report from Sweet & Aiken, Inc., in Appendix C), relative to the rotational flexibility of the test soil. The estimate was made to help determine whether any special attention should be directed toward the issue of the rotational flexibility of the inner cylinder. Rigid behavior is desired, mainly, so that the measured rotational motions will be entirely due to the rotation of the test soil. The study was conducted considering only the low frequency, cyclic, controlled-torque test for determining the degradation and liquefaction characteristics of sands and silts. This testing mode is expected to be the main capability of the proposed testing system.

Procedures

In this subsection, we discuss the procedures used to estimate the rotational flexibility of the inner cylinder relative to that of the test soil during low frequency, cyclic, controlled-torque tests. This relative flexibility was estimated for the initial cycle of loading, before significant degradation of the test soil is expected to have taken place. Estimates were also made for the ultimate cycle of loading during which the test soil is expected to have either liquefied or developed limited cyclic deformations. However, for the ultimate cycle of loading we estimated the rotational stiffness of the inner cylinder relative to that of the test soil. Estimates were made assuming expected ranges of soil conditions and applied torque. In our estimates, we did not consider the rotational flexibility of the inactive portion of the inner cylinder. Since this portion will not interact with the test soil, the effects of the rotational flexibility of this portion could be taken into account quite accurately.

The rotational flexibility of the inner cylinder relative to that of the test soil during the initial cycle of loading, F_i, was estimated using the following relationship:

$$F_i = \cfrac{Tl / \left[\frac{\pi}{32}(D_o^4 - D_i^4)\, G_i \right]}{-\left(\frac{r_o - r_i}{r_o}\right) \left[\cfrac{T/2\pi r_i^2 l}{(G_{mo}/\tau_{mo})\,(-T/2\pi r_i^2 l) - G_{mo}} \right]} \tag{11}$$

where

r_i = outer radius of inner cylinder
r_o = inner radius of outer cylinder
D_o = outer diameter of inner cylinder
D_i = inner diameter of inner cylinder
G_i = shear modulus of inner cylinder
l = active length of inner cylinder
G_{mo} = undegraded low amplitude dynamic shear modulus of test soil
τ_{mo} = undegraded shear strength of test soil
T = applied torque

Equation 11 gives the ratio of the static twist of the active portion of the flexible inner cylinder alone, caused by a static torque acting throughout the cylinder, to the static rotation of a rigid inner cylinder embedded in the undegraded test soil, caused by the same torque. Thus, the flexible inner cylinder and the test soil were considered independently. Also, rotations estimated for the rigid inner cylinder were caused entirely by the rotation of the test soil. A derivation for Eq. 11 is presented in Appendix B, pg. B-4.

The following equation was used to estimate the rotational stiffness of the inner cylinder relative to the rotational stiffness of the test soil during the ultimate cycle of loading, S_u:

$$S_u = -\frac{(\frac{r_o-r_i}{r_o})\,[\gamma_{ruA}(r_i)]}{T1/[\frac{\pi}{32}\,(D_o^4-D_i^4)\,G_i]} \tag{12}$$

where $\gamma_{ruA}(r_i)$ = amplitude of cyclic shear strain developed in test soil along wall of inner cylinder during ultimate cycle of loading

Equation 12 gives the ratio of the amplitude of the low frequency, cyclic rotation of the inner cylinder caused by a cyclic torque assuming the inner cylinder to be rigid and embedded in fully degraded test soil, to the static twist of the flexible inner cylinder alone, caused by a torque equal in value to the amplitude of the cyclic torque and acting throughout the cylinder. Thus, the flexible inner cylinder and the test soil were considered independently. Also, rotations estimated for the rigid inner cylinder were caused entirely by the rotation of the test soil. A derivation for Eq. 12 is presented in Appendix B, pg. B-6.

To apply Eq. 12, the amplitude of the cyclic shear strain developed in the test soil along the wall of the inner cylinder during the ultimate cycle of loading, $\gamma_{ruA}(r_i)$, was estimated using the published test data presented by Seed (28) and shown in Fig. 11(a). This figure gives, as a function of relative density, the limiting amplitudes of shear strain that developed in samples of sand in response to cyclic shear stresses applied to the samples. The data and the use of the data are discussed more fully in the subsection entitled Solution Procedures, pg. 36. In using Eq. 12, we assumed that the limiting strains were independent of the applied stresses (28). Thus, the applied torque did not affect our predictions of the rotations developed by the test soil during the ultimate cycle of loading.

Presentation and Discussion of Results

In this subsection, we present and discuss the results of our study of the rotational flexibility of the inner cylinder relative to that of the test soil. To estimate both the rotational flexibility and stiffness of the inner cylinder relative to the test soil, we assumed a steel inner cylinder having an active length of 8 in, an outer diameter of 1 in, and a wall thickness of 1/16 in. These dimensions were based roughly on the configuration of the inner cylinder in the preliminary design of the testing system (see report from Sweet & Aiken, Inc., Appendix C).

The soil properties considered correspond to those expected in a loose, uniform, saturated sand deposit having low resistance to degradation and liquefaction and those expected in a dense, uniform, saturated sand deposit having high resistance to degradation and liquefaction. The following conditions were assumed for the loose and dense deposits, respectively: coefficient of earth pressure at rest = 0.6 and 0.4, effective angle of internal friction = 32 deg and 41 deg, void ratio = 0.85 and 0.54, and the effective unit weight = 56 and 68 pcf. Properties were obtained for depths of about 5 and 75 ft. For these depths, the strengths of the loose deposit

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were estimated to be 100 and 1450 psf, and the strengths of the dense deposit were estimated to be 200 and 3600 psf, respectively. Also, for these depths the low amplitude dynamic shear moduli were estimated to be 514,000 and 1,990,000 psf for the loose deposit and 723,000 and 3,100,000 psf for the dense deposit, respectively. The shear strengths for the loose deposit and shear moduli for both deposits were calculated using expressions presented in Refs. 15 and 16. These shear strengths were averages of the shear strengths estimated separately for horizontal and vertical planes. The strengths for the dense deposit were estimated by calibrating a simple shear model to test data presented by Seed (28) (see Degradation and Liquefaction Characteristics of Sands and Silts, pg. 53). Torques representing various levels of applied shear stress ratio, $\tau_{rA}(r_i)/\bar{\sigma}_{vi}$, were considered. An amplitude of cyclic shear strain equal to 40% was taken to represent liquefaction (see Fig. 11(a)).

Summaries of results are provided in Table 2. Table 2(a) provides the rotational flexibility of the active portion of the inner cylinder, relative to that of the test soil, during the initial cycle of loading for the various soil and test conditions considered. Table 2(b) provides the rotational stiffness of the active portion of the inner cylinder, relative to that of the test soil, during the ultimate cycle of loading for the various soil and test conditions considered.

The results of the cursory study indicate that during the initial cycles of loading during low frequency, cyclic, controlled-torque tests for determining the degradation and liquefaction characteristics of sands and silts, the inner cylinder of the testing system may be rotationally flexible relative to the test soil. This relative flexibility may be a source of uncertainty in inferring soil properties from test results. This relative flexibility, and the uncertainty it may introduce, is predicted to be greatest when testing stiff, dense sand deposits at great depth using low levels of excitation. The rotational flexibility of the inner cylinder was predicted to decrease relative to the test soil with higher levels of excitation because higher torques caused decreases in the stiffness of the nonlinear test soil but not of the linear inner cylinder.

The results also indicate that, under all conditions, the inner cylinder will be rotationally very rigid in comparison to the test soil during the ultimate cycle of loading when the test soil has fully degraded. Thus, the rotational flexibility of the inner cylinder relative to that of the test soil should not be a large source of uncertainty in inferring soil properties from the later, more important, stages of a cyclic test.

Mitigating Steps

In this subsection, we discuss some of the steps which could be taken, if necessary, to decrease uncertainty introduced into test results by rotational flexibility of the inner cylinder. Generally, such uncertainty may be reduced by reducing the rotational flexibility of the inner cylinder relative to that of the test soil and by taking into account the rotational flexibility of the inner cylinder in the interpretation of test results.

There are a number of ways to greatly reduce the rotational flexibility of the inner cylinder relative to that of the test soil. The most promising

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		Depth (ft)	
		5	75
Shear Stress Ratio	0.05	1.0	3.5
	0.15	0.7	2.3

Loose Sand

		Depth (ft)	
		~5	75
Shear Stress Ratio	0.2	1.2	4.5
	0.4	0.7	3.0

Dense Sand

(a) Relative Rotational Flexibility, F_i, - Initial Cycle of Loading

		Depth (ft)	
		5	75
Shear Stress Ratio	0.05	13,000	900
	0.15	4,000	300

Loose Sand

		Depth (ft)	
		~5	75
Shear Stress Ratio	0.2	500	27
	0.4	250	14

Dense Sand

(b) Relative Rotational Stiffness, S_u, - Ultimate Cycle of Loading

Table 2: Rotational Flexibility and Stiffness of Active Portion of Inner Cylinder Relative to Rotational Flexibility and Stiffness of Test Soil

methods appear to be reducing the length of the inner cylinder or appropriately increasing the diameters and wall thicknesses of the inner and outer cylinders. Consistent with other in situ testing methods, testing may be optimized by the use of differently sized systems for different conditions. Additionally, for greater certainty in results from earlier stages of loading, the rotational flexibility of the inner cylinder could be reduced relative to that of the test soil by conducting tests at higher levels of excitation.

The most promising means of taking the rotational flexibility of the inner cylinder into account when interpreting soil properties from test results is directly modeling the rotational flexibility of this cylinder in the soil-probe interaction analysis. This is expected to be effective because it is not expected to be difficult to describe, in reasonably complete and accurate detail, the flexibility of the inner cylinder and the interaction of the flexible inner cylinder with the test soil. The inner cylinder will have a relatively simple geometry and relatively simple material properties, and the predominate stresses developed in the inner cylinder and the test soil during testing are expected to be simple torsional shear stresses. To additionally account for the rotational flexibility of the inner cylinder, greater weight could be placed on data obtained during the later stages of a cyclic test.

Conclusions

In this subsection, we present the conclusions arising from the study of the rotational flexibility of the inner cylinder relative to that of the test soil. The conclusions apply only to low frequency, cyclic, controlled-torque tests for determining the degradation and liquefaction characteristics of sands and silts.

1) The inner cylinder of the proposed testing system, as configured in the preliminary design, may be rotationally flexible relative to the test soil during the initial cycles of loading. This relative rotational flexibility is expected to be greater when testing denser soils at greater depths and when applying lower levels of excitation. This relative flexibility will be a source of uncertainty when inferring soil properties from test results.

2) The inner cylinder was predicted to be rotationally very rigid in comparison to the test soil, under all conditions considered, during the ultimate cycle of loading when the test soil has fully degraded. Thus, a relatively high degree of certainty is expected when inferring soil properties from test results obtained during the later stages of loading.

3) Uncertainty in the interpretation of test results, caused by the rotational flexibility of the inner cylinder relative to that of the test soil, may be reduced considerably by reducing this relative rotational flexibility. The most promising methods for reducing relative flexibility include decreasing the length of the inner cylinder or appropriately increasing the diameters and thicknesses of the inner and outer cylinders. Consistent with other in situ testing procedures, testing may be optimized by use of differently sized systems for different conditions. Additionally, to reduce relative flexibility, tests may be carried out at higher levels of excitation.

4) Uncertainty in the interpretation of test results, caused by the rotational flexibility of the inner cylinder relative to that of the test soil, may also be reduced considerably by accounting for the rotational flexibility of the inner cylinder in the interpretation of test results. The most promising method for doing this is to directly model the rotational flexibility of the inner cylinder in the soil-probe interaction analysis used to infer soil properties. Also, greater weight could be placed on data obtained during the later stages of a test.

5) Prior to developing a final design for a laboratory research prototype testing system, modest analytical studies should be conducted to optimize the dimensions of the probe.

6) Because of the many promising mitigating steps which could be taken, if necessary, to reduce the uncertainty in test results caused by the rotational flexibility of the inner cylinder, the proposed testing system is expected to serve its stated purpose well and to advance our ability to determine in situ cyclic and dynamic soil properties.

Operational Feasibility Study

Introduction and Summary

An operational feasibility study was conducted to determine the operational feasibility of a laboratory research prototype testing system. This system will be functionally similar to the field system scheduled for development during Phase III. Since our firm did not have the necessary expertise, we arranged for the mechanical engineering design firm of Sweet & Aiken, Inc., (Sweet & Aiken) to conduct the study. Studies were carried out to determine the operational feasibility of a laboratory testing system for determining 1) the low amplitude dynamic shear modulus and the variation in the dynamic shear modulus with shear strain by impulse test, 2) the cyclic degradation characteristics of clays by low frequency, cyclic, controlled-rotation test, and 3) the cyclic degradation and liquefaction characteristics of sands and silts by low frequency, cyclic, controlled-torque test. Based on the results of our theoretical feasibility study, these three testing modes were judged to be the most promising modes of testing.

Results from the operational feasibility study indicate that a laboratory testing system is operationally feasible. The main components of the system, which satisfy reasonably well specified design criteria, were found to be either available or readily producible and it was determined that these components could be assembled into a convenient, workable arrangement. The design criteria, developed by our firm as part of our theoretical feasibility study, specified mechanical, excitation, and measurement requirements. The operational feasibility study resulted in a preliminary design for a laboratory research prototype testing system.

Not all specified criteria were satisfied; however, the resulting

consequences are not expected to be severe. For example, Sweet & Aiken was unable to find a transducer capable of measuring accurately the smallest predicted angles of the rotation of the inner cylinder that might occur during low frequency, cyclic, controlled-torque tests for determining degradation and liquefaction characteristics of sands and silts. However, it appears that we will be able to measure reasonably small angles. Thus, the potential increase in uncertainty in inferring soil properties from test results was judged to be only modest.

Sweet & Aiken estimated a cost of $78,260 for the detailed design and construction of a laboratory research prototype testing system. This is a conservative estimate. Based on our experiences, we feel the estimated cost is consistent with that of the design and construction of comparable testing systems.

Details are provided in the remaining subsections. We state the objectives and procedures of the operational feasibility study, define the main components of the laboratory research prototype testing system, present and discuss our specified design criteria, present and discuss results from the operational feasibility study conducted by Sweet & Aiken, and present conclusions. In discussing the operational feasibility study, we concentrate on the geotechnical aspects of the testing system. The report from Sweet & Aiken, which summarizes the firm's work and is included in Appendix C, concentrates on the operational aspects of the testing system. Thus, the two reports are intended to complement each other.

Objectives

The main objective of the operational feasibility study was to determine the operational feasibility of a laboratory research prototype testing system for determining 1) the low amplitude dynamic shear modulus and the variation in the dynamic shear modulus with shear strain by impulse test, 2) the cyclic degradation characteristics of clays by low frequency, cyclic, controlled-rotation test, and 3) the cyclic degradation and liquefaction characteristics of sands and silts by low frequency, cyclic, controlled-torque test. These three testing modes were judged to be the most promising modes of testing based on the results of the theoretical feasibility study. The laboratory research prototype testing system will be functionally similar to, but somewhat simpler than, the field system scheduled for development during Phase III. If successful, the operational feasibility study was to result in a preliminary design for a laboratory research prototype testing system. A secondary objective of the operational feasibility study was to develop an accurate cost estimate for the detailed design and construction of the laboratory research prototype testing system.

Procedures

The operational feasibility of the laboratory research prototype testing system was determined by determining whether the main components (see Main Components, pg. 71), which satisfy specified design criteria, were available, and whether these components could be assembled into a convenient, workable arrangement. If it was found that the required components were available and could be assembled into a convenient, workable arrangement, then, we concluded that the laboratory research prototype testing system is operationally feasible.

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First, our firm specified design criteria for the testing system including mechanical, excitation, and measurement requirements. Then, Sweet & Aiken conducted the study to determine the operational feasibility of the laboratory system for determining 1) the low amplitude dynamic shear modulus and the variation in the dynamic shear modulus with shear strain by impulse test, 2) the cyclic degradation characteristics of clays by low frequency, cyclic, controlled-rotation test, and 3) the cyclic degradation and liquefaction characteristics of sands and silts by low frequency, cyclic, controlled-torque test.

The following main tasks were performed as part of the operational feasibility study: 1) the evaluation and selection of the excitation systems, 2) the evaluation and selection of the measurement systems, 3) the design of the configuration of the probe (including sizing of main components and selection of seals, bearings, and materials), 4) the design of the vertical pressure system, 5) the development of a preliminary design layout, 6) the optimization of cylinder wall friction, 7) the evaluation and selection of the power and support cables, 8) the identification of problem areas, and 9) the development of an accurate cost estimate for the detailed des.gn and construction of a laboratory system.

Main Components

The main components of the laboratory research prototype testing system considered in the operational feasibility study were those components which were judged essential to the performance of the stated functions of the testing system and were not obviously feasible. These components included mechanical, excitation, and measurement components and are expected to resemble closely the corresponding components of a field unit. The main mechanical components considered were the inner cylinder, the outer cylinder, the shield, the vertical pressure system, the sealing systems, and the bearing systems. The main excitation components considered included the power, control, and drive components required to provide the selected excitations. The main measurement components considered included the sensing, transmission, display, and recording components required to measure, transmit, display, and record appropriate excitation and response information. Components, which were not strongly addressed in the operational feasibility study, included a laboratory penetration system and an air pressure system for the probe. These items were considered feasible.

Design Criteria

In this subsection, we present the design criteria specified for the design or the selection of the main components of the laboratory research prototype te ting system. We specified mechanical, excitation, and measurement requirements. The specified criteria, which are preliminary criteria, are summarized in Table 3. In the following subsections, we present the mechanica' and the excitation and measurement requirements.

Mechanical Requirements--The mechanical requirements cover those specifications which affect the design or selection of the mechanical systems, including the vertical pressure system. The following items were specified: the sizes and geometries for the inner and outer cylinders, the

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Mechanical Requirements

	Dimensions					Penetration Forces			Maximum Differential Lateral Pressure (psf)
	Outer Diameter (in)	Wall Thickness (in)	Length Below Piston (in)	Active Length (in)	Shielded Length (in)	Bearing (lb)	Skin Friction (lb)	Total (lb)	
Outer Cylinder	3	1/12	12	N/A	N/A	4500	9500	14,000	30,000
Inner Cylinder	1	1/16	12	8	4	1500	2500	4000	30,000

Maximum Environmental Fluid Pressure: 100,000 psf

Range of Differential Vertical
Pressures to Be Applied and Measured
by Vertical Pressure System: 0 - 6000 psf

Excitation and Measurement Requirements

Mode	Excitation			Torque Amplitude Range (ft-lb)	Rotation Amplitude Range (deg) pk to pk single pk		Rotation Frequency (cps)	Acceleration Amplitude Range @ 1 inch radius (g)	Maximum Power (hp)
	Form	Period (sec)	Duration						
Controlled-Torque Cyclic Test	sine wave	0.1 - 50	≤100 cycles	0.1 - 15	0.002-30	0.001-15	0.02-10	N/A	0.20
Controlled-Rotation Cyclic Test	sine wave	0.1 - 50	≤100 cycles	0.1 - 15	0.4-3.0	0.2-1.5	0.02-10	N/A	0.03
Impulse Test	rectangular impulse	N/A	≥0.0002 s	0.01 - 20	N/A		400-2400	±0.3 - ±210	0.20

Table 3: Preliminary Design Criteria

maximum penetration forces, the desired wall characteristics of the inner and outer cylinders, the maximum differential lateral pressures acting on the inner and outer cylinders, the maximum water pressure, and the range of differential vertical pressures to be applied by the vertical pressure system. These items are discussed in the following paragraphs. Specifications are summarized in Table 3.

The sizes and geometries for the inner and outer cylinders were specified with the primary objective of developing a probe which is compatible with commonly used offshore drilling equipment. The field probe is intended to be lowered with a wireline through a drillstring, latched to the drillbit, and then operated (4). Thus, the outer diameter of the outer cylinder was specified to be 3 in. Larger systems could be developed specifically for onshore use. The diameter specified for the inner cylinder was judgmentally selected to be large enough to provide sufficient contact with the test soil and to allow soil to intrude into the inner cylinder during penetration, yet small enough to provide a sufficiently large and reasonably undisturbed sample. To avoid excessive disturbance to the test soil, the wall thickness of the outer cylinder was specified to be the same as that of normal, thin-walled, 3 in diameter sampling tubes. The wall thickness of the inner cylinder was specified to be even thinner to minimize disturbance to the test soil, to facilitate the intrusion of soil into the inner cylinder, and to minimize the amount of soil displaced by this cylinder. Also, we specified that the volume inside the inner cylinder should be large enough to prevent the development of significant confining pressure within the soil inside this cylinder. This feature was intended to help reduce the effect of the soil within the inner cylinder on the motion of this cylinder during testing. The active length (unshielded length) of the inner cylinder was specified to be 8 in, so that samples will have a length comparable to the length of samples for laboratory testing (~6 in) with additional length to allow for end effects. The 4 in length specified for the shield was judged sufficient to place the excitation at a depth below the base of the borehole where the soil is expected to be relatively undisturbed by the drilling process. The total length specified for the outer cylinder was equal to the sum of the shielded and unshielded lengths of the inner cylinder. The penetrating edges of both cylinders were specified to be sharpened in a manner that would minimize disturbances to the test soil during penetration. Additionally, the penetrating edges of the inner cylinder were specified to be jutted inward to divert soil entering the inner cylinder away from the wall of this cylinder during penetration. This feature is intended to help minimize the effects of the soil within the inner cylinder on the rotation of this cylinder during testing and also, to prevent the plugging of the soil within this cylinder during penetration due to wall friction.

We specified the following conservative maximum axial forces that might normally be developed during penetration: inner cylinder - 4000 lb, outer cylinder - 14,000 lb, and total - 18,000 lb. These forces correspond to penetration forces that might normally be developed in a dry, dense sand deposit at a depth of 50 ft or in a saturated, dense sand deposit at a depth of about 80 ft. Based on our experiences, we feel these specifications adequately cover penetration into the layers in which the greatest activity is expected to take place during earthquakes. Reasonable bending forces which may occur during penetration were covered partially by conservatism in our estimates of penetration forces and partially by conservatism in design (see report

from Sweet & Aiken, Appendix C). The following were assumed in calculating maximum penetration forces: a uniform, dry deposit of dense sand, depth = 50 ft, effective unit weight of soil = 122 pcf, coefficient of earth pressure at rest = 5, cohesion = 0 psf, effective angle of internal friction of soil = 45 deg, and effective angle of friction between the soil and the cylinder (clean steel) = 11 deg. The bearing capacities of the penetrating edges were roughly estimated assuming the edges to be blunt and to act as long footings. We used Meyerhof's conservative (maximum) bearing capacity factors (32): N_γ = 280 and N_q = 130.

We specified the maximum differential lateral pressure which might act on either of the two cylinders conservatively as 30,000 psf. For this specification we assumed the conditions itemized in the preceding paragraphs. The maximum differential lateral pressure acting on the inner cylinder was expected to occur immediately following the penetration of the probe and the application of vertical pressure to the test soil. At this stage, the test soil would experience full lateral pressure while the soil within the inner cylinder would be expected to experience very little lateral pressure. The maximum differential lateral pressure acting on the outer cylinder was expected to occur during low frequency, cyclic, controlled-torque tests in loose sands or silts at the occurrence of liquefaction. At this time, the test soil within the outer cylinder would bear negligible effective lateral stress, while the soil surrounding the outer cylinder would bear the original high lateral stress.

The maximum water pressure under which the testing system should operate was specified as 100,000 psf. This corresponds to a depth, in salt water, of about 1500 ft.

The differential vertical pressure to be applied by the vertical pressure system was specified to range from 0 to 6000 psf. The maximum pressure corresponds to the effective pressure developed in a uniform, dense, saturated deposit of sand at a depth of about 80 ft. Two modes of operation were specified for the vertical pressure system (see PROPOSED METHOD, pg. 22). In one mode, the constant pressure mode, the vertical pressure acting on the test soil is to remain constant throughout testing. In the second mode, the constant volume mode, the position of the piston of the vertical pressure system is to remain substantially fixed once an appropriate vertical pressure has been applied. We also requested that the feasibility of a piston system capable of operating along the entire length of the inner and outer cylinders be addressed (see PROPOSED METHOD, pg. 19).

We specified various characteristics for the inner and outer cylinders. For both the outside of the inner cylinder and the inside of the outer cylinder, we specified that the surface friction should be minimized in the vertical direction but that the traction should be maximized in the horizontal direction by providing appropriately sized vertical grooves along these surfaces. Thus, disturbance to the test soil during penetration caused by surface friction would be minimized while the tendency for slip between the test soil and the cylinder walls during testing would also be minimized. The outside surface of the outer cylinder was specified to have vertical grooves to provide more effective anchoring to the surrounding soil during

testing without causing excessive penetration force. The inside surface of the inner cylinder was specified to be smooth and to have a minimum of friction to minimize the effect of the soil within the inner cylinder on the motion of this cylinder during testing.

We specified that frictional torques developed by the sealing systems and the bearings supporting the inner cylinder be kept to an absolute minimum. This was to minimize the effects of bearing and seal friction relative to the effects of soil properties on the rotation of the inner cylinder during testing. For a similar reason, we specified that sources of viscous damping be minimized.

Excitation and Measurement Requirements--In this subsection, we present and discuss the requirements for the excitation and measurement systems for the most promising testing modes. The specified criteria were developed as part of our theoretical feasibility studies. To develop the criteria, we evaluated the behavior of the testing system considering upper and lower bounds for the appropriate soil properties. The specified preliminary design criteria are summarized in Table 3.

To determine the low amplitude dynamic shear modulus and the variation in the dynamic shear modulus with shear strain, an impulsive torque will be applied to the inner cylinder, as shown in Fig. 5(a). The cylinder is expected to respond by developing a high frequency, decaying rotational vibration. The torque and angular acceleration will be measured. We specified excitation and measurement requirements that would provide the properties of interest over a practical range of soil conditions. The range was roughly bounded at the soft end by conditions in a loose, uniform, saturated sand deposit at a shallow depth of 5 ft and at the stiff end by conditions in a dense, uniform, dry sand deposit at a significant depth of 50 ft. The corresponding range of properties is expected to cover the properties of most clay and silt deposits at comparable depths. Properties were calculated as described in·the subsections entitled Low Amplitude Dynamic Shear Modulus, pg. 39, and Variation in the Dynamic Shear Modulus with Shear Strain, pg. 43. A mass moment of inertia of 0.00028 ft-lb-sec^2 was calculated for the rotating mass.

To determine the degradation characteristics of clays, a low frequency, cyclic rotation having a uniform amplitude will be applied to the inner cylinder. In response, the inner cylinder is expected to develop the cyclic torque required to maintain the selected rotation. The torque and angular displacement will be measured. We provided excitation and measurement specifications that would result in a system that could provide degradation characteristics over a practical range of soil conditions. The range was roughly bounded at the soft end by conditions in a uniform, saturated, highly degradable deposit of clay at a depth of 5 ft, and at the stiff end by conditions in a uniform, saturated, nondegradable deposit of clay at a depth of 100 ft. Properties were calculated for these depths as described in the subsection entitled Degradation Characteristics of Clays, pg. 49. It was not necessary to consider a mass moment of inertia for the rotating mass since we used the static procedure to obtain solutions (see Solution Procedures, pg. 33).

To determine the degradation and liquefaction characteristics of sands

and silts, a low frequency, cyclic torque having a uniform amplitude will be applied to the inner cylinder, as shown in Fig. 5(b). The cylinder is expected to respond by developing a cyclic rotation. The torque and angular displacement will be measured. We provided excitation and measurement specifications that would result in a system that could provide degradation and liquefaction characteristics over a practical range of soil conditions. The range was roughly bounded at the soft end by conditions, at a depth of 5 ft, in a loose, uniform, saturated sand deposit having low resistance to degradation and liquefaction and at the stiff end by conditions, at a depth of 75 ft, in a dense, uniform, saturated sand deposit having high resistance to degradation and liquefaction. Properties were calculated for these depths as described in the subsection entitled Degradation and Liquefaction Characteristics of Sands and Silts, pg. 53. It was not necessary to consider a mass moment of inertia for the rotating mass since we used the static procedure to obtain solutions (see Solution Procedures, pg. 33).

Presentation and Discussion of Results

In this subsection, we present and discuss the results of the operational feasibility study conducted by Sweet & Aiken, concentrating on the geotechnical aspects of the testing system. Using the design criteria provided by our firm, Sweet & Aiken carried out the tasks itemized in the subsection entitled Procedures, pg. 71. In Appendix C, we present the report from Sweet & Aiken, which presents the results of the operational feasibility study, concentrating on the operational aspects of the testing system.

The results of the operational feasibility study indicate that a laboratory research prototype testing system is operationally feasible. The main components, which satisfy reasonably well our design criteria, were found to be either available or readily producible, and it was found that these components could be assembled into a convenient, workable arrangement. Thus, Sweet & Aiken developed a preliminary design for a laboratory research prototype testing system and provided a cost estimate for the detailed design and construction of such a system.

In the following subsections, we present the preliminary design of the laboratory research prototype testing system and the sequence of steps in the operation of the system. Then, we discuss the mechanical components, systems, and features, and the excitation and measurement systems of the preliminary design. Also, the cost estimate for the laboratory system is presented and discussed.

Preliminary Design of Laboratory Research Prototype Testing System—The preliminary design of the laboratory research prototype testing system consists of specified mechanical, hydraulic, and electronic components conceptually assembled into one possible arrangement. The preliminary design is presented and discussed in the report from Sweet & Aiken. Drawing 850227-10, attached to the report, is an engineering drawing showing the main components of the probe of the testing system arranged in a convenient, workable manner. Figure 2.1 in the report presents a schematic diagram of the drive and measurement systems while Fig. 3.1 presents a schematic diagram of the hydraulic circuitry for the excitation system. Sketch A-1 in the Appendix of the report from Sweet & Aiken presents a sketch showing accessory electronic components arranged in a circuit.

Operation of Laboratory Testing System—To carry out a laboratory test using a system corresponding to that of the preliminary design, the probe would be carefully penetrated into the test soil to the proper depth. The penetration force would then be relieved and the shield raised a small amount to free the inner cylinder. Then, an appropriate vertical pressure would be applied to the test soil using the vertical pressure system. The vertical pressure system would be set to operate in either the constant pressure or the constant volume mode (see PROPOSED METHOD, pg. 22). A test would be conducted by selecting and activating the desired excitation and displaying and recording the measured quantities. Recorded test results would be processed appropriately.

Mechanical Components, Systems, and Features—The mechanical components, systems, and features of the preliminary design of the laboratory research prototype testing system satisfy reasonably well the mechanical requirements specified in the subsection entitled Mechanical Requirements, pg. 71. The mechanical components, systems, and features are discussed, in detail, in the following paragraphs.

The mechanical components satisfy most specified size and geometry requirements. The only departure from specifications occurred as a result of the shield. To provide an adequately thick inner cylinder, an adequately thick shield, and adequate clearance between the cylinder and the shield, as shown in the engineering drawing, it was necessary to constrict the inner cylinder above the region of the bottom of the shield. The inside of the constriction was provided with jutted edges. These edges are intended to funnel soil in the inner cylinder away from the walls above the constriction to minimize the influence of the soil within the inner cylinder on the motion of this cylinder and to prevent the plugging of   soil within the inner cylinder during penetration.

The mechanical components, including the inner cylinder, the retractable shield, and the outer cylinder, were designed to withstand the specified maximum penetration forces and maximum differential lateral pressures. Additionally, there should be enough reserve strength in these components to withstand reasonable bending loads developed during penetration.

The probe was designed to operate under the specified maximum water pressure. The main systems which will enable satisfactory operation under this pressure include the sealing systems and a chamber air pressure system. The most important seal separates the instrumented head from the environmental fluids below. The seal can operate effectively only under a small pressure differential; thus, the chamber above the seal will be pressurized with air. This seal is the only seal which will provide resistance to the rotation of the inner cylinder, as may be seen in the engineering drawing. Effects of the seal on the motion of the inner cylinder during testing could be taken into account, if necessary, in soil-probe interaction analyses (see PROPOSED METHOD, pg. 21). The seal is a low friction seal estimated to develop a resisting torque of less than 0.15 in-lb. Air will be used, instead of a liquid, to provide the required pressure differential across the seal to minimize extraneous damping of vibrations during impulse tests. Air will be provided, at a controlled pressure, to the upper chamber of the laboratory system by a laboratory source through an air line.

The mechanical systems should perform required functions to specifications. The vertical pressure system, shown in the engineering drawing, will be provided with the selectable capability to maintain either a constant pressure on the top of the test soil or the position of the piston after application of a selected pressure. Hydraulic fluid will be introduced into the chamber at the top of the piston and the pressure or volume of the fluid will be controlled remotely.

The mechanical features of the testing system should provide specified performance characteristics. The cylinder walls will be provided with several specified features. The outside surface of the outer cylinder will be grooved vertically providing an effective reaction anchor to the surrounding soil. Both the inside surface of the outer cylinder and the outside surface of the inner cylinder will be grooved vertically and covered with a thin coat of polytetrafluoroethylene (PTFE), a low friction material. The coefficient of friction between PTFE and soil was estimated by Sweet & Aiken to be about 0.09. Thus, friction will be very low in the vertical direction to minimize disturbance to the test soil during penetration, while traction will be maximized in the tangential direction to minimize slip between the cylinders and the test soil during testing. The inside wall of the inner cylinder will be smooth and coated with PTFE to minimize friction. This will help minimize the effect of the soil inside the inner cylinder on the motion of this cylinder and the tendency of the soil to form a plug within the inner cylinder during penetration. Sweet & Aiken estimate that the PTFE coating should survive numerous tests without excessive wear. PTFE coats may be reapplied when desired. To avoid excessive difficulties with abraded cylinder coatings or grooves, the inner and outer cylinders will be removable and exchangeable. This capability would also prove useful in providing different grades of cylinder grooving for different grades of soil.

The positioning bearings, required to position the inner cylinder, were given special attention. To minimize the effects of bearing friction on the interpretation of test results, these bearings were placed above both the torque and motion measurement transducers in the preliminary design. While bearing friction will still effect behavior somewhat, this arrangement will help isolate the effects of soil properties on the behavior of the inner cylinder. Special precision bearings were selected to serve as positioning bearings to minimize friction and thus, the effects of the bearings on the motion of the inner cylinder. Sweet & Aiken indicated that the coefficient of friction of these bearings is 0.00018.

Excitation and Measurement Systems--The selected excitation and measurement systems satisfy most of the preliminary design criteria specified in the subsection entitled Excitation and Measurement Requirements, pg. 75. In the following paragraphs, we discuss the selected excitation and measurement systems.

To determine the low amplitude dynamic shear modulus and the variation in the dynamic shear modulus with shear strain, an impulsive torque will be applied to the inner cylinder, as shown in Fig. 5(a). In response, the inner cylinder is expected to rotate with a high frequency, decaying oscillation. The selected excitation system includes hydraulic motors and the equipment

required to control and operate the motors. The selected measurement systems include torque transducers, accelerometers, and the equipment required to operate these components.

Not all specified criteria were satisfied; however, the resulting consequences are not expected to be severe. The specified upper level of torque was not satisfied. We conservatively specified the highest level of torque to be applied to the inner cylinder to be 20 ft-lb. The motor and torque transducer systems selected have upper limits of about 17 ft-lb. Components could have been selected with higher limits but we preferred sacrificing increased capacity for greater accuracy at lower torques. The relatively small reduction in torque capacity should limit only moderately the capabilities of the testing system. We would probably not be able to define shear moduli at larger strains for stiffer soils at greater depths. This is not a serious limitation because, generally, it is the softer soils at shallower depths that pose the greatest threat during earthquakes.

Secondly, the response time for the servovalve which will control the motor is longer than ideal for producing an impulsive loading. Thus, we will not be able to apply loads as quickly as desired. As a result, the vibratory component of the response of the inner cylinder to a load of given amplitude may not be as pronounced as if the load were applied instantaneously. The manufacturer of the servovalve states, in the literature provided by Sweet & Aiken, that the time to develop 90% of the output in response to a step input of current is 0.0025 sec. The vibration frequencies of the inner cylinder during impulse tests were estimated to range from 700 (soft soil) to 2400 cps (stiff soil) when testing soils in the linearly elastic range and from 400 (soft soil) to 1400 cps (stiff soil) when testing soils well into the nonlinear range. Therefore, the ideal of an impulsive loading is approached more closely when testing softer soils at higher strains. Since softer soils driven to higher strains seem to cause the most problems during earthquakes, the proposed testing system would be expected to provide the greatest accuracy in defining shear moduli when testing the soils of greatest concern. If necessary, steps could be taken toward approaching more closely the ideal of an impulsive loading. One possibility includes reducing the natural frequency of the rotating mass either by shortening the active length of the inner cylinder or by increasing the mass moment of inertia of the inner cylinder.

Finally, neither a single motor nor a single torque transducer appears to be capable of satisfying specified criteria over the entire range. Thus, motors and torque transducers having different capacities will be interchanged in the laboratory research prototype testing system to provide the desired capabilities. Sweet & Aiken provided literature from a manufacturer describing a single motor, currently being developed, which may be able to satisfy our criteria. If we are unable to provide a single probe for field use that covers the entire range of interest, then, consistent with current practice, we plan to offer different probes covering different ranges.

To determine the degradation characteristics of clays, a low frequency, cyclic rotation having a uniform amplitude will be applied to the inner cylinder. In response, the cylinder is expected to develop the low frequency, cyclic torque required to maintain the selected rotation. The

selected excitation system includes the previously discussed hydraulic motors and the equipment required to control and operate the motors. The selected measurement systems include angular displacement transducers, the previously discussed torque transducers, and the equipment required to operate these components.

Not all specified criteria were satisfied; however, the resulting consequences are not expected to be severe. Sweet & Aiken was unable to find an available displacement transducer which the firm feels could be used to accurately measure angular displacements as small as those specified. However, Trans-Tek, Inc., (Trans-Tek) indicated that its 600 series angular displacement transducer could be modified and calibrated, for a modest cost, to provide the required small measurements. A letter from Trans-Tek confirming this information is included in the report from Sweet & Aiken. Two interchangeable transducers may be required to cover the specified range of angular displacements.

Also, Sweet & Aiken was unable to find a single torque transducer capable of satisfying the specified criteria over the entire range. Thus, as stated previously, for the laboratory research prototype testing system, interchangeable torque transducers having different capacities will be used.

Finally, Sweet & Aiken was uncertain whether a single hydraulic motor could accurately provide the smallest torques specified. Thus, the lower capacity motor required for impulse tests may also be required for low frequency, cyclic, controlled-rotation tests. The previously described motor which is under development may be able to satisfy the entire range of our criteria for these tests. As mentioned previously, if we are unable to provide a single probe for field use that covers the entire range of interest, then, consistent with current testing practice, we plan to offer different probes having different ranges.

To determine the degradation and liquefaction characteristics of sands and silts, a low frequency, cyclic torque having a uniform amplitude will be applied to the inner cylinder, as shown in Fig. 5(b). In response, the cylinder is expected to develop a low frequency, cyclic rotation. The selected excitation system includes the previously discussed hydraulic motors and the equipment required to control and operate the motors. The selected measurement systems include the previously discussed torque transducers and angular displacement transducers and the equipment required to operate these components.

Not all specified criteria were satisfied; however, the resulting consequences are not expected to be severe. Sweet & Aiken was unable to find a displacement transducer for accurately measuring the smallest angular displacements specified. However, it appears that a transducer could be provided that would accurately measure angles reasonably close to our specified lower limit. We specified that angular displacement amplitudes as low as 0.001 deg should be measured with reasonable accuracy. Trans-Tek indicated, in a letter included in the report from Sweet & Aiken, that a 600 series angular displacement transducer could be modified and calibrated, for a modest cost, to provide an angular displacement transducer with a resolution of 0.001 deg and a range of ±0.5 deg. Sweet & Aiken reported that Trans-Tek is certain that measurements down to 0.01 deg could be made

accurately with the testing system, but that measurements below 0.01 deg would be of questionable accuracy.

The consequences of not being able to measure accurately the smallest specified angles are not expected to be severe for several reasons. The conditions under which these small angles are expected are fairly limited. These angles are expected only when applying very low levels of shear stress to the test soil and only during the initial cycles of loading before significant degradation has taken place. To minimize uncertainty in interpreting soil properties, when these conditions occur we would place greater emphasis on behavior during the later stages of tests, during which measurements are more accurate. It would appear that enormously useful information such as the number of cycles to liquefaction or to any reasonable level of specified or limited strain could be defined quite accurately under all conditions with the testing system as currently designed. Thus, we believe that the lower bound of the range for accurate measurements indicated by Sweet & Aiken will lead to reasonably accurate and detailed descriptions of in situ cyclic degradation and liquefaction characteristics. Additionally, improvements in transducer technology may result in improvements in our ability to measure the smallest angles specified.

Two angular displacement transducers will be required in the testing system to carry out low frequency, cyclic, controlled-torque tests. A single Trans-Tek angular displacement transducer calibrated to give high accuracy in the measurement of small angles cannot cover the entire specified range of angular displacements. The entire specified range may occur during a single low frequency, cyclic, controlled-torque test. For example, when testing a loose sand at very low levels of stress, during the initial cycles of loading the inner cylinder will undergo small angular displacements. At a later stage during the test, after significant degradation has occurred, very large, virtually unrestrained, angular displacements may develop. Although Sweet & Aiken does not show two angular displacement transducers in the engineering drawing in their report, the firm indicated that two angular displacement transducers, each calibrated to a different range, may be included in the testing system. Alternatively, low frequency, cyclic, controlled-rotation tests may be carried out to determine the degradation and liquefaction characteristics of sands and silts.

Also, Sweet & Aiken were unable to find a single torque transducer capable of measuring the entire range of torques specified. As mentioned previously, interchangeable torque transducers having different capacities will be used in the laboratory research prototype testing system.

Finally, Sweet & Aiken was uncertain whether a single hydraulic motor could accurately provide the smallest torques specified. Thus, the lower capacity motor required for impulse tests may also be required for low frequency, cyclic, controlled-torque tests. The previously described motor which is under development may be able to satisfy the entire range of torques specified for the low frequency, cyclic, controlled-torque test. As mentioned previously, if we are unable to provide a single probe for field use that covers the entire range of interest, then, consistent with current testing practice, we plan to offer different probes having different ranges.

Cost Estimate--Sweet & Aiken estimated a cost of $78,260 for a

complete laboratory research prototype testing system. The estimate includes estimated costs for detailed design, purchases, manufacture, and construction. Individual costs are itemized in the report from Sweet & Aiken. The cost estimate is conservative. For the cost estimate, Sweet & Aiken assumed that all electronic components will be purchased. It is likely that we will rent many of these components for Phase II research leading to a considerable reduction in cost.

Conclusions

In this subsection, we itemize the main conclusions resulting from the operational feasibility study. These conclusions directly relate to the objectives presented in the subsection entitled Objectives, pg. 70.

1) Based on the results of the operational feasibility study conducted by the mechanical engineering firm of Sweet & Aiken, we concluded that a laboratory research prototype testing system for determining the low amplitude dynamic shear modulus and the variation in the dynamic shear modulus with shear strain by impulse test, the degradation characteristics of clays by low frequency, cyclic, controlled-rotation test, and the degradation and liquefaction characteristics of sands and silts by low frequency, cyclic, controlled-torque test is operationally feasible.

2) Sweet & Aiken developed a preliminary design for a laboratory research prototype testing system.

3) Sweet & Aiken estimated a cost of $78,260 for the design and construction of a complete laboratory research prototype testing system.

4) Not all specified criteria were satisfied by the preliminary design; however, the consequences are not expected to be severe and improvements are expected with advances in related technology. It appears that we will be unable to measure the smallest angles predicted during low frequency, cyclic, controlled-torque tests for determining the degradation and liquefaction characteristics of sands and silts. Thus, we may be unable to determine, as accurately as desired, soil behavior during the initial cycles of loading from tests conducted at lower levels of load. Also, it appears that we will not be able to apply loads as rapidly as desired during impulse tests. Thus, we may not observe as pronounced a vibratory response from the inner cylinder as would be expected during ideal impulse tests. Finally, to satisfy specified design criteria, some components of the laboratory research prototype testing system will have to be interchanged. If we are unable to provide a single probe for field use that covers entirely the range of interest, then, consistent with current testing practice, we plan to offer different probes covering different ranges. Each of the concerns identified is a product of limitations in current technology. Therefore, advances in appropriate technology may be expected to reduce or eliminate each of these concerns.

ESTIMATE OF FEASIBILITY

In this section, we provide our opinion concerning the feasibility of the proposed in situ testing system. First, we provide our general opinion, and then, we provide supporting details.

Based on the results of the feasibility studies discussed herein, we have concluded that the proposed in situ testing system is feasible. Ultimately, the system is expected to effectively provide information on in situ cyclic and dynamic soil properties to the level of accuracy and detail appropriate for the intermediate and final stages of the analysis and design of important structures located in seismically active areas. The Phase I studies indicate that a laboratory research prototype testing system is theoretically, operationally, and economically feasible. Since the additional equipment required for a field system is reasonably conventional, and since the additional problems that could be encountered in the field appear to be either small or surmountable, we believe that an in situ field testing system is feasible.

Much research and development will be required before the full potential of the system can be realized. This is consistent with comparable testing systems. Much effort will be needed because the problem of determining in situ cyclic and dynamic soil properties is not a simple one, because the testing system is relatively complex, and because the testing system has considerable safety and economic implications. As a result of our Phase I work we have already identified several areas in which work is required. We expect to identify more areas during later stages of work.

The theoretical and operational feasibility studies presented herein indicate that a laboratory research prototype testing system is theoretically, operationally, and economically feasible. To determine the theoretical feasibility of the laboratory testing system, we analytically simulated tests considering expected ranges of the soil properties of interest. The behavior of the testing system was predicted to be sensitive to each soil property considered in a clear, physically reasonable manner. As a result, we concluded that each soil property considered could be inferred from this behavior and therefore, that the testing system is a theoretically feasible means for determining each property. A simple analytical model was used to simulate tests. In practice, we feel that we will be able to infer the soil properties of interest to the required level of detail and accuracy by analytically simulating tests using descriptive analytical procedures. Powerful analytical procedures are available which are expected to be capable of describing, accurately and in detail, the geometry and performance of tests and most of the important phenomena occurring during tests. Also, results obtained using a testing system somewhat similar, in principle, to ours (5) roughly support our estimate of the theoretical feasibility of the proposed testing system.

Prior to and during the Phase I theoretical feasibility studies, areas were identified in which future research and development could improve the effectiveness of the proposed testing system. For example, advancing our ability to describe the effects of dilation on shear stress-strain behavior during cyclic loading is expected to improve our ability to accurately infer, from test results, the behavior of dense sands under cyclic loads.

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Similarly, optimizing the dimensions of several of the components of the probe is expected to improve our ability to accurately infer, from test results, all properties of interest.

We arranged for the mechanical engineering firm of Sweet & Aiken, Inc., to determine the operational feasibility of the laboratory research prototype testing system. Sweet & Aiken found that the main components, which satisfy reasonably well specified mechanical, excitation, and measurement requirements, are either available or could be produced at modest cost. Sweet & Aiken also found that these components could be assembled into a convenient, workable arrangement. Thus, we concluded that the laboratory research prototype testing system is operationally feasible. The many operating laboratory systems (37) which are similar, in principle, to our system, and which measure similar parameters at comparable magnitudes support this conclusion.

During the Phase I operational feasibility study, several areas were identified in which future development could improve the effectiveness of the proposed testing system. For example, improvements in the ability to measure small values of angular displacement would be expected to improve our ability to infer the cyclic degradation and liquefaction characteristics of sands and silts. Such improvements may realistically be expected as a result of motivation by fields such as robotics for which small precision measurement systems are required.

To help us determine the economic feasibility of the proposed testing system, Sweet & Aiken provided a cost estimate for the detailed design and construction of a laboratory research prototype testing system. In view of potential benefits and considering the estimated cost for the development of comparable testing systems, the laboratory research prototype testing system was judged to be economically feasible.

We believe that a field in situ testing system will also be theoretically, operationally, and economically feasible. A field system is expected to be theoretically feasible because the principle of operation of the field system will be identical to that of the laboratory research prototype testing system. However, under severe conditions, for example, when testing offshore in high sea states, behavior may differ somewhat from behavior under milder conditions. This may occur as a result of additional disturbance induced into the test soil due to wave and current-induced movement of the probe. A number of steps are routinely taken to reduce such movements during offshore undisturbed sampling and cone penetrometer and remote vane testing (4). Continued advancement in this area is expected. Additionally, it is expected that other steps unique to our testing system could be taken to further reduce the effects of such movements. Such difficulties are not expected during onshore tests.

We feel that a field testing system is operationally feasible because the additional components which may be required for a field testing system are considered to be existing or feasible technology. At this time, though other modes of operation may be possible, we intend for the field probe to be lowered with a wireline through a drillstring, latched to the drillbit, and then operated. The additional components which may be required include a penetration system, a drillbit latching system, a downhole electronics

package, an umbilical, and a winch. Such components have been used successfully with other in situ geotechnical testing systems (4) and also, as indicated in the report from Sweet & Aiken in Appendix C, in other comparable applications.

Also, we feel that a field testing system is operationally feasible because the additional problems which we expect to encounter in the field are expected to be either small or surmountable. Several problems may be encountered. These include problems caused by a corrosive environment, mechanical noise, and rocks.

As indicated by Sweet & Aiken, the effects of a corrosive environment can be minimized by proper maintenance, care, and selection of materials.

Mechanical noise, which may be present in the field, is expected to have the greatest effect on the interpretation of results from tests involving excitations and responses having small magnitudes. A number of seismic field tests, which involve the measurement of small quantities comparable in magnitude to the quantities to be measured by the proposed system (10) (17) (26) (34) (37), have been successfully carried out. Our proposed testing system is primarily intended to provide information on the in situ shear stress-strain behavior of a soil at higher amplitudes of strain. We do not expect major difficulties from mechanical noise during higher amplitude tests. Intuitively, it would appear that general mechanical noise would not create large problems with the proposed testing system. Vertical disturbances would not be expected to be sensed. Horizontal disturbances which do not tend to significantly rotate the inner cylinder would be similarly sensed by the two accelerometers and it would appear that the electrical signals arising from such disturbances could be made to cancel each other. Also, Stokoe et al. (33) have measured noise at the bottom of an offshore borehole. They found that horizontal motions due to noise were significantly less than vertical motions due to noise. Should mechanical noise present problems, one possible solution is special signal processing (33).

Rocks may damage the inner and outer cylinders during penetration. To minimize this difficulty, we plan to include the feature of removable and exchangeable cylinders in a field testing system. Also, we are considering a larger, somewhat stronger probe for use in coarser deposits.

We feel that a field testing system is economically feasible because neither the additional equipment required for a field testing system, nor the additional difficulty which may be encountered in the field is expected to present an unusually large, excessive cost. As stated previously, much of the additional equipment required for the field is relatively conventional and all of   this equipment   has been used either for in situ geotechnical testing or for other comparable applications. Also, as implied previously, we do not believe that particularly unusual measures will be required to minimize or avoid the problems which may be encountered in the field. Finally, the benefits of a working field system are expected to greatly exceed the overall cost. For example, based on our experiences, we believe that the cost saving alone involved in basing the design of a single major structure on appropriately accurate and detailed information on in situ cyclic and dynamic soil properties could far exceed the cost of the development of the testing system.

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APPENDICES

A-0

Appendix A - Validation of Computer Procedures

In this appendix, we present, in outline form, the validations of the computer procedures used in the theoretical feasibility study for simulating 1) impulse tests for determining the low amplitude dynamic shear modulus, 2) high frequency, cyclic tests for determining the low amplitude dynamic shear modulus, 3) impulse tests for determining the variation in the dynamic shear modulus with shear strain, 4) high frequency, cyclic tests for determining the variation in the dynamic shear modulus with shear strain, 5) low frequency, cyclic, controlled-torque tests for determining the degradation and liquefaction characteristics of sands and silts, and 6) low frequency, cyclic, controlled-rotation tests for determining the degradation characteristics of clays.

1) Validation of computer procedure for simulating impulse tests for determining the low amplitude dynamic shear modulus.

   Validation Procedure: Compared computer solution with closed-form analytical solution (6).

   Parameters: G_O = 1.72×10^6 psf, K_{TO} = 1.88×10^4 ft-lb/rad, I = 2.8×10^{-4} lb-ft-sec^2, C_T = 0 ft-lb-sec/rad, α = 1, R = 3, C_1 = 0.8, τ_m = 6.6×10^5 psf, computational time step = 1×10^{-5} sec.

   Excitation: Rectangular, impulsive torque which induced only linear behavior; amplitude = 10 ft-lb, duration = 6.5×10^{-4} sec.

   Response: Decaying, cyclic, rotations.

   Results: As shown in Figs. A-1(a) and (b), computed responses agree closely with theoretical responses. Also, as shown in Fig. A-1(c), energy balance requirements are satisfied.
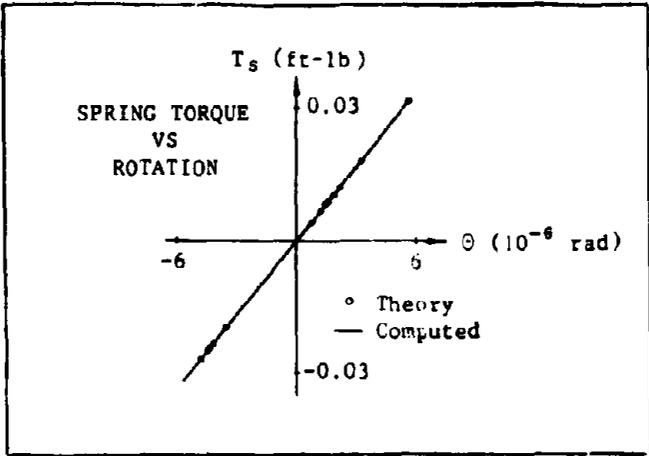
   Conclusion: The procedure for simulating impulse tests for determining the low amplitude dynamic shear modulus is valid.

2) Validation of computer procedure for simulating high frequency, cyclic tests for determining the low amplitude dynamic shear modulus.

   Validation Procedure: Compared computer solution with closed-form analytical solution (6).

   Parameters: G_O = 5.14×10^5 psf, K_{TO} = 5610 ft-lb/rad, I = 2.8×10^{-4} lb-ft-sec^2, C_T = 0.50 ft-lb-sec/rad, α = 1, R = 5, C_1 = 0.8, τ_m = 100 psf, computational time step = 3.5×10^{-5} sec.

   Excitation: High frequency, sinusoidal torque which induced only linear behavior; amplitude = 0.015 ft-lb, frequency = 570 cps.

   Response: Decaying transient and steady state, high frequency, cyclic rotations.
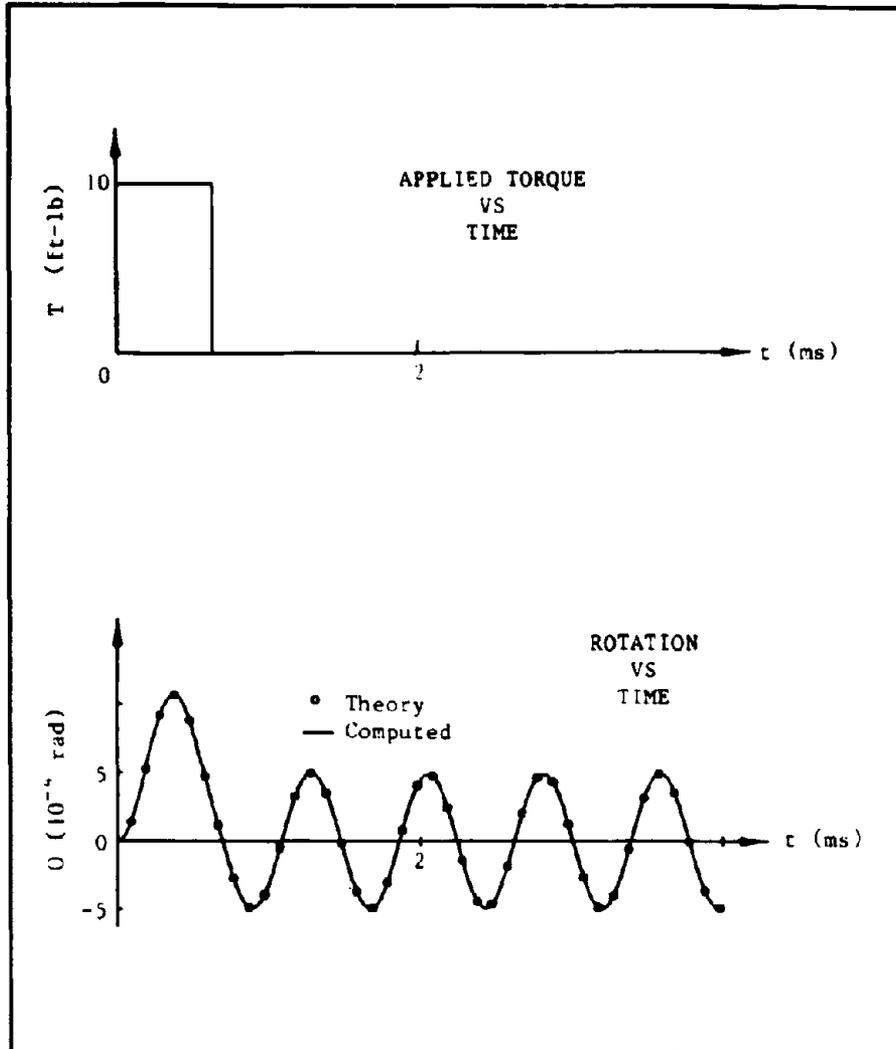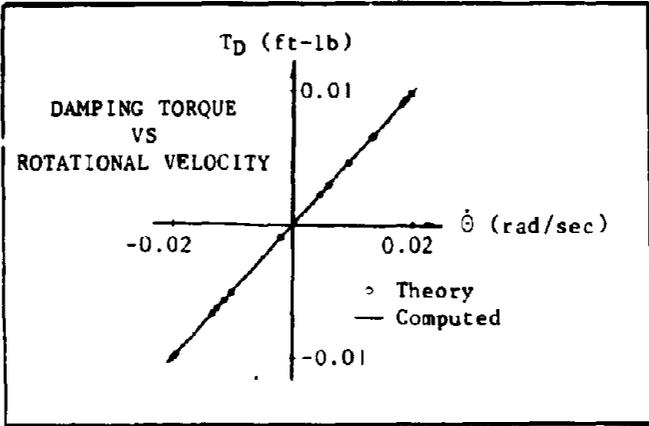
(a) Excitation/Response

Figure A-1: Validation of Computer Procedure for Simulating Impulse Tests for Determining the Low Amplitude Dynamic Shear Modulus
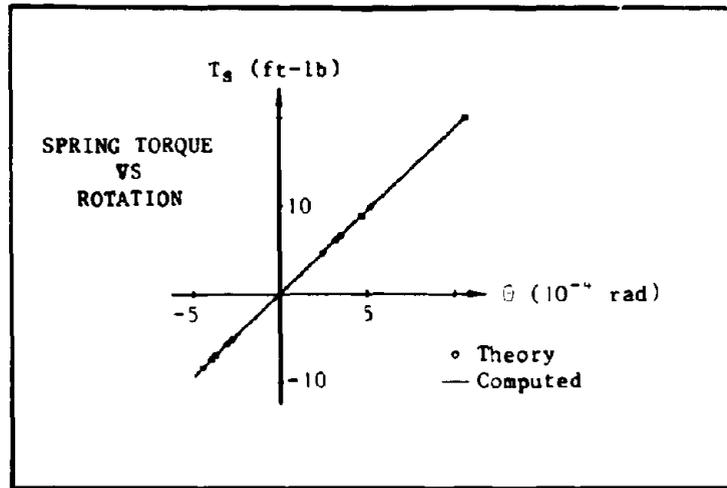
(b) Spring Torque-Rotation Behavior
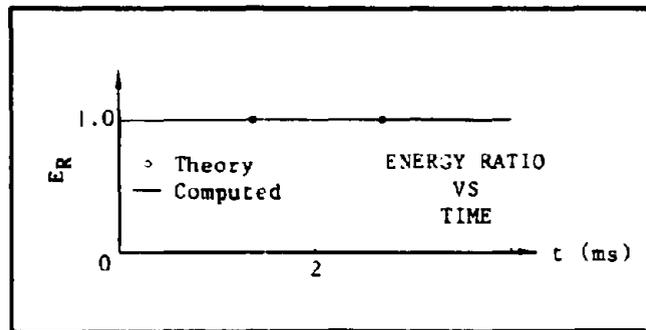
(c) Energy Ratio

Figure A-1: Validation of Computer Procedure for Simulating Impulse Tests for Determining the Low Amplitude Dynamic Shear Modulus

Results: As shown in Figs. A-2(a) through (c), all computed responses agree closely with theoretical responses. Also, as shown in Fig. A-2(d), energy balance requirements are satisfied.

Conclusion: The procedure for simulating high frequency, cyclic tests for determining the low amplitude dynamic shear modulus is valid.

3) Validation of computer procedure for simulating impulse tests for determining the variation in the dynamic shear modulus with shear strain.

Validation Procedure: Obtained computer solution, checked energy balance and torque-rotation behavior, and judgmentally checked solution.

Parameters: $G_0 = 2 \times 10^6$ psf, $K_{To} = 2.2 \times 10^4$ ft-lb/rad, $I = 2.8 \times 10^{-4}$ lb-ft-sec^2, $C_T = 0.1$ ft-lb-sec/rad, $\alpha = 1$, $R = 5$, $C_1 = 0.8$, $\tau_m = 1000$ psf, computational time step $= 8.9 \times 10^{-6}$ sec.

Excitation: Rectangular, impulsive torque which induced nonlinear behavior; amplitude $= 5.2$ ft-lb, duration $= 1.1 \times 10^{-3}$ sec.

Response: Decaying, cyclic rotations and permanent rotation.

Results: As shown in Fig. A-3(d), energy balance requirements are satisfied. Checking the solution judgmentally, as seen in Fig. A-3(a), during the initial period of higher amplitudes of motion, the apparent natural frequency was lower than during the subsequent period of lower amplitude motions. This is to be expected with a test soil that decreases in stiffness with increasing shear strain. As would be expected, the frequency of the later, lower amplitude free-vibrations corresponds closely to the damped natural frequency of the system at low amplitudes of rotation. The rate of decay of these free-vibrations corresponds to that expected of a system having the specified level of viscous damping. Also, as shown in Fig. A-3(a), permanent rotation, expected when testing a nonlinear, inelastic soil at high levels of strain, was predicted. As shown in Fig. A-3(b), the computed, nonlinear spring torque-rotation curve agrees closely with that estimated directly using Ramberg-Osgood stress-strain equations (24). Differences between these curves were found to decrease with decreases in the computational time step. The computed damping torque-rotational velocity curve, shown in Fig. A-3(c), is linear with the correct slope.

Conclusion: The procedure for simulating impulse tests for determining the variation in the dynamic shear modulus with shear strain is valid.

4) Validation of computer procedure for simulating high frequency, cyclic tests for determining the variation in the dynamic shear modulus with shear strain.
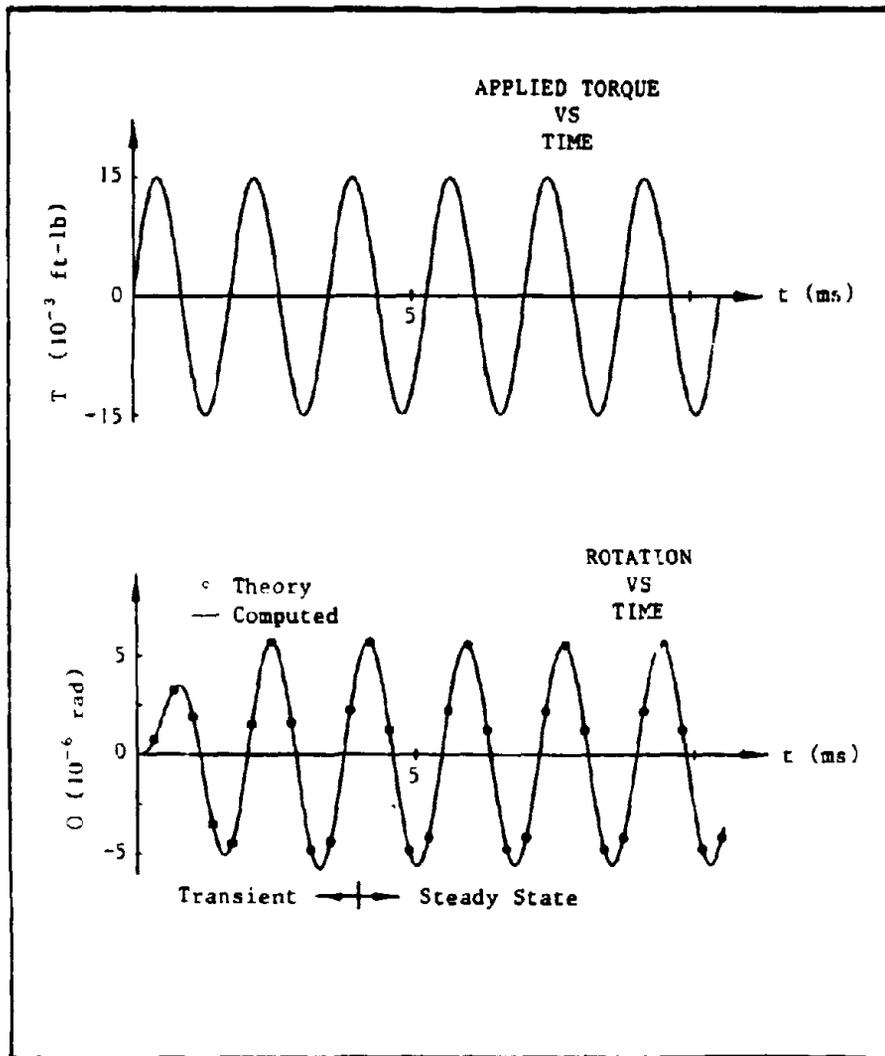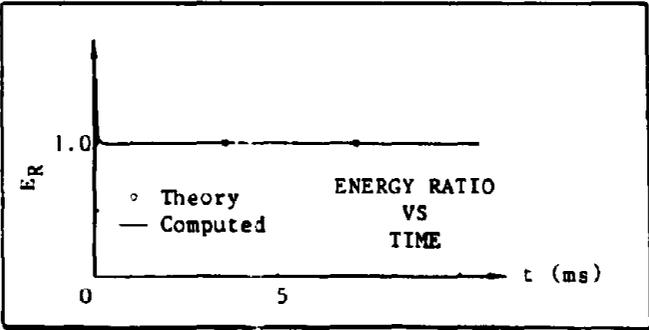
(a) Excitation/Response

Figure A-2: Validation of Computer Procedure for Simulating High Frequency, Cyclic Tests for Determining the Low Amplitude Dynamic Shear Modulus

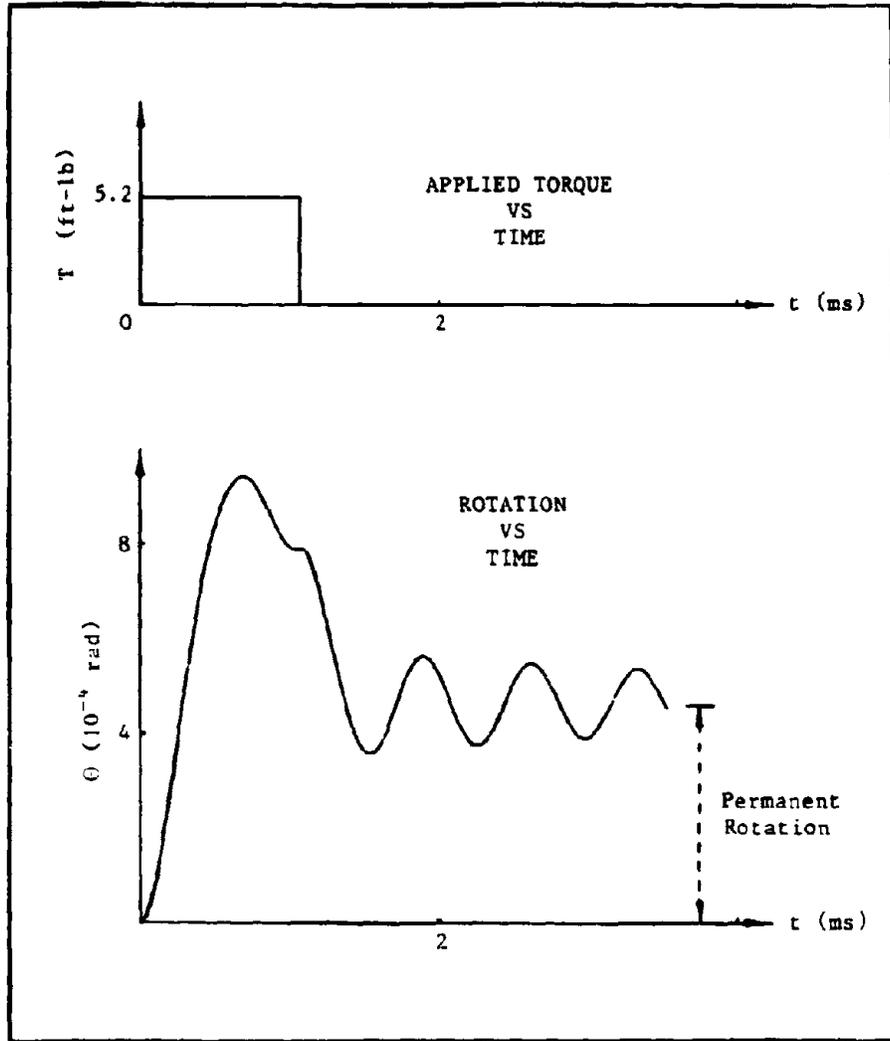(b) Spring Torque-Rotation Behavior

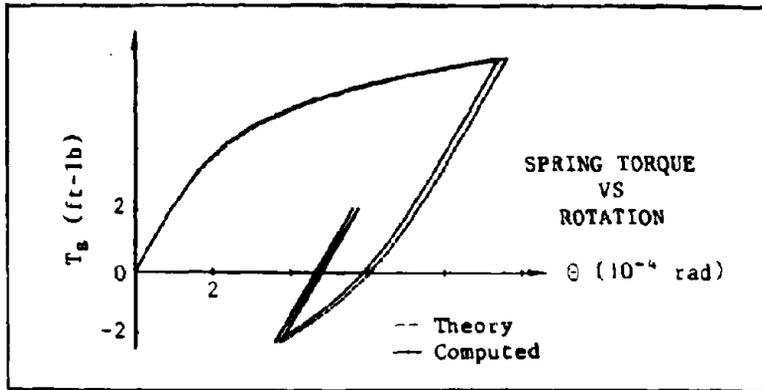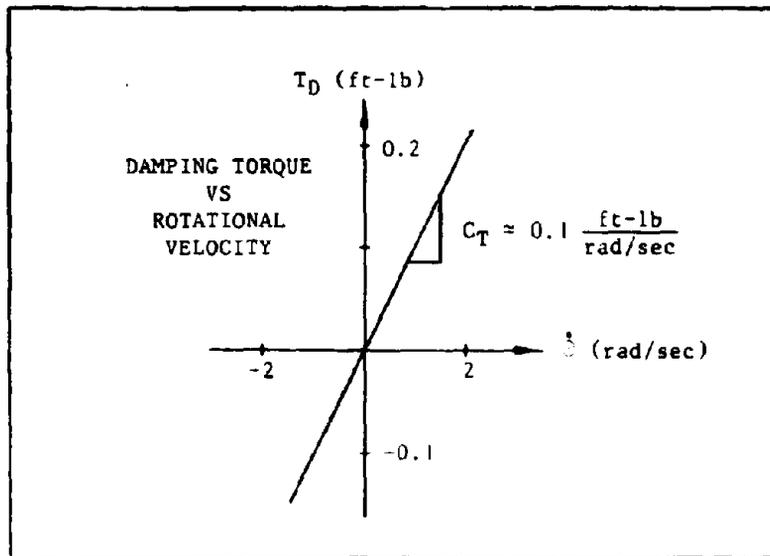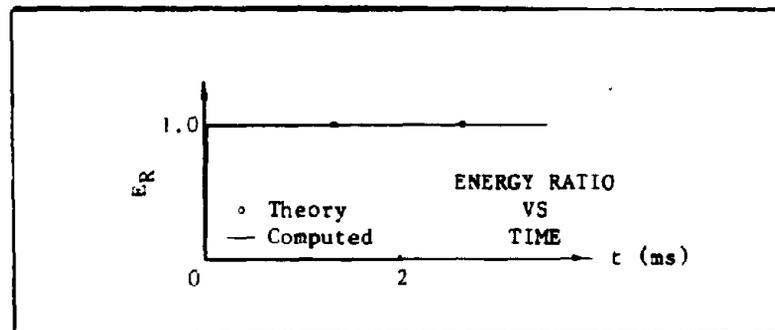(c) Damping Torque-Rotational Velocity Behavior

(d) Energy Ratio

Figure A-2:   Validation of Computer Procedure for Simulating High
Frequency, Cyclic Tests for Determining the Low Amplitude Dynamic Shear
Modulus

A-6

(a) Excitation/Response

Figure A-3: Validation of Computer Procedure for Simulating Impulse Tests
for Determining the Variation in Dynamic Shear Modulus with Shear Strain

(b) Spring Torque-Rotation Behavior

(c) Damping Torque-Rotational Velocity Behavior

(d) Energy Ratio

Figure A-3: Validation of Computer Procedure for Simulating Impulse Tests for Determining the Variation in Dynamic Shear Modulus with Shear Strain

Validation Procedure: Obtained computer solution, checked energy balance and torque-rotation behavior, and judgmentally checked solution.

Parameters: G_O = 5.14x10^5 psf, K_{T_O} = 5610 ft-lb/rad, I = 2.8x10^{-4} lb-ft-sec^2, C_T = 0.5 ft-lb-sec/rad, R = 5, α = 1, C_1 = 0.8, τ_m = 100 psf, computational time step = 1.8x10^{-5} sec.

Excitation: High frequency, sinusoidal torque which induced nonlinear behavior; amplitude = 2.7 ft-lb, frequency = 570 cps.

Response: Decaying transient and steady state, high frequency, cyclic rotations.

Results: As shown in Fig. A-4(d), energy balance requirements are satisfied. Checking the solution judgmentally, as seen in Fig. A-4(a) and as would be expected, the frequency of the response, under steady state conditions, is equal to that of the excitation. As shown in Fig. A-4(b), the computed, nonlinear spring torque-rotation curve agrees closely with the curve estimated directly using Ramberg-Osgood stress-strain equations (24). Differences between these curves were found to decrease with decreases in the computational time step. The computed, linear damping torque-rotational velocity curve, shown in Fig. A-4(c), is linear with the correct slope.

Conclusion: The procedure for simulating high frequency, cyclic tests for determining the variation in the dynamic shear modulus with shear strain is valid.

5) Validation of computer procedures for simulating low frequency, cyclic, controlled-torque tests for determining the degradation and liquefaction characteristics of sands and silts.

Dynamic Solution Procedure

Validation Procedure: Obtained computer solution, checked energy balance and torque-rotation behavior, and judgmentally checked solution.

Parameters: G_{mo} = 1.55x10^6 psf, K_{T_O} = 1.7x10^4 ft-lb/rad, I = 2.8x10^{-4} lb-ft-sec^2, C_T = 0.09 ft-lb-sec/rad, τ_{mo} = 800 psf, C_1 = 0.666, C_2 = 1.968, C_3 = 4.761, C_4 = 3.865, k_2 = 0.0025, m = 0.43, n = 0.62, computational time step = 8x10^{-5} sec.

Excitation: Low frequency, sinusoidal torque; amplitude = 1.7 ft-lb, frequency = 1 cps.

Response: Negligible transient rotation and low frequency, cyclic rotation having an amplitude which increased with an increase in the number of cycles of loading.

Results: As shown in Fig. A-5(d), energy balance requirements are

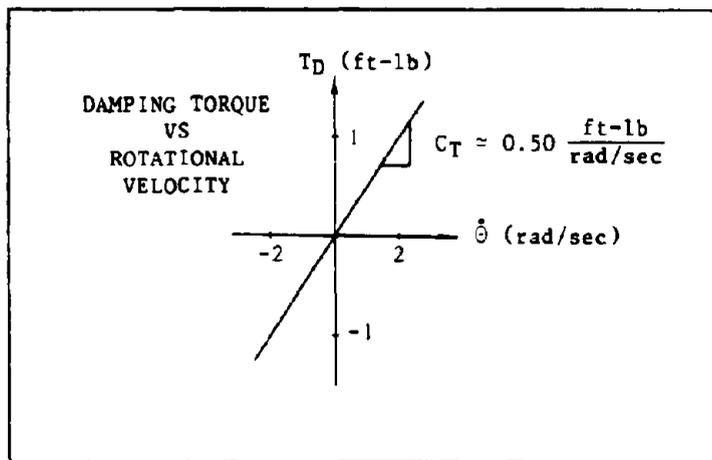APPLIED TORQUE
VS
TIME
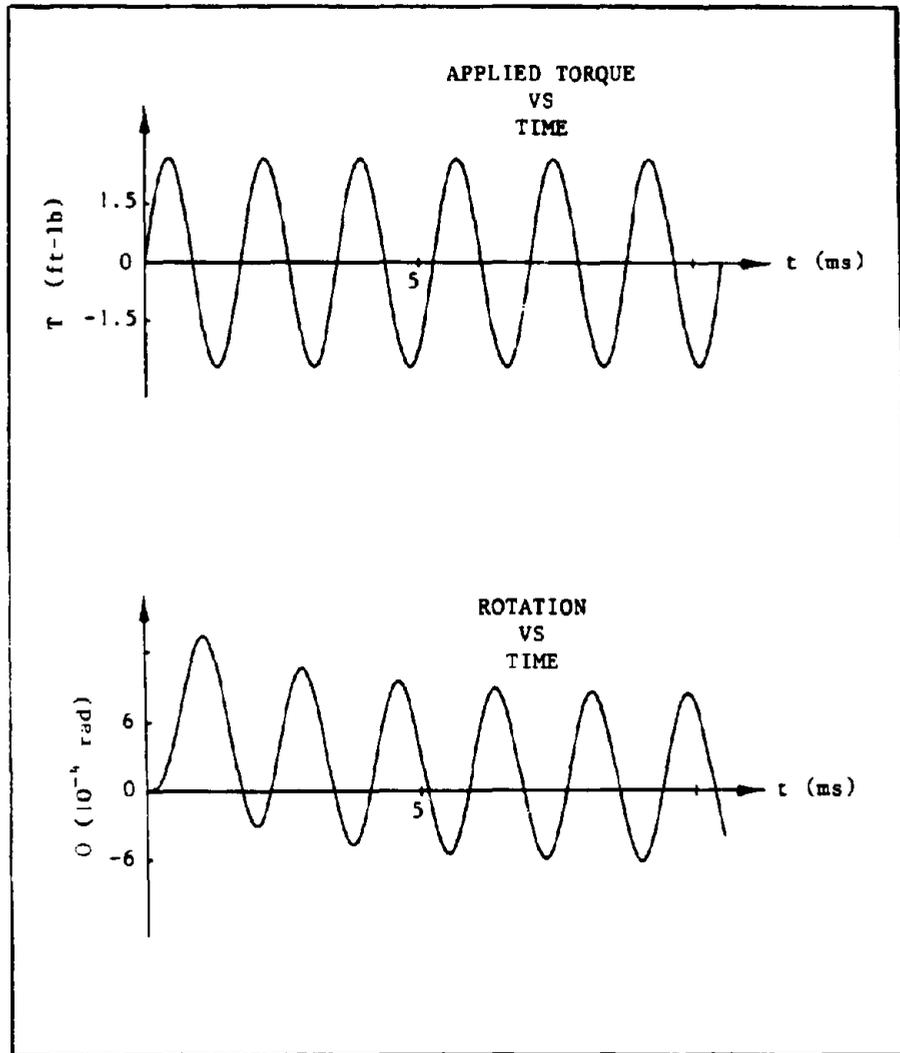
ROTATION
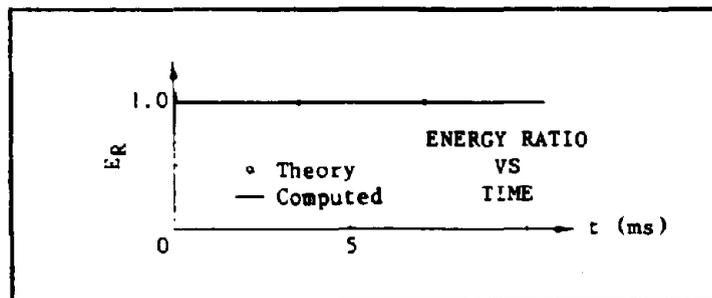VS
TIME

(a) Excitation/Response

Figure A-4: Validation of Computer Procedure for Simulating High
Frequency, Cyclic Tests for Determining the Variation in Dynamic Shear
Modulus with Shear Strain
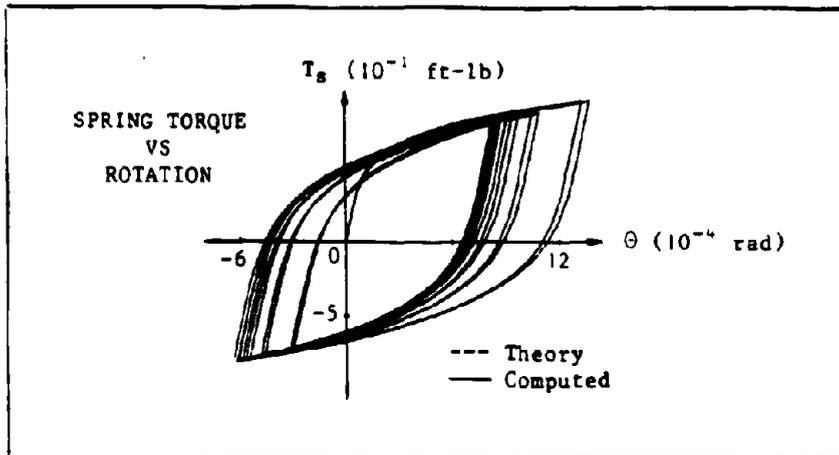
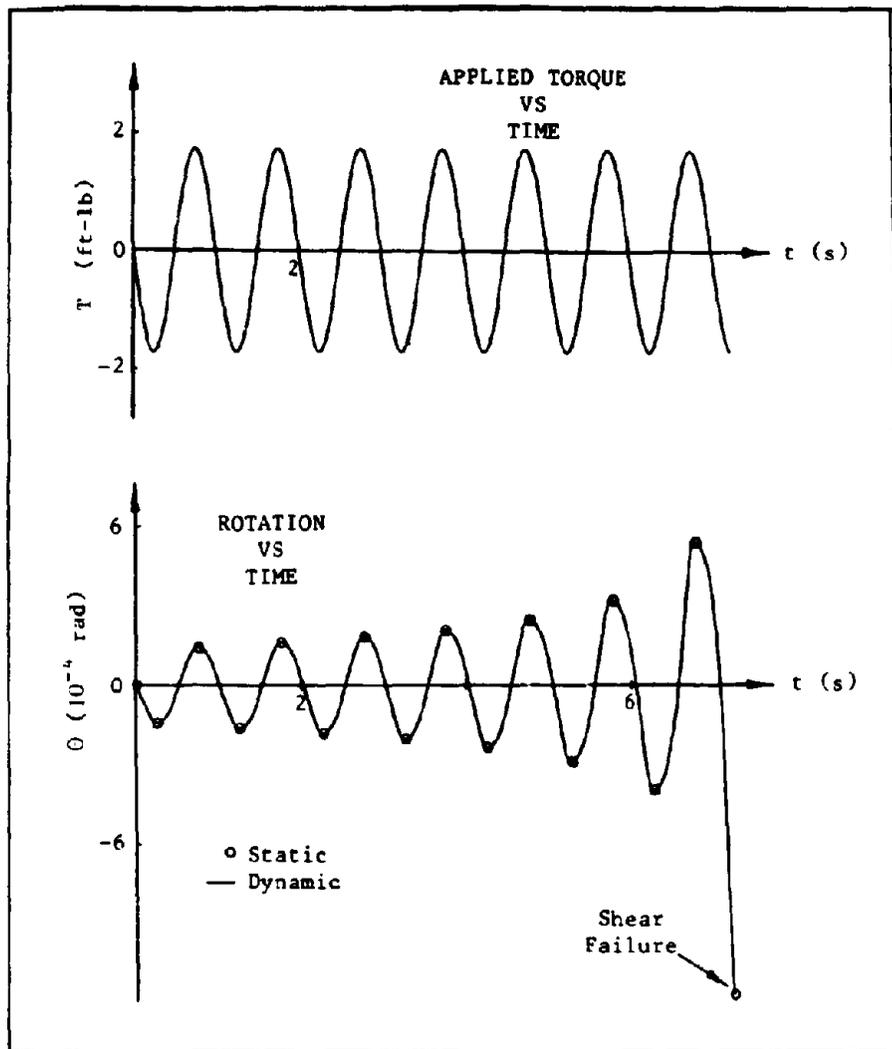(b) Spring Torque-Rotation Behavior

(c) Damping Torque-Rotational Velocity Behavior

(d) Energy Ratio

Figure A-4: Validation of Computer Procedure for Simulating High Frequency, Cyclic Tests for Determining the Variation in Dynamic Shear Modulus with Shear Strain

A-11

(a) Excitation/Response

Figure A-5: Validation of Computer Procedures for Simulating Low Frequency, Cyclic, Controlled-Torque Tests for Determining Degradation and Liquefaction Characteristics of Sands and Silts
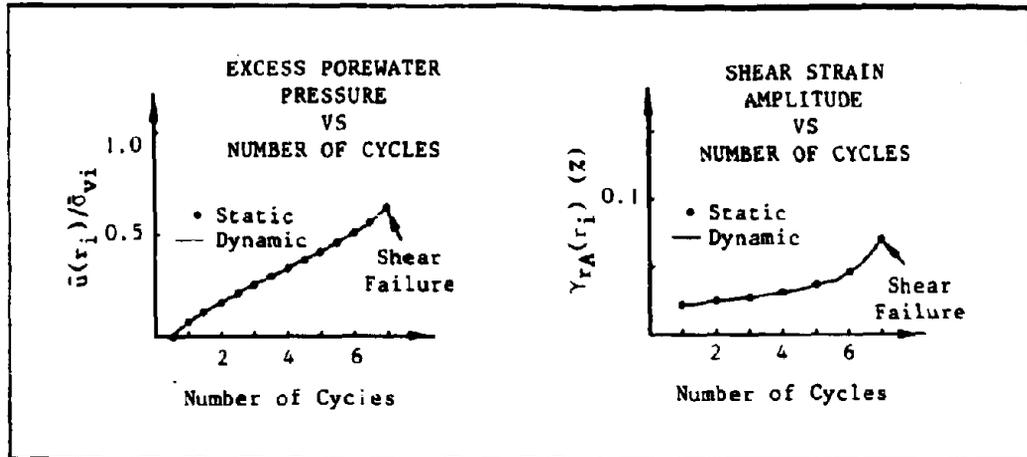
(b) Excess Porewater Pressure and Shear Strain Histories
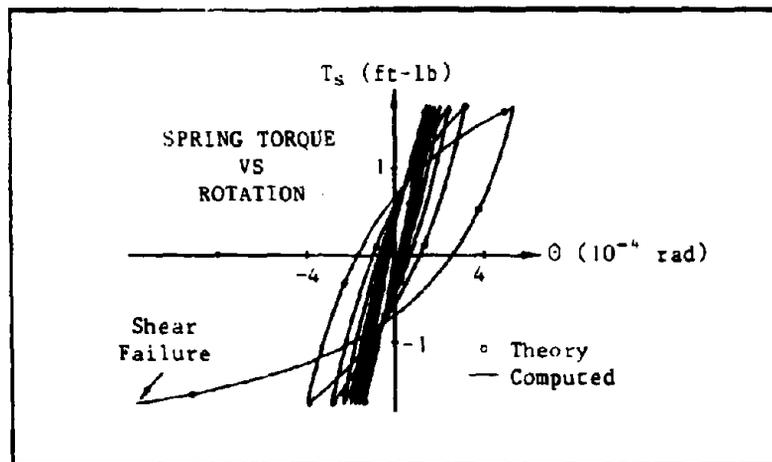
(c) Spring Torque-Rotation Behavior
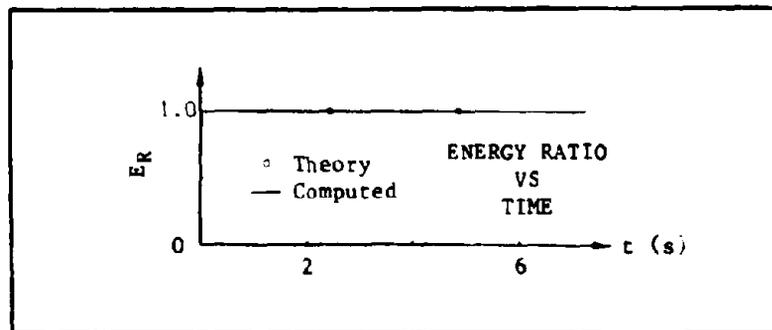
(d) Energy Ratio

Figure A-5: Validation of Computer Procedures for Simulating Low Frequency, Cyclic, Controlled-Torque Tests for Determining Degradation and Liquefaction Characteristics of Sands and Silts

A-13

satisfied. Checking the solution judgmentally, as seen in Figs. A-5(a) and (b), the amplitude of the cyclic rotation, the excess porewater pressure ratio, and the amplitude of the cyclic shear strain increased in the manner expected with an increase in the number of cycles of loading. A shear failure of the test soil was predicted during the 8th cycle of loading. As shown in Fig. A-5(c), the computed spring torque-rotation curve shows the expected degradation and agrees closely with the curve estimated directly using hyperbolic stress-strain equations (12). The effects of viscous damping were found to be negligible because of the low excitation and response frequencies considered.

Conclusion: The dynamic solution procedure for simulating low frequency, cyclic, controlled-torque tests for determining the degradation and liquefaction characteristics of sands and silts is valid.

Static Solution Procedure

Validation Procedure: Using static procedure, obtained computer solution and compared solution to above-discussed dynamic computer solution.

Parameters: $G_{mo} = 1.55 \times 10^6$ psf, $K_{To} = 1.7 \times 10^4$ ft-lb/rad, $\tau_{mo} = 800$ psf, $C_1 = 0.666$, $C_2 = 1.968$, $C_3 = 4.761$, $C_4 = 3.865$, $k_2 = 0.0025$, m $= 0.43$, n $= 0.62$.

Excitation: Cyclic torque having a uniform amplitude; amplitude = 1.7 ft-lb.

Response: Cyclic rotation having an amplitude which increased with an increase in the number of cycles of loading. The static procedure was programmed to give response parameters only at every half-cycle.

Results: As shown in Fig. A-5(a), the peaks of the cyclic rotation, computed using the static solution procedure, agree quite well with those computed using the validated dynamic procedure. Additionally, as shown in Fig. A-5(b), the histories of the excess porewater pressure ratio and the amplitude of the cyclic shear strain, computed using the static solution procedure, agree quite well with those computed using the dynamic procedure. Finally, as implied by Fig. A-5(a), the peaks of the torque-rotation history, computed using the static procedure, agree well with those computed using the dynamic procedure.

Conclusions: The static solution procedure for simulating low frequency, cyclic, controlled-torque tests for determining the degradation and liquefaction characteristics of sands and silts can provide good approximations of dynamic solutions when excitation frequencies are low. The static solution procedure for simulating such tests is valid.

6) Validation of computer procedure for simulating low frequency,

cyclic, controlled-rotation tests for determining the degradation
characteristics of clays.

Dynamic Solution Procedure

Based on our experiences with the dynamic solution procedure for
simulating low frequency, cyclic, controlled-torque tests for
determining the degradation and liquefaction characteristics of
sands and silts, and considering the reasonable similarity between
the characteristics of clays and those of sands and silts, we decided
that it was unnecessary, for the theoretical feasibility study, to develop
a dynamic solution procedure for simulating low frequency, cyclic controlled-
rotation tests for determining the degradation characteristics of clays.

Static Solution Procedure

Validation Procedure: Obtained computer solution, checked
torque-rotation behavior, and judgmentally checked solution.   An
energy balance was not carried out because the energy balance is
not an effective method for checking static solutions.

Parameters: G_{mo} = 7.2×10^5 psf, K_{TO} = 7900 ft-lb/rad, degradation
parameter curve for highly degradable clay (see Fig. 19, pg. 50), T_{mo}
= 660 psf.

Excitation: Cyclic rotation having uniform amplitude; amplitude =
0.02 rad.

Response: Cyclic torque having an amplitude which decreased with
an increase in the number of cycles of loading. The static
procedure was programmed to give response parameters only at every
half-cycle.

Results: As shown in Fig. A-6, the peaks of the cyclic torque
decrease with an increase in the number of cycles of loading as
would be expected. Additionally, the peaks of the cyclic torque
computed using the static solution procedure, agree closely with
those estimated directly using hyperbolic stress-strain equations (12).
This also implies that the spring torque-rotation peaks obtained
using the static solution procedure agree closely with those
estimated directly using hyperbolic stress-strain equations (12).

Conclusions: Based on our experiences in validating the procedures
for simulating tests for determining the degradation and
liquefaction characteristics of sands and silts, the static
solution procedure was judged to be an effective means for
simulating low frequency, cyclic, controlled-rotation tests for
determining the degradation characteristics of clays. The static
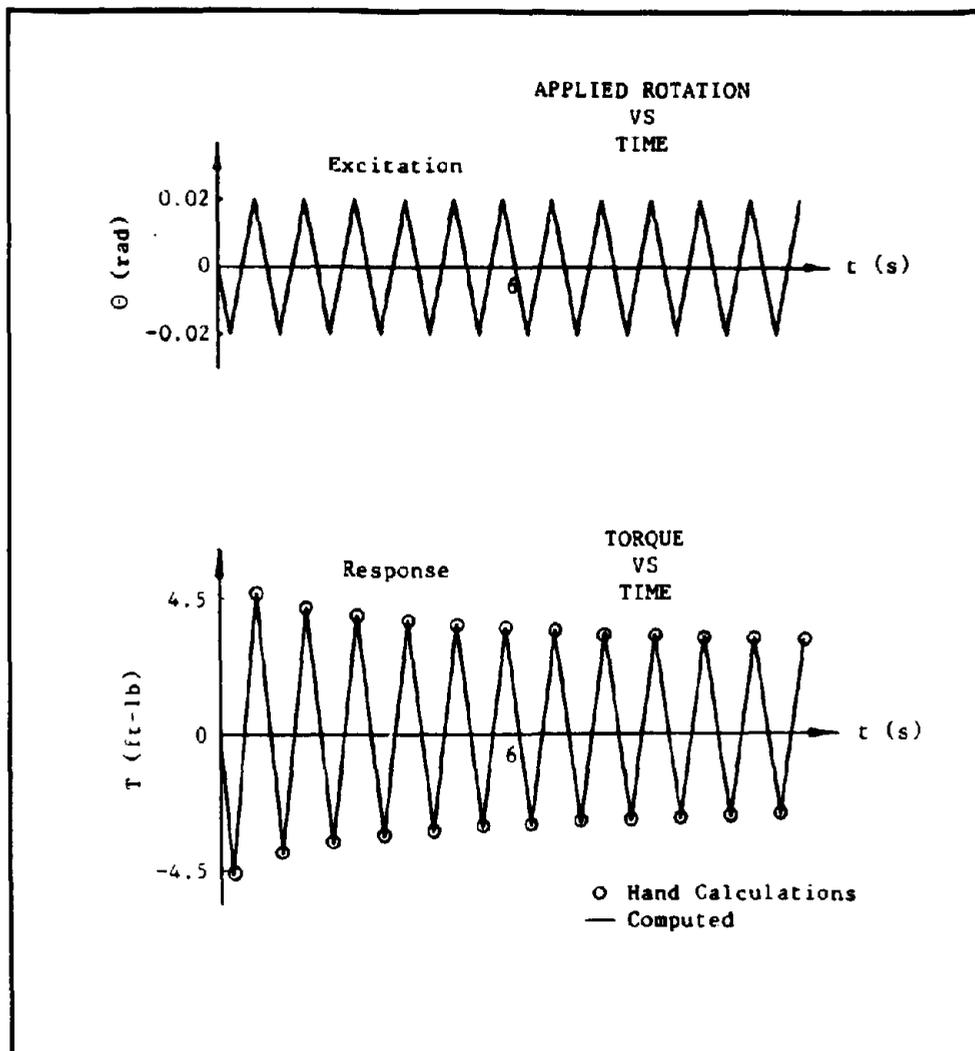solution procedure for simulating such tests is valid.

Figure A-6: Validation of Computer Procedure for Simulating Low Frequency,
Cyclic, Controlled-Rotation Tests for Determining the Degradation
Characteristics of Clays

Appendix B - Derivations for Key Relationships

In this appendix, we present our derivations for 1) the equation relating the spring torque, T_s, to the corresponding shear stress developed in the test soil along the wall of the inner cylinder, $\tau_{rs}(r_i)$, and related equations, 2) the equation relating the rotation of the inner cylinder, Θ, to the shear strain developed in the test soil along the wall of the inner cylinder, $\gamma_r(r_i)$, 3) the equation for the tangent spring stiffness, K_T, of the single-degree-of-freedom system, 4) the equation defining the rotational flexibility of the inner cylinder relative to the rotational flexibility of the test soil during the initial cycle of loading, F_i, and 5) the equation defining the rotational stiffness of the inner cylinder relative to the rotational stiffness of the test soil during the ultimate cycle of loading, S_u.

1) Derivation for the equation relating the spring torque, T_s, to the corresponding shear stress developed in the test soil along the wall of the inner cylinder, $\tau_{rs}(r_i)$, and related equations.

From Fig. 9 and basic principles of mechanics, we may derive the following relationship assuming a uniform distribution of shear stress in the vertical direction within the test soil:

$$T_s = -\tau_{rs}(r_i)Ar_i \qquad (B-1)$$

where A = surface area of active (unshielded) portion of inner cylinder
    r_i = outer radius of inner cylinder

Using basic geometry and rearranging terms, Eq. B-1 may be rewritten as follows:

$$\tau_{rs}(r_i) = -\frac{T_s}{2\pi r_i^2 l} \qquad (B-2)$$

where l = length of active portion of inner cylinder

We may infer, from Eq. B-2 the equation of interest:

$$\Delta\tau_{rs}(r_i) = -\frac{\Delta T_s}{2\pi r_i^2 l} \qquad (1)$$

If viscous damping and inertia forces are negligible, $T \simeq T_s$, and $\tau_r(r_i) \simeq \tau_{rs}(r_i)$ and we may rewrite Eq. B-2 as follows:

$$\tau_r(r_i) = - \frac{T}{2\pi r_i^2 l}$$

(B-3)

where T = torque applied to inner cylinder

$\tau_r(r_i)$ = shear stress in test soil along wall of inner cylinder

Equation B-3 may be rearranged and rewritten in terms of the amplitudes of cyclic quantities, T_A and $\tau_{rA}(r_i)$, giving the following expression of interest:

$$T_A = -2\pi r_i^2 l \bar{\sigma}_{vi} \left(\frac{\tau_{rA}(r_i)}{\bar{\sigma}_{vi}}\right)$$

(2)

where $\bar{\sigma}_{vi}$ = initial effective vertical stress

$\tau_{rA}(r_i)/\bar{\sigma}_{vi}$ = shear stress ratio

2) Derivation for the equation relating the rotation of the inner cylinder, Θ, to the shear strain developed in the test soil along the wall of the inner cylinder, $\gamma_r(r_i)$.

Assuming an axisymmetric, linear, horizontal displacement distribution in the test soil, as shown in Fig. 10, we may write the following equation:

$$u(r) = \frac{r_o - r}{r_o - r_i} u(r_i)$$

(B-4)

where $u(r)$ = horizontal displacement of test soil

r = radius

r_o = inner radius of outer cylinder

r_i = outer radius of inner cylinder

From basic principles of mechanics (16):

$$\gamma_r(r) = \frac{\partial u(r)}{\partial r} - \frac{u(r)}{r}$$

(B-5)

where $\gamma_r(r)$ = shear strain in horizontal planes

B-2

Applying Eq. B-5 to Eq. B-4 leads to:

$$\gamma_r(r) = -(\frac{r_o}{r_o-r_i}) \frac{u(r_i)}{r}$$ (B-6)

Evaluating Eq. B-6 along the wall of the inner cylinder gives:

$$\gamma_r(r_i) = -(\frac{r_o}{r_o-r_i}) \frac{u(r_i)}{r_i}$$ (B-7)

The following equation is based on the assumption that slip does not occur between the test soil and the wall of the inner cylinder:

$$u(r_i) = \Theta r_i$$ (B-8)

where Θ = rotation of inner cylinder

Substituting Eq. B-8 into Eq. B-7 gives:

$$\gamma_r(r_i) = -(\frac{r_o}{r_o-r_i}) \Theta$$ (B-9)

We may infer from Eq. B-9 the equation of interest:

$$\Delta\gamma_r(r_i) = -(\frac{r_o}{r_o-r_i}) \Delta\Theta$$ (3)

3) Derivation for the equation for the tangent spring stiffness, K_T, of the single-degree-of-freedom system.

We derived the tangent spring stiffness, K_T, from the following shear stress-strain relation:

$$G(r_i) = \frac{\Delta\tau_{rs}(r_i)}{\Delta\gamma_r(r_i)}$$ (B-10)

where $G(r_i)$ = shear modulus of test soil along wall of inner cylinder
$\tau_{rs}(r_i)$ = shear stress in test soil along wall of inner cylinder caused by shear strain
$\gamma_r(r_i)$ = shear strain in test soil along wall of inner cylinder
r_i = outer radius of inner cylinder

Substituting Eq. 3, pg. B-3, and Eq. 1, pg. B-1, into Eq. B-10 gives the following equation:

$$G(r_i) = \frac{1}{2\pi r_i^2 l} \left(\frac{r_o - r_i}{r_o}\right) \frac{\Delta T_s}{\Delta \Theta} \tag{B-11}$$

The term $\Delta T_s / \Delta \Theta$ in Eq. B-11 may be recognized as the tangent spring stiffness, K_T, of the single-degree-of-freedom system. Substituting K_T for $\Delta T_s / \Delta \Theta$ in Eq. B-11 and rearranging terms gives the equation of interest:

$$K_T = \frac{2\pi r_i^2 r_o l}{r_o - r_i} G(r_i) \tag{4}$$

4) Derivation for the equation for the rotational flexibility of the inner cylinder relative to the rotational flexibility of the test soil during the initial cycle of loading, F_i.

We defined the rotational flexibility of the inner cylinder relative to the rotational flexibility of the test soil during the initial cycle of loading, F_i, as:

$$F_i = \frac{\Theta_i}{\Theta_{si}} \tag{B-12}$$

where Θ_i = static twist of active portion of flexible inner cylinder due to static torque, T, acting throughout cylinder
Θ_{si} = static rotation of rigid inner cylinder embedded in undegraded test soil due to static torque, T

To determine Θ_i, we considered the following expression (7), giving the static twist between the ends of a circular shaft, $\bar{\Theta}$, caused by a static torque, $\bar{T}$, acting throughout the shaft:

$$\bar{\Theta} = \frac{\bar{T}\bar{L}}{\bar{G}.\bar{T}} \tag{B-13}$$

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where $\bar{G}$ = shear modulus of shaft
         $\bar{L}$ = length of shaft
         $\bar{J}$ = polar moment of inertia of cross section of shaft

For a hollow, circular shaft, (7),

$$\bar{J} = \frac{\pi}{32} (\bar{D}_o^4 - \bar{D}_i^4) \tag{B-14}$$

where $\bar{D}_o$ = outer diameter of shaft
         $\bar{D}_i$ = inner diameter of shaft

Substituting Eq. B-14 into Eq. B-13 and writing the resulting expression in terms of the variables used in the theoretical feasibility study, an expression may be obtained for the static twist between the ends of the active portion of the flexible inner cylinder caused by a static torque, T, acting throughout the inner cylinder:

$$\Theta_i = \frac{Tl}{\frac{\pi}{32} (D_o^4 - D_i^4) G_i} \tag{B-15}$$

where G_i = shear modulus of inner cylinder
         l = active length of inner cylinder
         D_o = outer diameter of inner cylinder
         D_i = inner diameter of inner cylinder

To determine Θ_{si}, we assumed the static torque, T, to be applied to a rigid inner cylinder embedded in the undegraded test soil. The shear strain developed in the undegraded test soil along the wall of the inner cylinder, $\gamma_r(r_i)$, as a result of the applied torque, T, may be estimated using the following relationship based on hyperbolic shear stress-strain relationships (12):

$$\gamma_r(r_i) = - \frac{\tau_r(r_i)}{\frac{G_{mo}}{\tau_{mo}} \tau_r(r_i) - G_{mo}} \tag{B-16}$$

where $\tau_r(r_i)$ = shear stress in test soil along wall of inner
                    cylinder caused by applied torque T
         G_{mo} = undegraded low amplitude shear modulus
         τ_{mo} = undegraded shear strength

The shear stress, $\tau_r(r_i)$, induced in the test soil as a result of the applied torque, may be estimated using Eq. B-3. Equation B-16 is valid only for positive shear strains. A similar equation is used for negative

B-5

shear strains.

Using Eq. B-9, the static rotation of the rigid inner cylinder, Θ_{si}, resulting from the applied torque, T, may be estimated in terms of the shear strain developed in the test soil along the wall of the inner cylinder, $\gamma_r(r_i)$, as:

$$\Theta_{si} = -(\frac{r_o - r_i}{r_o}) \gamma_r(r_i) \qquad (B-17)$$

Appropriately combining Eq. B-3, Eq. B-16, and Eq. B-17, we may obtain the following expression for the static rotation of the rigid inner cylinder embedded in undegraded test soil due to torque T:

$$\Theta_{si} = -(\frac{r_o - r_i}{r_o}) \left[\frac{T/2\pi r_i^2 l}{(G_{mo}/\tau_{mo})(-T/2\pi r_i^2 l) - G_{mo}} \right] \qquad (B-18)$$

Equation B-15 and Eq. B-18 may be substituted into Eq. B-12 giving the desired expression for F_i:

$$F_i = \frac{Tl/[\frac{\pi}{32}(D_o^4 - D_i^4) \, G_i]}{-(\frac{r_o - r_i}{r_o}) \left[\frac{T/2\pi r_i^2 l}{(G_{mo}/\tau_{mo})(-T/2\pi r_i^2 l) - G_{mo}} \right]} \qquad (11)$$

5) Derivation for the equation for the rotational stiffness of the inner cylinder relative to the rotational stiffness of the test soil during the ultimate cycle of loading, S_u.

We defined the rotational stiffness of the inner cylinder relative to the rotational stiffness of the test soil during the ultimate cycle of loading, S_u, when the test soil has either liquefied or stabilized at some limiting cyclic deformation as:

$$S_u = \frac{\Theta_{suA}}{\Theta_i} \qquad (B-19)$$

where Θ_i = static twist of active portion of flexible inner cylinder due to torque, T, acting throughout cylinder
Θ_{suA} = amplitude of low frequency (inertia and viscous damping forces negligible), cyclic rotation of rigid inner cylinder embedded in fully degraded test soil caused by low frequency, cyclic torque having amplitude, T.

The quantity Θ_i may be determined using Eq. B-15. To determine Θ_{suA}, we assumed that the cyclic torque having an amplitude T was sufficient to induce either liquefaction in loose soils or limiting deformations in dense soils. Using Eq. B-9, we may write the following expression giving Θ_{suA} in terms of the peak shear strain developed in the test soil along the wall of the inner cylinder during the ultimate cycle of loading, $\gamma_{ruA}(r_i)$, when the soil has fully degraded:

$$\Theta_{suA} = -(\frac{r_o - r_i}{r_o}) [\gamma_{ruA}(r_i)] \tag{B-20}$$

Substituting Eq. B-15 and Eq. B-20 into Eq. B-19, gives the desired expression for S_u:

$$S_u = -\frac{(\frac{r_o - r_i}{r_o})[\gamma_{ruA}(r_i)]}{Tl/[\frac{\pi}{32} (D_o^4 - D_i^4) G_i]} \tag{12}$$

<u>Appendix C - Report on Operational Feasibility</u>
<u>Study from Sweet & Aiken</u>

     In this appendix, we present the entire report on the operational
feasibility study from Sweet & Aiken, Inc. The report includes a presentation
and discussion of the operational aspects of the proposed testing system,
an engineering drawing presenting a preliminary design of a laboratory
research prototype testing system, and an appendix which includes discussions
of electronic equipment and provides appropriate literature from
manufacturers.

PRELIMINARY DESIGN OF A PROTOTYPE IN SITU TESTING TOOL

Prepared For
DYNAMIC IN SITU GEOTECHNICAL TESTING
Houston, Texas

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PRELIMINARY DESIGN OF A PROTOTYPE IN SITU SOIL TESTING TOOL

1.0 INTRODUCTION

A prototype tool has been designed for in situ measurement of
cyclic and dynamic soil properties. The final field tool is
intended to be used to test soil on dry land or on the ocean
floor. The prototype has most of the important features of a
field tool, but is intended to be used primarily for
laboratory testing.

This report summarizes the tool. A discussion of mechanical
aspects and explanations of modifications and special
features are included in this report. A discussion of
electronic systems is provided in two reports from an
independent consultant, included in the Appendix.

The preliminary design criteria for the tool are presented in
Table 1.1, and were taken from documents provided by Dynamic
In Situ Geotechnical Testing, Inc.

In operation the tool will be pushed into the soil to be
tested. The test specimen will be an annular volume of soil
captured between an inner barrel and an outer barrel. The
outer barrel will remain stationary with the surrounding soil
and the inner barrel will be rotated in various fashions.

Excitation and Measurement Requirements

Mode	Torque Amplitude Range (ft-lb)	Excitation			Rotation Amplitude Range (deg)		Rotation Frequency (cps)	Acceleration Amplitude Range @ 1 inch radius (g)	Maximum Power (hp)
		Form	Period (sec)	Duration	pk to pk	single pk			
Torque-Controlled Cyclic Test	0.1 - 15	sine wave	0.1 - 50	≤100 cycles	0.002-30	0.001-15	0.02-10	N/A	0.20
Rotation-Controlled Cyclic Test	0.1 - 15	sine wave	0.1 - 50	≤100 cycles	0.4-3.0	0.2-1.5	0.02-10	N/A	0.03
Impulse Test	0.01 - 20	rectangular impulse	N/A	≥0.0002 s	N/A		400-2400	±0.3 - ±210	0.20

Mechanical Requirements

	Dimensions					Penetration Forces			Maximum Differential Lateral Pressure (psf)
	Outer Diameter (in)	Wall Thickness (in)	Length Below Piston (in)	Active Length (in)	Shielded Length (in)	Bearing (lb)	Skin Friction (lb)	Total (lb)	
Outer Barrel	3	1/12	12	N/A	N/A	4500	9500	14,000	30,000
Inner Barrel	1	1/16	12	8	4	1500	2500	4000	30,000

Maximum Environmental Fluid Pressure: 100,000 psf

Range of Differential Vertical
Pressure to Be Applied by Piston: C - 6000 psf

Table 1.1: Preliminary Design Criteria
(Includes All Revisions)

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The tool was designed for testing in two modes:

1. Impulse Testing

2. Cyclic Testing.

For the impulse mode, a step torque will be input to the inner barrel. The torque will be held constant for a fraction of a second and rapidly released. The inner barrel will be allowed to freely vibrate in response to the impulse. The tool will measure the input torque and the rotational response of the inner barrel.

Cyclic testing can be done in two different ways. In one way a cyclic rotation can be input to the inner barrel. The response will be the cyclic torque necessary to generate this motion. In the other way, a cyclic torque will be input to the inner barrel and the response will be the resulting cyclic rotation. Torques and rotational motions will be measured when testing in either way.

The limits of penetration forces, environmental pressure, applied pressure, torque, rotation, power, acceleration, time duration, and frequency are presented in Table 1.1. Manufacturer's literature on all available components are presented in the Appendix. Calculations are provided in Section 7.0.

2.0 DESCRIPTION OF TOOL

Figure 2.1 is a schematic diagram of the drive and measurement systems while Drawing 850227-10 shows a full scale assembly drawing of the prototype in situ soil testing tool. The overall length excluding a latch/penetration device is 44 inches, and the outer diameter is 3 inches. These dimensions are such that the tool will fit inside a standard coring bit for testing ocean floor soil.

The soil test specimen is contained between the outer barrel and the inner barrel. Its dimensions are

    Outer Diameter - 2.833 inches

    Inner Diameter - 1.000 inch

    Length   - 8.000 inches.

The tool captures an additional 4 inches of soil above the test specimen. This soil is expected to be disturbed and is not intended to be tested; it is isolated from the rotating inner barrel by a shield which remains stationary and does not touch the rotating inner barrel.

The tool simulates the overburden effect of the soil removed from the borehole by applying a force to the top of the soil specimen. The force is applied hydraulically through the vertical pressure piston. The pressure will be measured by a pressure transducer.
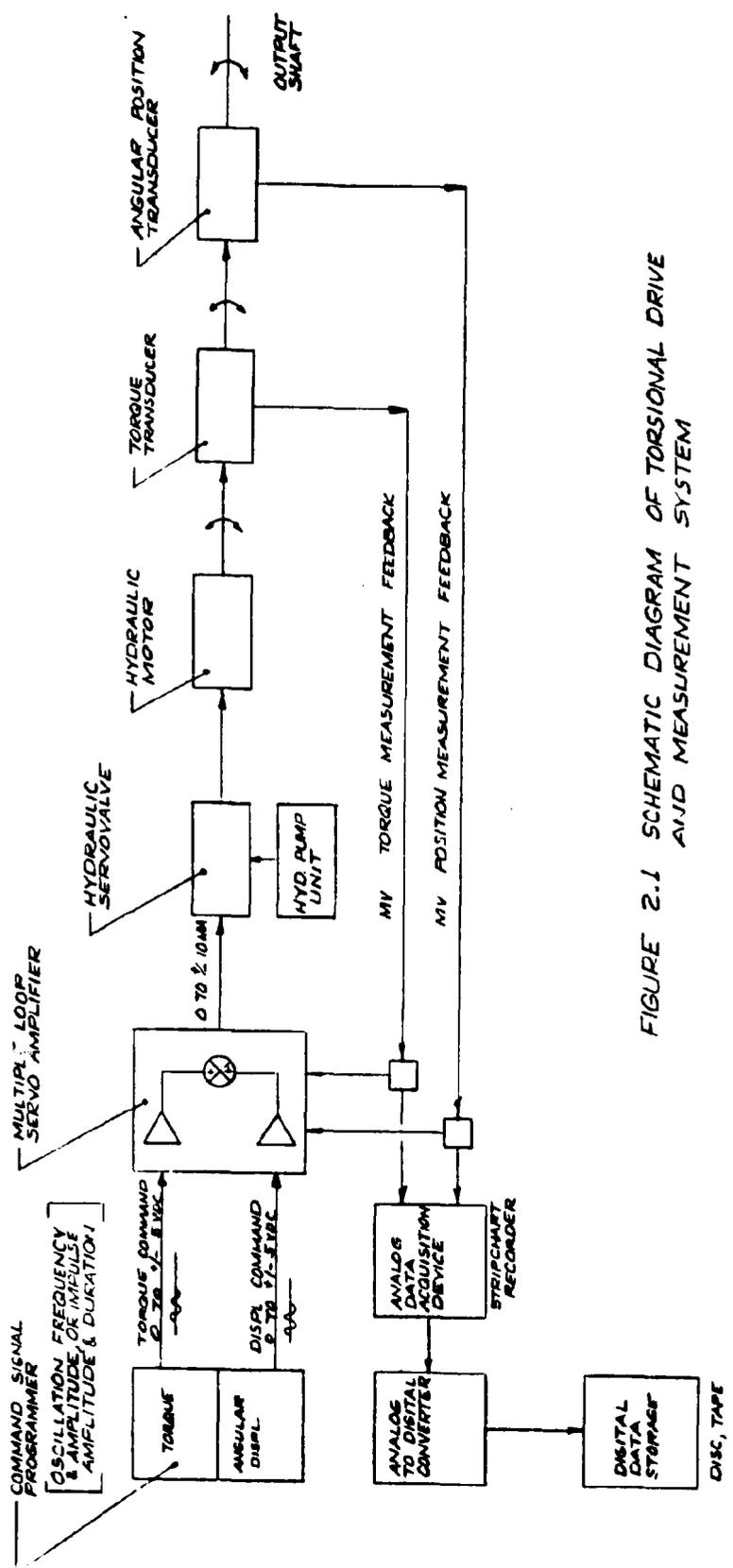
FIGURE 2.1 SCHEMATIC DIAGRAM OF TORSIONAL DRIVE AND MEASUREMENT SYSTEM

The lower ends of the inner and outer barrels are designed
with tapers that drive the displaced soil away from the test
specimen. The soil displaced by the outer barrel is forced
into the surrounding soil; the soil displaced by the hollow
inner barrel is forced inside the inner barrel. Jutted edges
are provided in this barrel to force soil away from the wall
of this barrel. The volume of space inside the inner barrel
and the soil holding chamber was designed to hold all soil
displaced by the inner barrel and shield. The various
surfaces of the barrels will be treated for different
characteristics. Relief ports are provided for soil chambers
to prevent porewater pressure build-up during penetration.

The force necessary to drive the inner barrel into the soil
is larger than the tranducers and bearings can support.
Thus, the inner barrel driving force is transferred to the
shield through a load shoulder. Then it is transferred to
the outer barrel through 3 pins where it combines with the
outer barrel driving force and is transferred to the coring
tool.

It is probable that the inner barrel will be transversely
loaded during the driving operation. This may induce
excessive shear and bending on the inner barrel and
tranducers, unless lateral support is provided. The tool was
designed with two lateral support shoulders: one at the
lower end of the shield and one at the upper end of the

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shield. These support shoulders are designed so that reasonable bending loads will not damage the tool.

The lateral supports can induce frictional resistance to rotation that might be of the same order of magnitude as the input torque. Since torque is measured above the supports, the support induced torque would contaminate the test results. To alleviate this problem the shield was designed so that it could be moved axially upward after the driving operation, eliminating all contact with the inner barrel.

Excessive transverse forces developed during actual testing may cause the inner barrel to contact the shield. This will prevent the completion of the test; however, it should also prevent damage to the transducers due to excess bending. This is not expected to occur under normal conditions.

The only component that will touch the inner barrel below the transducers is a lip seal between the inner barrel and outer barrel. This seal is necessary to keep the in situ fluid away from the transducers and bearings. The seal used is a Parker U packing seal designed to reduce contact forces to a minimum. The pressure differential across the seal will be kept below 10 psi. This low pressure, along with the low friction features of the seal, should reduce signal contamination by seal contact to less than 0.15 inch lbs.

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One important requirement of the oscillating components for testing in the impulse mode is that they be underdamped. Underdamped behavior was promoted by limiting contact with stationary parts to a bare minimum and enclosing all but the inner barrel in air instead of oil. During testing the only contacts between moving and stationary parts are in the hydraulic motor, at the two tapered roller bearings, and at the lip seal.

Measurement tranducers are located between the inner barrel and the drive shaft. These transducers measure inner barrel rotation, torque on the inner barrel, and rotational acceleration of the inner barrel.

Based on the search conducted for the preliminary design, it was concluded that the torque range requirements are too large for a single torque transducer. Thus, for the prototype system, two interchangeable transducers will be used: a low range torque transducer and a high range torque transducer. Small torques will be measured using a Lebow Model 2127 torque transducer, and large torques will be measured using a Lebow Model 2102 torque transducer. A second Lebow Model 2127, with a lower capacity, could be used for accuracy in the lowest torque range. Separate tools would be considered for field use.

Angular acceleration will be measured using two PCB

Piezotronics model 303A quartz accelerometers. The two accelerometers are mounted on the inner barrel 180 degrees apart at a known radial distance from the centerline. They measure linear acceleration which is converted to rotational acceleration using the kinematic relationships between rectilinear and angular motion. Using two accelerometers avoids imbalance and more accurately provides rotational motion.

Angular rotation will be measured using a Trans-Tek Series 604 angular displacement transducer. The housing of the transducer will be connected to the outer barrel and the transducer shaft will be connected to the main shaft.

Inner barrel excitation is provided by a miniature orbit type hydraulic motor manufactured by Lamina. This motor will run with equal torque in either direction and is instantly reversible. Its housing is sleeved into the outer barrel and its shaft is sleeved into the drive shaft. Based on the search conducted for the preliminary design, it was concluded that, just as with the torque transducers, the torque range requirements may be too large for a single motor. Thus, for the laboratory prototype, two interchangeable motors may be required. For the field, two tools may be required. The small torques will be provided by a Model A-25F motor, and the large torques will be provided by a Model A-50F motor.

The torque or rotation output of the motor is to be
controlled by a Moog type 30 Series 30 servovalve. The
position of the valve stem will be controlled by a signal
from the torque transducer or angular rotation transducer. It
will compare the signal with the desired value. If the
signal is too low, the valve will open and if it is too high,
the valve will close. The test operator will control the
servovalve through a Moog series F120 rack mounted
electronics package.

3.0 DESCRIPTION OF COMPONENTS

3.1 Outer Barrel

The outer barrel serves as the housing for the test specimen
and all internal components of the soil testing tool. The
removable lower end of the outer barrel contains the test
specimen. The upper portion contains the internal components
and will connect, in the field model, to the coring bit or
thread into a pipe.

The outer barrel has a uniform diameter of 3 inches, is 44
inches long, and is made of type 410 stainless steel. The
lower portion has a nominal wall thickness of 1/12 inch and
is 12 inches long.

Type 410 is a general purpose stainless steel that has good

corrosion resistance, is strong enough to resist penetration
forces, is readily available, and is low cost compared to
most other stainless steels. The upper end is made of
annealed 410 having a minimum tensile strength of 75 ksi and
a minimum yield strength of 35 ksi. The lower end is made of
quenched and tempered type 410 having a minimum tensile
strength of 110 ksi and a minimum yield strength of 85 ksi.

The lower end, containing the soil specimen, will be
hydraulically pushed at a controlled rate into the soil
formation. It has special inside and outside surface
treatments so that disturbances are minimized and it properly
interacts with the surrounding soil and the test specimen.

It is necessary that the outside surface provides reasonably
low axial resistance with the soil to reduce the penetration
force. More importantly, it must have high rotational
resistance so that it helps transfer the driver reaction
torque to the surrounding soil. These characteristics
are developed by machining axial grooves all around the outer
circumference of the lower end. Cylinders with different
groove dimensions would be used for different soils.

The inner surface must be as frictionless as possible in the
axial direction. This minimizes disturbances to the test
specimen. As with the outer surface, this surface must have
a high rotational resistance to minimize slip between the

outer barrel and the soil specimen. These characteristics
are achieved by machining axial grooves along the inside of
the lower end and coating the surface with a
polytetrafluoroethylene (PTFE) coating. This will produce a
surface coefficient of friction of less than 0.090.

PTFE is a fluoroplastic having excellent chemical inertness,
good high and low temperature stability, and very low
friction. The static coefficient of friction decreases with
increasing load. Thus, PTFE bearing surfaces do not seize,
even under extremely high loads. PTFE is relatively soft and
is not resistant to wear unless improved by compounding the
resins with inorganic fibers or particulate materials. Most
PTFE coating companies include proprietary additives to
improve wear characteristics. These improved PTFE coatings
will allow numerous in situ soil tests before wear makes them
nonfunctional. When this occurs, the barrels can be returned
to the coating company for recoating.

The lower end of the outer barrel is provided with a taper so
that displaced soil will be pushed into the surrounding soil
instead of into the test soil.

A fluid relief port is provided at the top of the lower end
of the outer barrel so that entrapped fluid will displace
during penetration. A filter stone will be provided in the
port to prevent the escape of soil.

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3.2 Inner Barrel

The inner barrel is the lowermost portion of the oscillating components. It forms the inside of the soil test specimen and transfers the input torques and rotations to the specimen. It has a 1 inch outer diameter at the lower end, a 1.8 inch diameter at the upper end, and is about 18 inches long. The lower end adjacent to the soil test specimen has a wall thickness of 1/16 inch and is threaded into the upper section for easy removal for repairing or recoating.

The inside of the inner barrel is hollow to reduce the amount of soil displaced during penetration. The lower end has a taper to drive the displaced soil inside the bore instead of outside to the test specimen. The top of this taper is jutted to minimize contact between the tool surface and the soil inside the bore. A second jutted taper is provided near the base of the shield where the inner cylinder is constricted. The top end has a larger bore to provide room for the soil displaced by its wall. A relief port is provided at the uppermost part of the inner barrel wall to allow entrapped fluid to escape during penetration. A filter stone will be provided in the port to prevent the escape of soil.

The outside surface of the inner barrel must have the same

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characteristics as the inside surface of the outer barrel;
thus, it has the same surface treatments. The outside
surface is axially grooved and coated with PTFE resin so that
it will properly transfer driver actions to the test
specimen, but will not excessively disturb the test specimen
during penetration. The inside surface is to be machined to
a 16 RMS finish and coated with PTFE resin so that
interaction with the entrapped bore soil is minimized.
Durability characteristics described for the outer barrel
coatings also apply to the inner barrel coatings.

The inner barrel is made of quenched and tempered type 410
stainless steel having a minimum tensile strength of 110 ksi
and a minimum yield strength of 85 ksi. This material has
adequate strength to resist the penetration force, is hard
enough to resist damage from soil contact, and is corrosion
resistant.

3.3 Vertical Pressure System

A hydraulic piston is provided in the top of the annular
space between the inner and outer barrels. This piston may
be pressurized after penetration to simulate the vertical in
situ force removed by the coring operation.

The outside diameter is 2.833 inches, the inside diameter is
1.0 inch, and the length is 0.75 inch. The piston is made of

annealed type 410 stainless steel for its corrosion resistance.

The pressure area of the piston is the same as the cross sectional area of the soil specimen; thus, the applied soil pressure is approximately the same as the hydraulic pressure applied to the piston. The pressure is monitored by a pressure transducer located at the top of the tool. This transducer is described in Section 3.5 of this report.

The vertical pressure system is designed either to provide a constant pressure on the specimen regardless of the soil movement or to remain stationary during testing. A valve is provided to control the vertical pressure system mode. Thus, in the constant pressure mode, with the valve open, if the soil specimen contracts during testing, the piston is compliant and will move down so that the pressure remains constant. In the stationary mode, after pressurizing the piston, the valve is closed so that the piston remains stationary regardless of the soil movement.

3.4 Shield

A 12 inch long soil specimen is trapped between the inner and outer barrels. Of this, the lower 8 inches is considered the test specimen and the top 4 inches is probably disturbed and not intended to be tested. A stationary shield is provided

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between the oscillating inner cylinder and the top 4 inches of the soil specimen. This shield prevents the inner cylinder movements from being transferred to the inner surface of the disturbed part of the soil specimen.

Another function of the shield is to react the penetration forces and transmit them directly to the outer barrel so that they do not damage the transducers or bearings. The axial penetration force on the lower end of the inner barrel is transferred to a bearing shoulder located on the lowermost end of the shield. It passes axially up the shield to the three radial pins connecting the shield and outer barrel.

It is very likely that transverse forces will act on the inner barrel during penetration due to nonuniform soil properties. This causes an end moment as well as transverse shear forces. These actions are transferred to the shield by circumferential bearing areas located at the bottom of the shield and near the top of the shield. Thus, reasonable transverse forces may be sustained.

Since the input torques may be very small, it is necessary that contact between the shield and inner barrel be eliminated during testing. This is achieved by applying 400 psi hydraulic pressure to the bottom face of the shield flange. The pressure forces the ~~two~~ three (all) connecting pins out of engagement on the shield and slowly drives the shield upward

3/32 inch where it contacts a landing surface on the inner barrel. This axial movement disengages the shield bearing surfaces from contact with the inner barrel leaving enough clearance between the shield and inner cylinder to prevent contact under normal conditions. The shield can be reengaged in two ways. It can be moved back into position by hand after disassembling the tool. Alternatively, a second hydraulic line can be provided to apply pressure to the upper face of the shield flange so that the shield can be driven downward until the connecting pins reengage. The second method would be preferred in a field tool.

3.5 Transducers

The tool incorporates three types of transducers to measure actions and motions input to the inner barrel. These include an angular displacement transducer, a torque transducer, and two accelerometers. The angular displacement transducer will mainly provide measurements during low frequency cyclic tests; whereas, the accelerometers will provide measurements during impulse tests. The torque transducer will provide measurements during both tests.

A Trans-Tek series 604 angular displacement transducer is attached to the upper end of the inner barrel. It measures the angular position of the inner barrel relative to the outer barrel and converts it into a DC voltage directly

proportional in amplitude and polarity to displacement from
the electrical null position. It is a precision differential
capacitor device and has integral voltage regulation,
oscillator, demodulator and output buffer. The DC output can
be fed directly into voltmeters, recorders, A/D converters or
other devices. The standardized output allows interchanging
units without system recalibration. A locking wedge is
provided between the inner barrel and shield at the upper
support surface. The locking wedge prevents rotation of the
inner cylinder and shaft to prevent deviations from zeros
during running and penetration. The lock is disengaged when
the shield is moved out of contact with the inner barrel.

Standard series 604 transducers are designed to measure a 60
degree rotation range. This is much larger than required for
the in situ soil testing tool which requires a maximum peak
to peak rotation range of 30 degrees and a minimum range of
0.002 degree. Trans-Tek specifies factory calibration to any
specified output sensitivity and Trans-Tek technical experts
are certain the series 604 can be calibrated to accurately
read rotations down to 0.01 degree with a resolution of 0.001
degree. Accuracy below 0.01 degree is questionable. The
range would be ±0.5 degree. As indicated in a letter from
Trans-Tek included in the Appendix, they are willing to make
the necessary modifications at a reasonable cost.

The full range of rotations can be measured with a single

tool using two angular displacement transducers in series.
One would be calibrated for the large rotations and the other
for the smaller rotations. All but the very smallest
rotations (less than 0.01 degree) would be accurately
measured with this system. The full scale assembly drawing
shows only a single angular displacement transducer. The
Trans-Tek series 604 meets all other requirements including
pressure and temperature requirements.

The shaft of the angular displacement transducer fits into a
precision hole in the upper end of the inner barrel. It is
held in place with a small set screw. The body is attached
to the outer barrel with three mounting brackets bolted to
the three tapped holes in the bottom face of the transducer.
These brackets pass through slots in the shaft coupler to the
outer barrel.

The two accelerometers are PCB Piezotronics model 303A. They
are quartz sensors having built in electronics so they can
operate over a coaxial or two conductor cable; one lead
conducts both signal and power.

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The two accelerometers are mounted 180 degrees apart on angle
brackets attached to the shaft coupler just above the angular
displacement transducer. They are oriented to measure
tangential acceleration at a radius of 0.75 inch. These
transducers meet all requirements, including pressure and

temperature requirements.

The torque transducer is mounted between the motor shaft and the angular displacement transducer. A Lebow model 2127 with a 50 in-oz capacity will be used to measure torques less than 0.26 ft-lbs and a Lebow model model 2102-200 will be used to measure torques between 0.26 and about 17 ft-lbs. If necessary, a Lebow model 2127 with a 10 in-oz capacity can be used to measure very small torques more accurately. The upper limit is somewhat less than specified by the client, but was judged acceptable by the client. These are all in-line reaction torque devices using foil type strain gages to measure torsional shear strains. They produce minimal friction error.

The model 2127 has a flange at one end with 3 holes for boltup. This end will attach to the shaft coupler. The other end has a shaft that will fit into a mating hole in the lower end of the drive shaft. It will be held rigidly to the drive shaft with a small set screw.

The higher torque transducer has different end fittings. Both ends of this transducer have four drilled and tapped holes for flanges. For the prototype tool, a flange adapter will be necessary to make the high torque transducer interchangeable with the low torque transducer.

Both torque transducers have lead terminal boxes extending
radially outward. Since these boxes will interfere with the
outer barrel they will be removed and pigtail lead wires
will be draped out of the main body. The torque transducers
satisfy all requirements including temperature and pressure
requirements.

The tool will also include three pressure transducers. These
will be used to measure the differential pressure between the
vertical pressure piston and ambient, between the
transducer/bearing chamber and ambient, and between the space
under the shield flange and the ambient. If the upper face
of the shield flange is pressurized, a fourth pressure
transducer will be used. All transducers will be Entran
model EPX101 strain gage sensors that can be specified with a
pressure range from 5 to 5000 psi. These pressure tranducers
meet all requirements. They will be threaded into a mounting
plate attached to the inner barrel above the driver.

3.6 Drive Shaft

The drive shaft connects the driver to the transducers. It
is guided by two tapered roller bearings. It is about 7
inches long and has a maximum diameter of 1.25 inches. It is
made out of annealed type 410 stainless steel having a
minimum ultimate strength of 75 ksi and a minimum yield
strength of 35 ksi. The shaft is very lowly stressed during

operation; thus the material was selected solely for availablity, cost, and corrosion resistance.

The lower end of the shaft has a precision drilled hole for the torque transducer shaft and the upper end has a precision drilled hole for the motor shaft. Both ends have radial drilled and tapped holes for set screws to hold in the mating shafts.

3.7 Drive Shaft Coupler

The drive shaft coupler is used to connect the inner barrel to the torque transducer. The lower end is provided with a female thread which connects with a mating male thread on the top end of the inner barrel. A set screw will prevent unthreading during tests. The top end has drilled and tapped holes for bolts which connect it to the torque transducer and accelerometers.

The drive shaft coupler is made out of annealed type 410 stainless steel. This material was chosen for its availability, cost, and corrosion resistance.

3.8 BEARINGS

The bearings supporting the shaft that drives the torque and angular displacement transducers are single row, straight

bore Timken precision bearings, identified by the number
07100-07196.

The approximate 4.5 inch spacing of the bearings in the
"indirect" installation mode provides high shaft stability
and accurate alignment with the axis of the tool. The
tapered roller bearing elements not only minimize shaft axial
movement, but also are ideal for oscillatory service of the
type anticipated for the tool. Each bearing has a
coefficient of friction of approximately 0.00018. The
torsional resistance of the bearings will not contaminate the
torque measurements of the tool since the torque transducer
is between the soil test specimen and the bearings.

The very low radial and thrust loads that will be imposed on
the bearings in the anticipated application of the tool
should insure long bearing life provided lubrication is
adequate and the operating environment is clean.

3.9 SEALS

The single dynamic seal isolating the transducers,
accelerometers, motor, servovalve, and other components from
the environment in which the tool is operating is a Parker
8500 series U-ring. Parker's identifying number is 8506-
0175.

The very low friction characteristics of the Parker seal
result from its cross sectional design and from the seal
material. The cross section is a nominal "U" shape, but
actually resembles a truncated "Y". This shape provides
almost knife edge contacts between the sealing edges and the
sealing surfaces. The seal material is a very low friction
polyurethane compound having optimum wear and abrasion
resistance.

As stated previously, the pressure differential across the
seal will be kept below 10 psi. The upper chamber of the
tool will be pressurized with air to 10 psi above ambient
using a standard air line and surface control unit. The
pressure will be monitored with a pressure transducer
described in Section 3.5.

All other seals are conventional "O" ring seals used as
static or gasket type seals or as piston seals. There is
nothing unique or unusual about these "O"-ring type seals.

3.10 Hydraulic Motor

As stated previously in Section 2.0, excitation of the inner
barrel comes from a miniature, high torque, medium speed
orbit type hydraulic motor built by Lamina under license from
Char-Lynn, a division of Eaton Corporation. Also as stated
previously, two sizes of this motor will be required to

handle the range of torque requirements: the A-25F model
for the lower torque range and the A-50F model for the higher
torque range.

These motors will run in each direction with equal torque and
are instantly reversible. Maximum torque output is 100 inch
pounds for the A-25F model and 200 inch pounds for the A-50F
model. As with the torque transducers, this maximum torque
is slightly less than that specified by the client, but was
judged to be adequate by the client. Otherwise the motors
meet all requirements including temperature and pressure
requirements.

Mounting in the prototype tool will require some
modification, including turning down the mounting flange
diameter, turning down the shaft diameter, and changing the
hydraulic ports as illustrated in the assembly drawing.

3.11 Hydraulic Controller

A Moog standard series 30 servovalve provides control of
torque output and of rotation. The torque transducer sends a
signal to the servovalve, which will compare the signal with
the desired torque value and make the required adjustment.
Similarly, the angular displacement or rotation transducer
will send a signal to the servovalve giving the measured
angular displacement for comparison with the desired value

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and correction if required. The frequency response of the
servovalve is satisfactory for low frequency cyclic tests.
During impulse tests, in which an impulsive rectangular
torque will be applied, an accumulator will be used to insure
constant pressure, and thus, constant torque. This is needed
because the servovalve could not follow the high frequency
fluctuations that would otherwise develop.

The servovalve does not have as rapid a rise time as desired
by the client. As indicated in the manufacturer's
specifications the approximate time to develop 90% output in
response to a step input of current is 0.0025 second. The
client has accepted this limitation. Additionally, the
servovalve does not appear to satisfy environmental pressure
requirements. Moog provides special designs for high
pressure applications and such a design may be required.
Alternatively, in the field unit, the controller may be
placed in the low pressure environment of the downhold
electronics package (see Appendix for discussion of
electronics package). The standard servovalve will satisfy
all other requirements including temperature requirements.
The servovalve can be mounted on a plate in the interior of
the tool as indicated in the assembly drawing.

The servovalve can be controlled by the operator using a Moog
series F120 rack mounted electronics package. The package
will consist of an F120-101 modular cage with the following

electronic cards:

F128-201 Power Supply

F122-202 Servo Amplifier

F123-202 Test Module

F123-206 Signal Generator.

3.12 Hydraulic Drive Schematic

Figure 3.1 shows a schematic layout of the hydraulic drive
system for operation of the tool. This schematic diagram
illustrates the functions of the various components and how
they relate to each other.

For the laboratory prototype, the Moog servovalve and the
Lamina motor will be housed in the tool. All other hydraulic
components will be outside the tool for convenience. For the
field tool, the accumulator will also be housed in the tool.
Including the accumulator in the tool is feasible. The
remaining hydraulic components are standard and pose no
feasibility problems for remote operation over 1500 of cable.

3.13 Power and Support Cable

It will be necessary to support the field tool from the
surface with a cable, and to run electrical wires, hydraulic
lines and a pneumatic line from the surface to the tool. The

Legend

1. OIL RESERVOIR
2. PUMP & ELECTRIC MOTOR
3. RELIEF VALVE
4. PUMP UNLOADING VALVE
5. ACCUMULATOR
6. FLOW CONTROL VALVE
7. PRESSURE FILTER
8. HOSE
9. QUICK DISCONNECTS
10. SERVOVALVE, MOOG SERIES 30
11. MINIATURE HYDRAULIC MOTOR
12. DISPLACEMENT TRANSDUCER (ANGULAR)
13. TORQUE TRANSDUCER

OUTSIDE TOOL / INSIDE TOOL

RETURN
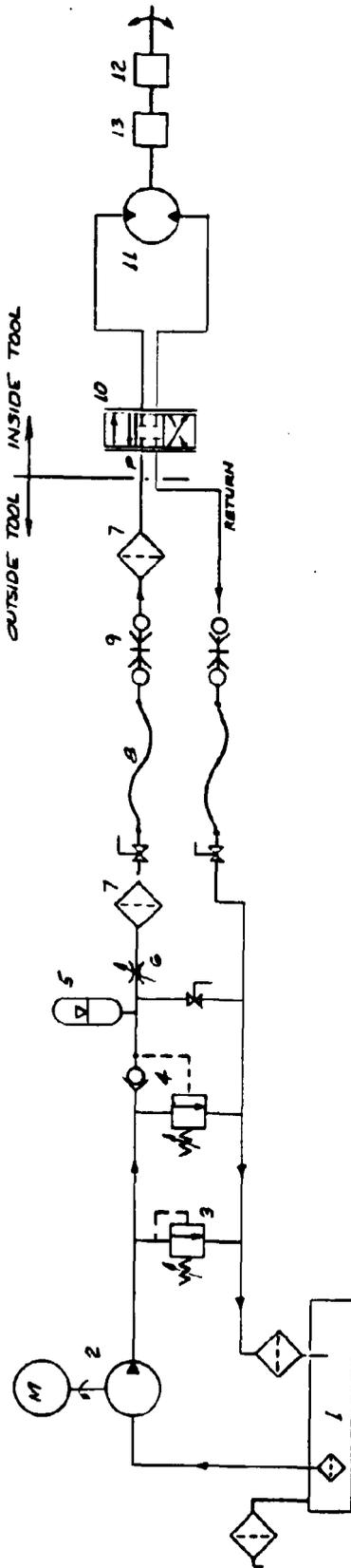
FIG. 3.1 HYDRAULIC CIRCUIT DIAGRAM FOR TORSIONAL DRIVE SYSTEM

field tool will use a single umbilical with all the wires, hoses, and support cable contained within an overall jacket. There are several local cable companies that routinely manufacture custom cables like this.

For the laboratory prototype a hand made umbilical will be formed by strapping or taping the hoses and electrical wires together.

Electrical noise is a source of potential problem especially while taking small readings when the electrical signals will be very small. It will be necessary to shield all sensor cables and eliminate any sources of electrical noise while testing.

3.14 Latching and Penetration Systems

Based on my experiences, I believe that a drill string latching unit and a controlled-rate penetration system for a field tool will involve only current technology, and thus, should not present any feasibility problems.

3.15 Electronic Systems

Required electronic systems were established by Lionel Milberger, a consultant. His work is discussed in two brief reports given in the Appendix. Cost estimates are included.

4.0 POTENTIAL PROBLEMS

The major source of potential problems is the very small
rotations required to be measured by the tool. Based on the
study, it was concluded that probably the smallest angular
displacement that can accurately be measured with the tool is
0.01 degrees; thus, it seems that the requirements for the
torque-controlled cyclic tests cannot be fully satisfied.
Rectilinear capacitance transducers and LVDT devices were
considered but rejected because they are extremely sensitive
to electrical noise and temperature effects. Optical
encoders were also considered. A standard optical encoder
was not found that would meet the criteria of the in situ
test tool. However, an optical encoder could probably be
designed that would meet the criteria. This is a very
definite possibility for the field units.

5.0 POTENTIAL IMPROVEMENTS

The most important improvement would be replacement of the
angular displacement transducer with a transducer that could
measure smaller angles. Optical encoders are devices
currently under development that might do this. These are
relatively small displacement transducers that measure light
passing through a perforated target. As the measured
displacement changes more or fewer perforations are exposed
or covered. The perforations are calibrated to measure

displacement.

Other developments in control and measurement technology, motivated by the robotics industry, may improve tool design. One example is a miniature servovalve/oscillator unit currently being developed by Moog. This would replace both the servovalve and hydraulic motor incorporated in the prototype design. Manufacturer's information on this unit is provided in the Appendix. It is very possible that a single Moog servovalve/oscillator that meets the criteria of the in situ test tool will be available when the field units are designed.

Another improvement which seems to be feasible will be to modify the vertical pressure piston so that it can be at the bottom of the inner barrel prior to penetration and will apply a load to the soil specimen as the tool penetrates.

6.0 COST ESTIMATE FOR LABORATORY PROTOTYPE TESTING SYSTEM

Detail Design and Drafting........... $16900.00

Machined Components.................. 12500.00

Four Pressure Transducers........... 1860.00

Angular Displacement Transducers

     Small Angles..................... 1500.00

     Large Angles..................... 300.00

Two Accelerometers................... 900.00

Hydraulic Components (Including

     Servovalve, Two Motors,

     Controller, and, Misc. Items).... 13000.00

Accessory Electronic Equipment....... 20400.00

Torque Transducers

     High Torque...................... 1400.00

     Low Torque - 10.................. 2300.00

     Low Torque - 50.................. 2300.00

Two Roller Bearings.................. 100.00

Seals................................ 50.00

Miscellaneous Hardware............... 350.00

Laboratory Penetration System........ 4400.00

Total Cost Estimate.................. $78260.00

7.0 CALCULATIONS

1.0 ANALYSIS OF PENETRATION LOAD STRESSES

1.1 PENETRATION LOADS

F_0 = MAXIMUM PENETRATION FORCE ON
OUTER BARREL   (AXIAL)

F_0 = 14,000 lbs $\Rightarrow$ 4500 lbs FROM BEARING
4500 lbs FROM SKIN FRICT

F_I = MAXIMUM PENETRATION FORCE ON
INNER BARREL   (AXIAL)

F_I = 4000 lbs $\Rightarrow$ 1500 lbs FROM BEARING
2500 lbs FROM SKIN FRICTION

1.2 AXIAL STRESSES IN BARRELS (LOWER END)

S_{AI} = MAXIMUM AXIAL STRESS IN INNER BARREL

$$S_{AI} = \frac{F_I}{A_{MIN}}$$

A_{MIN} = MINIMUM CROSS SECTIONAL AREA

A_{MIN} OCCURS IN UPPER SECTION COINCIDENT TO
SHIELD, WHERE

D_0 = 1.000 in   (OUTSIDE DIA.)

D_1 = 0.875 in

W_T = 0.0625 in   (WALL THICKNESS)

$$A_{MIN} = \frac{\pi}{4}(1.000^2 - .875^2)$$

$$A_{MIN} = 0.1841 \text{ in}^2$$

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$$S_{AI} = \frac{4000}{0.1841}$$

$$S_{AI} = 21,730 \ PSI$$

R = RATIO OF STRESS TO YIELD STRENGTH

$$S_y = 85,000 \ PSI \qquad (\text{YIELD STRENGTH})$$

$$R = \frac{21,730}{85,000}$$

$$R = 0.256 \quad \Rightarrow \quad OK$$

S_{AO} = MAXIMUM AXIAL STRESS IN OUTER BARREL

$$S_{AO} = \frac{F_O}{A_{MIN}}$$

A_{MIN} OCCURS AT LOWER SECTION OF OUTER BARREL WHERE,

$$D_O = 3 \ in$$
$$D_1 = 2.833 \ in$$
$$W_T = 0.0833 \ in$$

$$A_{MIN} = \frac{\pi}{4} (3^2 - 2.833^2)$$

$$A_{MIN} = 0.765 \ in^2$$

$$S_{AO} = \frac{14,000}{0.765}$$

$S_{AO} = 18,299$ PSI

$S_y = 85,000$ PSI

$R = \dfrac{18,299}{85,000}$

$R = 0.2153 \implies$ OK

1.3 STABILITY OF BARRELS

$L_c =$ CRITICAL BUCKLING LENGTH

$L_{cr}^* = \sqrt{\dfrac{\pi^2 E I}{F}} \implies$ CONSERVATIVELY ASSUMES PINNED ENDS

WHERE,

$E =$ YOUNG'S MODULUS $= 29 \times 10^6$ PSI

$I =$ MOMENT OF INERTIA $= \dfrac{\pi}{64}(D_o^4 - D_i^4)$

$F =$ AXIAL FORCE

FOR THE OUTER BARREL

$I = \dfrac{\pi}{64}(3^4 - 2.933^4)$

$I = 0.8141$ in^4

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* PG 49 OF <u>THEORY OF ELASTIC STABILITY</u>, BY S TIMOSHENCO McGRAW-HILL, 2nd EDITION.

C3c

$$L_{CR} = \sqrt{\frac{(\pi)^2(29 \times 10^6)(.8141)}{(14,000)}}$$

$L_{CR} = 129.0$ inches $>> L = 12$ in

FOR INNER BARREL

$$I = \frac{\pi}{64}(1^4 - 0.875^4)$$

$$I = 0.02031 \text{ in}^4$$

$$L_{CR} = \sqrt{\frac{(\pi)^2(29 \times 10^6)(.02031)}{(4000)}}$$

$L_{CR} = 38.1$ in $> L = 12$ in (INNER BARREL
LENGTH + SHIELD LENGTH)

$\therefore$ BARRELS WILL NOT BUCKLE

1.4   AXIAL STRESS IN SHIELD

S_A = AXIAL STRESS IN SHIELD

$$S_A = \frac{F_I}{A_{MIN}}$$

A_{MIN} = SAME AS INNER BARREL

$A_{MIN} = 0.1941 \text{ in}^2$

$$S_A = \frac{4000}{0.1841}$$

$$S_A = 21,727 \text{ PSI}$$

$$S_y = 85,000 \text{ PSI}$$

$$R = \frac{21,727}{85,000}$$

$$R = 0.256 \implies \text{OK}$$

1.5 BEARING STRESS BETWEEN SHIELD & LOCK PIN

$S_B = $ BEARING STRESS

$$S_B = \frac{F_I}{3 A_P} \implies \text{FACTOR OF 3 FOR 3 PINS}$$

WHERE,

$A_P = $ PROJECTED AREA OF PIN INSERT
INTO SLEEVE FLANGE (DIAMETER $= D_P$)

$$A_P = D_P L$$

$$D_P = 0.200 \text{ in} \qquad L_P = 0.080 \text{ in}$$

$$A_P = (.200)(.080)$$

$$A_P = 0.0160 \text{ in}^2$$

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$$S_B = \frac{4000}{(3)(.0160)}$$

$$S_B = 83,333 \text{ PSI}$$

$$S_y = 85,000 \text{ PSI}$$

$$R = \frac{83,333}{85,000}$$

$$R = 0.980 \implies \text{OK FOR BRG STRESS}$$

1.6 BEARING STRESS OF LOCK PIN ON OUTER BARREL
     INSERT

$S_B =$ BEARING STRESS

$$S_B = \frac{F_I}{3 A_P}$$

$A_P =$ PROJECTED AREA OF PIN WITH
       DIAMETER D_P AND ENGAGEMENT
       LENGTH, L

$D_P = 0.325$ in $L = 0.1$ in

$A_P = D_P L$

$A_P = (.325)(0.1)$

$A_P = 0.0325 \text{ in}^2$

$$S_B = \frac{4000}{(3)(.0325)}$$

$$S_B = 41,026 \text{ PSI}$$

$$S_y = 85,000 \text{ PSI}$$

$$R = \frac{41,026}{85,000}$$

$$R = 0.483 \quad \Rightarrow \quad OK$$

1.8 SHEAR STRESS IN SHIELD LOCK PIN

S_s = SHEAR STRESS IN LOCK PINS

$$S_s = \frac{F_I}{3 A_s} \quad \Rightarrow \quad \text{FACTOR OF 3 FOR} \\ \text{3 PIN/INSERT ASSEMBLIES}$$

A_s = CROSS SECTIONAL AREA OF EACH PIN AT NOSE DIAMETER D_p = 0.200 in

$$A_s = \frac{\pi}{4} D_p^2$$

$$A_s = \left(\frac{\pi}{4}\right)(.200)^2$$

$$A_s = 0.0314 \text{ in}^2$$

$$S_s = \frac{4000}{(3)(0.0314)}$$

$$S_s = 42,463 \text{ PSI}$$

$S_y = 140,000$ PSI

$R = \dfrac{42,463}{140,000}$

$R = 0.303 \quad \Rightarrow \quad$ OK

1.9 AXIAL STRESS IN TOP OF UNTER BARREL
 (ASSUMES ALL PENETRATING FORCE GOES UP
  THRU BARREL)

$S_A =$ AXIAL STRESS

$S_A = \dfrac{F_0 + F_I}{A_{MIN}}$

$A_{MIN} =$ MIN. CROSS SECTIONAL AREA

$A_{MIN} = \dfrac{\pi}{4}(D_0^2 - D_I^2)$

$D_0 = 3$ in $D_I = 2.7$ in

$A_S = \dfrac{\pi}{4}(3^2 - 2.7^2)$

$A_S = 1.343$ in^2

$S_A = \dfrac{4000 + 14,000}{1.343}$

$S_A = 13,403$ PSI

$S_y = 35,000$ PSI

$R = \dfrac{13,403}{35,000} = 0.383 \quad \therefore \quad$ OK.

2.0   ANALYSIS OF PRESSURE TO OPEN SHIELD
      LOCK PINS & RESULTING STRESSES

   2.1   PRESSURE TO OPEN, P_I

$$P_I = \frac{F_0}{A_P}$$

F_0 = FORCE REQUIRED TO COMPRESS RUBBER
       CYLINDER OF DIAMETER D_R AND
       LENGTH L_R A DISTANCE δ

$D_R = 0.200$ in

$L_R = 0.15$ in

$\delta = 0.08$ in

ASSUMING ELASTIC BEHAVIOR,

$$\frac{F_0}{A_R} = E \frac{\delta}{L_R}$$

A_R = CROSS SECTION OF RUBBER

$A_R = \frac{\pi}{4} D_R^2$

$A_R = \left(\frac{\pi}{4}\right)(0.200)^2$

$A_R = 0.0314$ in^2

E = COMP. MODULUS OF RUBBER

$E = 1000$ PSI   (CONTROLLABLE BY RUBBER
                    TYPE & DUROMETER)

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$$F_o = \frac{(1000)(0.08)(.0314)}{(0.15)}$$

$$F_o = 16.7 \text{ lbs}$$

A_p = PRESSURE AREA ON INSERT HAVING PRESSURE
DIAMETER $D_p = 0.325$ in

$$A_p = \frac{\pi}{4} D_p^2$$

$$A_p = \left(\frac{\pi}{4}\right)(.325)^2$$

$$A_p = 0.0830 \text{ in}^2$$

$$P_I = \frac{16.7}{.0830}$$

$$P_I = 201 \text{ PSI}$$

∴ USE OPENING PRESSURE OF 400 PSI

2.2 SHEAR STRESS ON LOCK PIN INSERT THREADS

S_s = SHEAR STRESS IN THREADS DUE TO PRESSURE, A_s

$$S_s = \frac{F_o}{A_s}$$

WHERE.

F_o = PRESSURE FORCE

$F_o = P_I A_p$

$F_0 = (400)(0.083)$

$F_0 = 33.2$ lbs

$A_s =$ SHEAR AREA OF THREAD

$A_s \geq 0.3 \, \pi \, D_T \, L_T$

WHERE,

    $D_T =$ NOM. THRD. DIA. $= 0.5$ in

    $L_T =$ THRD ENGAGEMENT LNG $= 0.25$ in

$A_s \geq (0.3)(\pi)(0.5)(0.25)$

$A_s = 0.1178$ in^2

$S_s \leq \dfrac{33.2}{0.1178}$

$S_s \leq 282$ PSI

$S_y = 35,000$ PSI

$R = \dfrac{282}{35,000}$

    $R = .0081$ OK

2.3   BEARING STRESS BETWEEN SHIELD
          & OUTER BARREL

S_B = BEARING STRESS

$$S_B = \frac{F_P}{A_B}$$

WHERE, F_P IS THE AXIAL FORCE CAUSED BY
PRESSURE ACTING ON PRESSURE AREA A_P. BOUNDED
BY SEAL DIAMETERS D_0 & D_1 AND A_B
IS THE BEARING AREA.

D_0 = 2.4 in

D_1 = 1.08 in

$A_P = \frac{\pi}{4} (D_0^2 - D_1^2)$

$A_P = \frac{\pi}{4} (2.4^2 - 1.08^2)$

$A_P = 3.608$ in^2

$F_P = P_I A_P$

$F_P = (400)(3.608)$

$F_P = 1443$ lbs

A_B IS AN ANNULAR AREA BOUNDED BY DIAMETERS
D_0 & D_1

D_0 = 2.35 in

D_1 = 2.19 in

$$A_B = \frac{\pi}{4}(D_o^2 - D_2^2)$$

$$A_B = \frac{\pi}{4}(2.35^2 - 2.19^2)$$

$$A_B = 0.571 \text{ in}^2$$

$$S_B = \frac{1443}{0.571}$$

$$S_B = 2527 \text{ psi}$$

$$S_y = 35,000 \text{ psi}$$

$$R = \frac{2527}{35,000}$$

$$R = 0.0722 \Rightarrow OK$$

2.4   HOOP STRESS IN SHIELD CYLINDER

$S_H =$ HOOP STRESS DUE TO EXTERNAL PRESSURE, P_1

$$S_H = \frac{P_1 D_o}{2 W_t}$$

WHERE,

$D_o =$ OUTSIDE DIAMETER $= 1$ in

$W_T =$ WALL THICKNESS $= 0.0625$ in

$$S_H = \frac{(-400)(1)}{(2)(.0625)}$$

$$S_H = -3200 \text{ psi}$$

$S_y = 85,000$ PSI

$R = \dfrac{3200}{85,000}$

$R = 0.0376 \quad \Rightarrow \quad$ OK

$\delta =$ RADIAL DEFORMATION DUE TO S_H

$\delta = \dfrac{S_H D_o}{2 E}$

$E =$ YOUNG'S MODULUS $= 29 \times 10^6$ PSI

$\delta = \dfrac{(-3200)(1)}{(2)(29 \times 10^6)}$

$\delta = 0.000055$ in $\quad \Rightarrow \quad$ OK FOR SEALS

2.5 HOOP STRESS IN OUTER BARREL DUE TO P_I

$S_H = \dfrac{P_I D_I}{2 \cdot W_t}$

$D_I =$ INSIDE DIAMETER $= 2.4$ in

$W_T =$ WALL THICKNESS $= 0.32$ in

$S_H = \dfrac{(400)(2.4)}{(2)(0.32)}$

$S_H = 1500$ PSI

$S_y = 35,000$ PSI

$R = \dfrac{1500}{35,000} = 0.0429 \quad \Rightarrow \quad$ OK

3.0 ANALYSIS OF STRESSES DUE TO DIFFERENTIAL LATERAL PRESSURES

3.1 PRESSURE

ΔP = PRESSURE DIFFERENTIAL ACROSS WALLS OF INNER & OUTER CYLINDERS (POSITIVE TOWARD ₵)

ΔP_{max} = 30,000 PSF = 208 PSI

3.2 HOOP STRESS IN OUTER BARREL

S_H = HOOP STRESS

$$S_H = -\frac{\Delta P \, D_0}{2 W_T}$$

D_0 = OUTSIDE DIA. = 3.0 in

W_T = WALL THICKNESS = 0.0933 in

$$S_H = \frac{-(208)(30)}{(2)(.0933)}$$

$S_H = -3744$ PSI

$S_y = 85,000$ PSI

$$R = \frac{3744}{85,000}$$

$R = 0.0441 \implies$ OK

$\delta = $ RADIAL DISP

$$\delta = \frac{S_H D_o}{2E}$$

$$\delta = \frac{(3744)(3)}{(7)(29 \times 10^6)}$$

$\delta = 0.000194$ in $\Rightarrow$ OK

3.3 HOOP STRESS IN INNER BARREL

$S_H = $ HOOP STRESS

$$S_H = -\frac{\Delta P D_o}{2 W_r}$$

$D_o = $ OUTSIDE DIAMETER $= 1.0$ in

$W_r = $ WALL THICKNESS $= 0.0625$ in

$$S_H = -\frac{(208)(1.0)}{(2)(.0625)}$$

$S_H = -1664$ PSI $\Rightarrow$ OK

4.0   ANALYSIS OF ALLOWABLE SHEAR AND BENDING
         ON INNER BARREL WITH SHIELD ENGAGED

F_S = ALLOWABLE SHEAR ON EACH SUPPORT
        SURFACE OF SHIELD

$F_S = S_B A_P$

WHERE,

S_B = ALLOWABLE BEARING STRESS ON
        PROJECTED AREA

$S_B = 0.25 S_y$ $\Rightarrow$ CONSERVATIVE

$S_B = (0.25)(85,000)$

$S_B = 21,250$ PSI

A_P = PROJECTED AREA

$A_P = D_c L_c$

D_o = CONTACT DIAMETER = 0.95 in

L_c = LENGTH OF CONTACT SURFACE 0.09 in

$A_P = (0.09)(0.95)$

$A_P = 0.068$ in^2

$F_S = (21,250)(0.068)$

$F_S = 1445$ lbs

M = ALLOWABLE BENDING MOMENT ON
        INNER BARREL

$M = F_S L_W$

WHERE,

L_W = DISTANCE BETWEEN SHIELD CONTACT
        POINTS

L_W = 7.9 in

$M = (1445)(7.9)$

$M = 11,416$ in-lbs = 951 FT-lbs

5.0 FRICTIONAL RESISTANCE OF LIP SEAL

T_s = TORSIONAL RESISTANCE OF PARKER SEAL

$T_s = F_R R_s$

WHERE,

F_R = RADIAL FORCE OF SEAL ON COUPLER

R_s = SEAL IR = 0.875 in

$F_R = 2\pi R_s W \Delta P M_s$

WHERE,

ΔP = PRESSURE DROP ACROSS SEAL = 10 PSI

W = CONTACT WIDTH BETWEEN SEAL LIP & COUPLER

W ≤ 0.03 in

M_s = COEF. OF FRICTION BETWEEN SEAL
& COUPLER ≈ 0.08

$F_R = (2\pi)(.875)(.03)(10)(.08) = 0.1319$ lbs

$T_s = (.1319)(.875) = 0.1155$ in-lbs = 0.0096 ft-lbs

APPENDIX - SUPPLEMENTARY DOCUMENTS

TECHNICAL PRODUCTS, INC.

July 10, 1985

Mr. William B. Aiken, P.E.
Sweet & Aiken, Inc.
13810 Champion Forest Drive
Suite 235
Houston, TX   77069

Dear Mr. Aiken:

Confirming our past conversations, Trans-Tek will be able to furnish
a modified 600 Series angular displacement transducer on a state of
the art best effort basis.

The proposed specifications are nominal values which we believe
are attainable at a reasonable cost.

    SENSITIVITY: 100 millivolts per 0.01 degree
          RANGE: + 0.5 degrees
         OUTPUT: + 5 volts
          POWER: + 15 volts at approximately 30 milliamps

The transducer will not have an electrical zero adjustment on the
transducer. Zero output will have to be achieved by the mechanical
position of the transducer. Because of the high sensitivity of
this device the success of your application will depend on whatever
degree of mechanical isolation from extreneous loads your fixture
will allow. We also understand that the transducer will not be
subject to extremes of temperature, humidity, electromagnetic
interference, shock or vibration.

The unit price for this transducer will be approximately $1500.00
including all materials, calibration, and special engineering
charges. Shipment should be about 12 weeks after receipt of a
firm order. Should you have any further questions, please do not
hesitate to contact us.

Best Regards,

Michael B. Stone

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Low capacity torque sensors.

Fig. 1

Model 2105
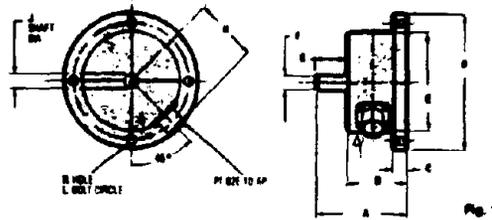
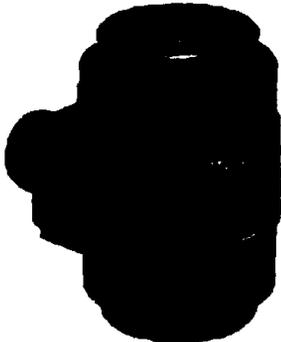|		D	E	F	G	H	J	K	L	M
	in.	in.	in.	in.	in.	in.	in.	in.	in.	in.
2105	2½	1¹¹⁄₆	3	3.491	2⅛	¾	.34	1¹⁄₈	.375	¼	3.082	45°
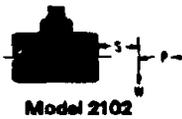
Model 2105   Model 2102

LOAD CARRYING CAPACITY

W = weight of test device   S = distance to center of gravity of test unit
W x S = overhung moment

Do not exceed moment (W x S) or shear (W), whichever value is attained first. P = thrust.

Small flanged reaction torque sensors.

Fig. 5
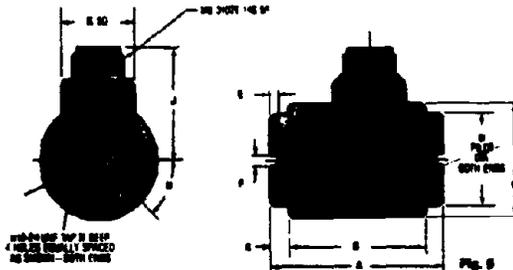
Model 2102

in.	in.	in.	in.	in.	in.	in.	in.	in.	in.	in.	in.
2102	3	2¾	2	1.625	⁵⁄₁₆	.251	⁹⁄₁₆	45°	2	1½	1.250	¾

Specifications	Standard	"H" Option*	Specifications	Standard	"H" Option
Output at rated capacity: millivolts per volt nominal	2 to 2.5	5	Temperature range, useable: °F	−65 to +200	−65 to +200
Nonlinearity: of rated output	±0.1%	±0.15%	Temperature range, useable: °C	−18 to +93	−18 to +93
Hysteresis: of rated output	±0.1%	±0.15%	Temperature effect on output: of reading per °F	±0.002%	±0.003%
Repeatability: of rated output	±0.05%	±0.07%	Temperature effect on output: of reading per °C	±0.0036%	±0.0036%
Zero balance: of rated output	±1.0%	±1.0%	Temperature effect on zero: of rated output per °F	±0.002%	±0.003%
Bridge resistance: ohms nominal	350	350	Temperature effect on zero: of rated output per °C	±0.0036%	±0.0036%
Temperature range, compensated: °F	+70 to +170	+70 to +170	Excitation voltage, maximum: volts DC or AC rms	20	20
Temperature range, compensated: °C	+21 to +77	+21 to +77	Insulation resistance, bridge/case: megohms at 50 VDC	>5000	>5000
Number of bridges	1	1			

*Not available on Models 2105-50 and 2105-100

METRIC DIMENSIONS AND SPECIFICATIONS ARE PURELY MATHEMATICAL CALCULATIONS FROM STANDARD ENGLISH DIMENSION CONTROL DRAWINGS. REQUEST CERTIFIED DRAWINGS BEFORE DESIGNING MOUNTINGS OR FIXTURES. DIMENSIONS AND SPECIFICATIONS ARE SUBJECT TO CHANGE WITHOUT NOTICE.

FEATURES
- Reaction measurements eliminate speed limitations
- Minimal friction error
- No maintenance of slip rings, bearings, or brushes
- Compact "low mass" physical size

Model No.	Rated Capacity Oz. in.	Dimensions	Over-Load Oz. in.	Torsional Stiffness Oz. in./Rad.*	Max. Overhung Moment W x S Oz. in.	Max. Shear W Oz.	Max. Thrust P Oz.
2105-50	50		75	12,000	100	160	320
2105-100	100		150	18,000	150	240	640
2105-200	200	See Figure 1	300	51,600	200	320	960
2105-500	500		750	95,500	250	400	1,600
2105-1000	1,000		1,500	258,000	400	480	2,400

*NOTE: Torsional stiffness given for sensor less shaft extension(s).

Model No.	Rated Capacity Oz. in.	Dimensions	Over-Load Oz. in.	Torsional Stiffness Oz. in./Rad.	Max. Oz. in.	Max. Lbs.	Max. Lbs.
2102-50	50		75	2,350	50	15	200
2102-100	100		150	6,125	100	20	250
2102-200	200	See Figure 5	300	18,600	200	25	400
2102-500	500		750	73,000	250	500	500
2102-1K	1,000		1,500	127,000	500	600	650

C 56

B-2127-D

AVAILABLE CAPACITIES:
@ 10.0, 25.0, & 50.0 IN-OZS.
OUTPUT ~ 1.5mV/V NOM.
©

.1247 / .1245 DIA.

1/8 DIA.

1/32

1/4

5/16

1.0

1/2

7/16

2 5/16

1/32

3/16

RECEPTACLE * PTO2E-10-6P. ④

#27 (.1440) DIA. THRU
3 HOLES EQ. SPACED

1/8

1/4 DIA.

1.500 DIA. B.C.

1.0

1/2

5 CENT./16 TYP

3/4 DIA.

1 1/4

DESCRIPTION

OUTSIDE DIM'S OF MINIATURE R.T.S.
MODEL NO. 2127

ECN414

2127   2107

LEBOW ASSOCIATES, INC.
OAK PARK, MICHIGAN

B-2127-D

C ADDED OUTPUT NOTE
B ECN-1938
A ECN 1631

C-57

TRANS-TEK
INCORPORATED

ANGULAR DISPLACEMENT TRANSDUCER   SERIES 604
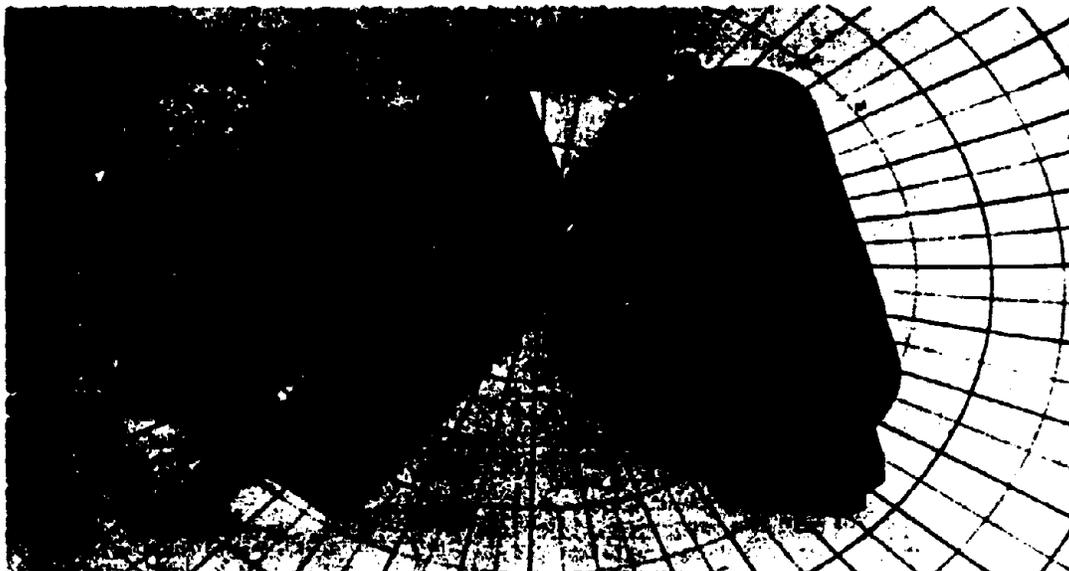
FEATURES

☐ ±0.25% zero base — terminal linearity
☐ 60° linear range
☐ DC in, DC out
☐ 100 mv output per degree of rotation
☐ Infinite resolution
☐ Precision ball bearings for long life
☐ Built in voltage regulator
☐ Can be battery excited
☐ Shaft rotational speed up to 3000 rpm

APPLICATIONS

Series 604 ADT's are ideal for use in:

☐ Servo position feedback
☐ Rotary actuators
☐ Robotic wrist and elbow position
☐ Rotary valves, throttle and antenna position
☐ Torque displacement
☐ Inclinometers, vertical references
☐ Positioning optical devices
☐ Web or film tension control

THEORY

Series 604 Angular Displacement Transducers (ADT's) are precision differential capacitors with integral voltage regulation, oscillator, demodulator and output buffer amplifier. The angular position of the shaft is converted into a DC voltage directly proportional in amplitude and polarity to displacement from the electrical null position.

CONSTRUCTION

Optimum performance and reliability is obtained by our selection of components and factory calibration. Mechanical parts are machined to exacting tolerances from stable materials and are corrosion protected where required. The shaft is supported by ball bearings for low torque, resistance to side load and to insure a long life.

EASE OF USE

After mechanical installation, connect the transducer to a dual polarity power supply. The 0604 series internal voltage regulator allows it to operate from widely ranging, unregulated voltage sources or regulated ones, such as Trans-Tek's D15.100. The high level, DC output can be fed directly into voltmeters, recorders, A/D converters or other devices. The standardized output allows interchanging units without system recalibration.

INSTALLATION

The precision pilot diameter and tapped mounting holes on the base plate are provided for quick installation. Once mounted, approximate mechanical zero is established by aligning the mark on the shaft midway between the zero and span controls on the cover. The electrical zero adjustment can then be used to correct for small offsets. Electrical connections are via terminal strip, eliminating the need for mating connectors or terminal lugs.

C58

ELECTRICAL SPECIFICATIONS

MODEL NO.	LINEAR[1] DISPL. RANGE	MAX. NON-LIN[2] LINEAR RANGE	USABLE DISPL. RANGE	MAX. NON-LIN. USABLE RANGE	OUTPUT SENSITIVITY	TYP. TEMP. COEF. OF SPAN
0604-0000	± 30°	± 0.25%	± 35°	± 1.0%	100mVDC/°	< ± 0.01% Output/°F
0604-0001	0 to 60° CW	± 0.25%	−5° to +65° CW	± 1.0%	100mVDC/°	< ± 0.01% Output/°F
0604-0002	0 to 60° CCW	± 0.25%	−5° to +65° CCW	± 1.0%	−100mVDC/°	< ± 0.01% Output/°F

(1) CW defined as clockwise direction of shaft rotation, when viewed from shaft end.
(2) Zero base terminal average, expressed as max % deviation of total range.

POWER SUPPLY	VOLTAGE	± 14.5 to ± 30VDC unregulated. input polarity protected.
	CURRENT	± 20maDC maximum when load current is less than ± 5maDC.
OUTPUT	VOLTAGE	Factory calibrated to deliver 100mVDC per degree of rotation unless otherwise ordered.
	CURRENT	Load can draw up to ± 5maDC without significant effect on non-linearity; Output is protected against short circuits to power supply lines (50maDC max. short circuit current).
	RIPPLE	Max. peak-to-peak ripple voltage is 0.5% of total DC voltage over linear range; ripple is at 500KHz nom.
	IMPEDANCE	Less than 1 ohm when output current is less than ± 5maDC.

REPEATABILITY	< .02% of range.
RESOLUTION	Infinite.
ZERO CONTROL	10% of total linear range via multi turn pot.
SPAN CONTROL	20% range via multi turn pot.
ANGULAR VELOCITY	2% max. attenuation of signal when operating at 10,000 °/second.

MECHANICAL SPECIFICATIONS

DISPLACEMENT RANGE		Continuous; there are no stops.
TORQUE	STARTING	5 gram-cm maximum.
	RUNNING	3.5 gram-cm maximum.
MOMENT OF INERTIA (SHAFT)		0.6 gram-cm²
LOAD	RADIAL	10 lbs. maximum.
	AXIAL	7 lbs. maximum.
BEARING LIFE ex: 17,000 hours expected at 10 RPM with 10 lb. radial load at shaft end.		

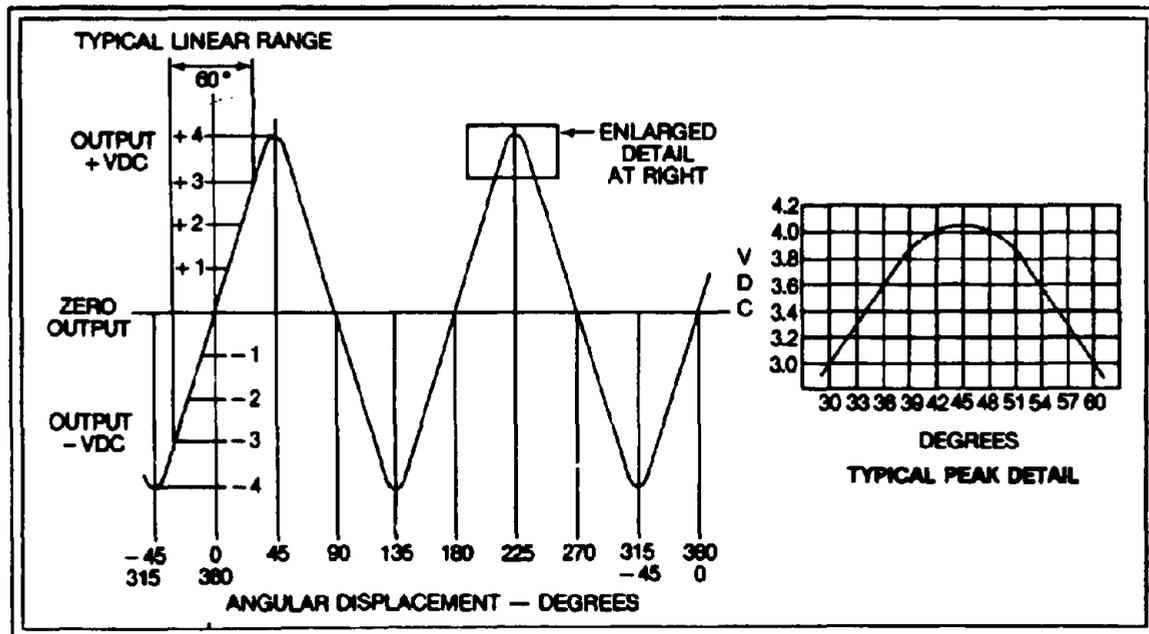| TEMPERATURE OPERATING STORAGE	0°-70°C −55° to + 125°C
MOUNTING	Any position. gravity insensitive
NET WEIGHT	4.3 oz. (120 grams)

OPTIONS:
The following options are available at added cost at time of purchase.
1. Factory calibrated to any specified output sensitivity, output voltage limited to ± 8VDC.
2. Calibrated for inverse slope; standard output with reversed shaft rotation.
3. Operation over −55°C to + 125°C, weight increases to 7.4 oz. (210 grams).

CIRCUIT BLOCK DIAGRAM

SHAFT POSITION

SPAN ADJUST

ZERO ADJUST

| +DC IN — 1 | VOLTAGE REGULATOR | OSCILLATOR | VARIABLE DIFFERENTIAL CAPACITOR | DEMODULATOR | BUFFER AMPLIFIER | 4 — SIGNAL OUT |

+DC IN 1

−DC IN 3

INPUT RETURN 2

SIGNAL COMMON 5

CASE, SHAFT, TERM. 2 AND TERM. 5 ARE COMMON
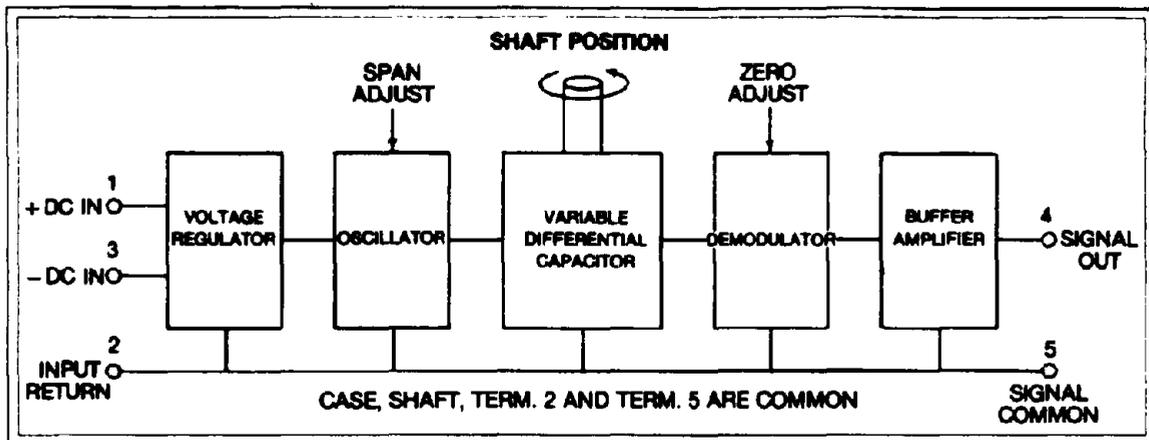
CIRCUIT BLOCK DIAGRAM

OUTPUT FUNCTION

TYPICAL LINEAR RANGE

60°

OUTPUT +VDC: +4, +3, +2, +1

ZERO OUTPUT

OUTPUT −VDC: −1, −2, −3, −4

ENLARGED DETAIL AT RIGHT

V D C: 4.2, 4.0, 3.8, 3.6, 3.4, 3.2, 3.0

30 33 36 39 42 45 48 51 54 57 60

DEGREES

TYPICAL PEAK DETAIL

−45 / 315, 0 / 360, 45, 90, 135, 180, 225, 270, 315 / −45, 360 / 0

ANGULAR DISPLACEMENT — DEGREES

OUTPUT FUNCTION

The illustration above is a plot of output voltage versus shaft position for Model 0604-0000. By definition the output is 0 VDC at the zero position. Rotating the shaft CW provides a positive voltage proportional to the angular displacement from the zero position. CCW rotation from the zero position produces a negative voltage, again proportional to the shafts position in relation to 0°. The 0604-0001 is designed to provide a 0 to +6 VDC output proportional a 0° to 60° CW rotation, Model 0604-0002 gives a 0 to −6 VDC output proportional to a 0° to 60° CCW rotation.

DC-DC ADT
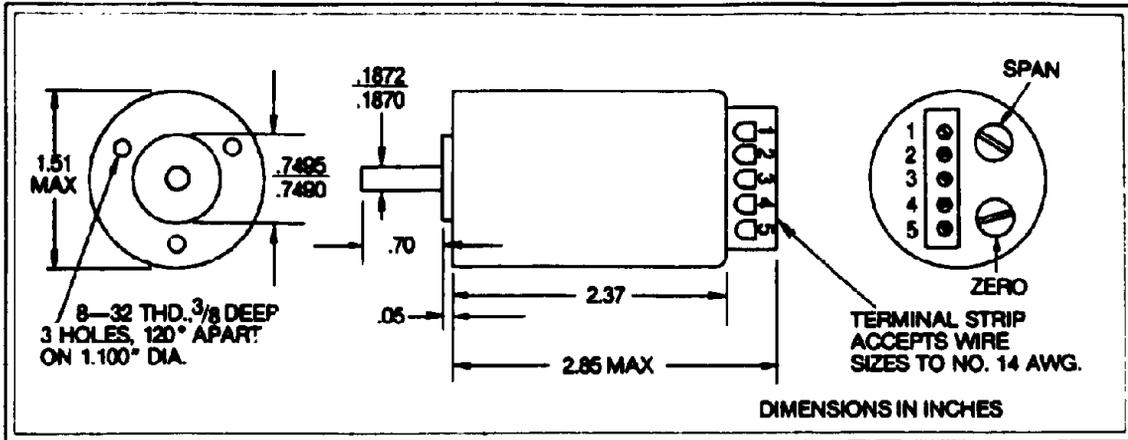
DIMENSIONAL DIAGRAM

WIRING
TERMINAL FUNCTION
1.   + DC input
2.   Input return
3.   − DC input
4.   Signal output
5.   Signal common
NOTE: Case, shaft, and terminals 2 & 5 are common

CALIBRATION
This Model 0604 ADT has been factory calibrated as noted below. User should check calibration before placing instrument in service. To check calibration, or to recalibrate, proceed as follows:
1. Secure the housing and shaft to a suitable test rig.
2. Power the ADT and allow a 10 minute warmup.
3. Rotate the shaft to the zero position (coarsely identified by aligning mark on end of shaft between zero and span access holes.)
4. The span pot is used to adjust the output for 6.000 ± 0.001VDC difference between the − 30° and + 30° or 0° and 60° positions, depending on the model. Units ordered with special zero and sensitivity values should be calibrated to these.

INSTALLATION
The ADT should be mounted and bolted in position with due care taken to avoid damage to the precision ball bearings. Rotate the shaft to the proper zero position, coarsely identified by aligning mark on shaft end to be between ZERO and SPAN controls. After powering the ADT, set the mechanism to be monitored at its zero position, and then lock the ADT shaft to the rotating mechanism. Small misalignments between the ADT zero position and the mechanism zero position can be eliminated by using the ADT ZERO control.

MODIFICATIONS
Transducers for special applications are available. Consult TRANS-TEK, Incorporated on your particular requirement.

CALIBRATION RECORD
This section will be completed during final inspection.

Model No. _____ Date Code _____

Serial No. (optional) _____ _____

Calibration Temperature _____ °F

Calibrated to deliver 0.1000 VDC per degree of rotation with input power ± 14.5 to ± 15.5 VDC unless otherwise noted below.

Special Calibration (optional) _____

Unit is supplied with following options (checked if applies):
☐ 1. Special Sensitivity _____.
       Zero Offset _____.
☐ 2. Calibrated for Inverse Slope.
☐ 3. For − 55°C to + 125°C Operation.

NOTES
All specifications are subject to change without notice. Contact TRANS-TEK for quantity discount prices available on all models. (For areas beyond the United States — contact the international representative.)

ORDER PLACEMENT
Orders should be made out to TRANS-TEK, Incorporated, sent in care of your local TRANS-TEK representative, or directly to Box 338, Ellington, Connecticut 06029.
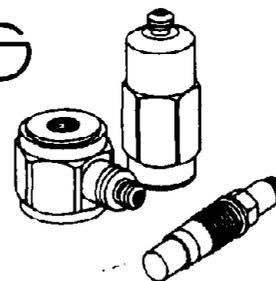
WARRANTY
All TRANS-TEK transducers are warranteed against defective materials and workmanship for one year.

C61

Quartz sensors

C62

ICP QUARTZ
ACCELEROMETERS
with built-in amplifiers[1]
Series 300

STANDARD DESIGN CONFIGURATIONS – Photos are not proportional to each other. See page following chart for dimensional drawings.

Series 303A
303A02 303A03
309A Series 302A 302A09 307A Series 308B 308B04 308B12
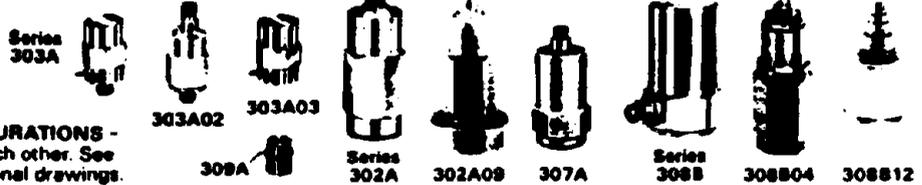
Application	MINIATURE, LOW MASS					GENERAL PURPOSE – HIGH								
	Stud Mtg Side Conn	Stud Mtg Top Conn	Adh Mtg Side Conn	High Sens 100mV/g	Low Gram Weight	Low Strain	Gnd Iso, Lo Strain	Gnd Iso 10K	Integral Mtg Base 0 Strain	Hi Sens Lo Mass Gnd Iso	High Sens	High Freq Low Mass	Side Connector	
Model Number (see optional models below specifications)	303A	303A02	303A03	303A11	309A	302A	302A04	302A06	302A09	308B15	302B03	307A	308B	308B03
Sensitivity Peak[2] mV/g	10	10	10	100★	5	10±2%	10±2%	10±2%	10±2%	100±2%★	300±2%★	100±5%★	100±2%	50±2%
Range (for ±5V out) ±g pk	500	500	500	10 (±1V)	1000	500	500	500	100 (±1V)	50	15	50	50	100
Range (for ±10V out)[1] ±g pk	—	—	—	—	—	1000	1000	1000	—	—	32	100	100	200
Resolution g pk	.01	.01	.01	.005★	.02	.01	.01	.01	.005	.001	.0025	.005	.001	.002
Resonant Freq (mtd) >kHz	70	100★	70	70	120★	45	45	45	20	25	20	>40	25	25
Freq Range ±5%[7] Hz	1 to 10000	1 to 10000	1 to 10000	5 to 10000	5 to 10000	1 to 5000	1 to 5000	1 to 10000★	1 to 5000	1 to 3000	1 to 5000	2 to 10000	1 to 3000	1 to 3000
Freq Range ±10%[7] Hz	.7 to 20000	.7 to 20000	.7 to 20000	3 to 20000	2 to 20000	.7 to 10000	.7 to 10000	—	.7 to 10000	.7 to 6000	.7 to 7000	1.3 to 15000	.7 to 6000	.7 to 6000
Overload Recovery μs	10	10	10	10	10	10	10	10	10	10	10	10	10	10
Discharge Time Constant[2] @ 70°F >s	0.5	0.5	0.5	0.2	0.1	0.5	0.5	0.5	0.5	0.5	0.2	0.3	0.5	0.5
Amplitude Linearity (zero based best straight line) %	1	1	1	1	1	1	1	1	1	1	1	1	1	1
Output Impedance★ <ohm	100	100	100	100	100	100	100	100	100	100	100	100	100	100
Transverse Sens (max) %	5	5	5	5	5	5	5	5	5	5	5	5	5	5
Strain Sens g/μin/in	.05	.05	.05	.05	.005	.01	.01	.01	0★	.05	.01	1	.05	.05
Temperature Range[4] °F	-40 to +200	-40 to +200	-40 to +200	-40 to +120	-40 to +150	-100 to +250	-100 to +250	-100 to +250	-100 to +250	-100 to +250	-65 to +175	-100 to +250	-100 to +250	-100 to +250
Temp Coefficient %/°F	.03	.03	.03	.03	.05	.03	.03	.03	.03	.03	.03	.03	.03	.03
Vibration (max) ±g pk	1000	1000	1000	500	1000	2000	2000	2000	1000	1000	100	500	500	1000
Shock (max) g pk	2000	2000	2000	1000	2000	5000★	5000★	5000	2000	10000	5000	5000	5000★	10000
Structure (mechanical compression)	upright	upright	upright	upright	upright	inverted	inverted	inverted	inverted	upright	inverted	upright	upright	upright
Size (hex x height) in	.281 x .48	.281 x .81	.281 x .48	.281 x .48	.23 dia★ x .30	.50 x 1.3	.50 x 1.3	.50 x 1.3	.50 x 1.5	.75 x 1.10	.562 x 1.81	.625 x 1.50	.75 x 1.32	.75 x 1.07
Weight gm	2★	2.3★	1.9★	2★	1★	25	25	25	38	55★	39★	31	87	55
Connector	integral[5] 10' cable	5-44 micro	integral[5] 10' cable	integral[5] 10' cable	integral[5] 8' cable	10-32	10-32	10-32	10-32	10-32	10-32	10-32	10-32	10-32
Case Material SS	316	316	316	316	316	316	316	316	316	SS	316	SS	SS	SS
Sealing	epoxy	epoxy	epoxy	epoxy	epoxy	epoxy	epoxy	epoxy	epoxy	hermetic	hermetic	hermetic	epoxy	epoxy
Ground Isolation[6]	no	no	no	no	no	no	yes	yes	no	yes	no	yes	no	no
OPTIONAL MODELS (see specific data sheets)				303A13 adhesive mount		302A07 low phase		302A08 non-iso		308B14 low cost non-iso non-herm			308B09 308B10	308B11 hermetic sealed

NOTES: (★ denotes exceptional characteristic)
(1) PCB power units provide 2 to 20 mA from +18 to -28V DC supply. For 10V output, +24 to +28V DC is required. For driving cables longer than 100 ft, use 4 mA.
(2) Discharge time constant relates to low frequency response per chart and to transient events lasting up to several percent of discharge TC. For half sine events, discharge TC should be 25 times event duration. Use 484B or 484B02 Series power units to utilize the full transducer discharge TC.
(3) Standardized to tolerance indicated nominal otherwise.
(4) Special cryogenic models operate to 440°F. Special high temperature to +325°F.
(5) Supplied with attached 10 ft ribbon wire termination. 10-32 coaxial connector

C63

					Series 306A		301A Vibration Calibration Std in 394A03 System (see specific data sheet)	PCB PIEZOTRONICS	MOTION shock & vibration

SENSITIVITY					**INDUSTRIAL**		**TRIAXIAL**			**SHOCK**				
Top Conn. Gnd Iso	High Temp ±325°F	2nd Order LP Filter Built-in	Seismic Low Mass	Seismic Low Freq	Low Strain	Norm Seal Gnd Iso	One Inch Cube	5/8 Inch Cube	Three 303A's on Block	General Purpose	High g, Low Mass			
308B12	308B08	308B30	308B02	393C	308B04	328B	306A	306A06	303A06	302A02	305A	305A02	306A03	305A04
100 ±5%	100 ±2%	100 ±2%	1000 ±5%	1000	100 ±2%	100 ±5%	10 ±2%	10	10	10 ±2%	05	1	5	1
50	50	50	5	2.5 ±2.5V	50	50	500	500	500	500	100 000	50 000	10 000	5000
100	100	100	10	—	100	100	1000	—	—	1000	—	100000	20 000	10 000
001	001	001	0005	0001	001	001	01	02	01	01	2	1	2	1
30	25	25	25	3.5	25	18	8	28	7	45	60	60	60	60
1 to 3000	4 to 3000	1 to 20	2.5 to 3000	025 to 800	2 to 5000	1 to 3000	1 to 1000	1 to 3000	1 to 1000	05 to 5000	25 to 8000	25 to 8000	25 to 8000	25 to 8000
7 to 5000	2 to 6000	7 to 24	15 to 5000	01 to 1200	13 to 7000	7 to 5000	7 to 3000	7 to 10000	—	03 to 10000	1 to 12000	1 to 12000	1 to 12000	1 to 12000
10	10	10	10	10	10	10	10	10	10	10	10		10	10
05	02	05	02	20	0.5	05	05	05	05	10	20	20	20	20
1	1	1	1	1	1	1	1	1	1	1	1	1	1	1
100	100	100	100	100	100	100	100	100	100	100	100	100	100	100
5	5	5	5	5	5	5	5	5	5	5	5	5	5	5
05	05	05	05	001	01	1	01	05	01	01	2	2	2	2
-100 to +250	-100 to +325	-100 to +250	-65 to +175	-100 to +200	-100 to +250	-50 to +250	-100 to +250	-100 to +250	-100 to +200	-100 to +175	-100 to +250	-100 to +250	-100 to +250	-100 to +250
03	03	03	03	03	03	05	03	03	03	03	03	03	03	03
200	500	500	100	100	500	500	1000	1000	1000	2000	100 000	75 000	20 000	10 000
5000	5000	5000	200	100	5000	5000	2000	2000	2000	5000	125K axial 30K trans	100K axial 30K trans	50K axial 30K trans	20K axial 20K trans
upright	inverted	upright	upright	upright	inverted	upright	inverted	inverted	upright	inverted	inverted	inverted	inverted	inverted
75 x 1.73	75 x 1.07	75 x 1.32	75 x 1.32	2.2 dia x 2.2	75 x 2.0	10 dia x 2.5	10 cube	625 cube	86 cubic envelope	50 x 1.3	312 x 68	312 x 68	312 x 68	312 x 68
60	55	95	65	1000	97	180	130	17	22	25	4	4	4	4
10-32	10-32	10-32	10-32	10-32	5/16-32 Con-hex	10SL 4P-20-0A	10-32	integral 6' cable	integral 10' cable	10-32	10-32	10-32	10-32	10-32
SS	SS	SS	SS	SS	SS	316	316	titanium	SS	316	17-4PH	17-4PH	17-4PH	17-4PH
hermetic	hermetic	epoxy	epoxy	hermetic	hermetic	hermetic	epoxy	epoxy	epoxy	epoxy	epoxy	epoxy	epoxy	epoxy
yes	no	no	no	yes	no	yes	no	no	no	no	no	no	no	no
308B13 50mV/g		308B31-34 50-100-200 & 500 Hz			308B05 base iso		306A82 shock		303A87 100mV/g			305A12	305A13	305A14 built in second order 10K LP filter

(6) Accessory isolation pads and studs are available

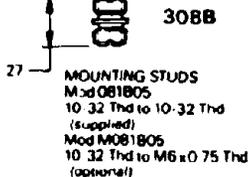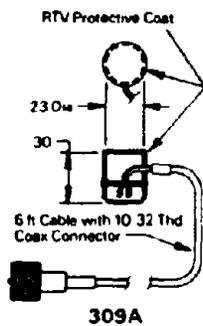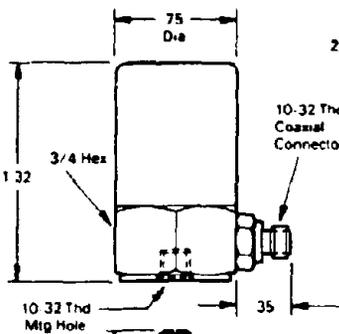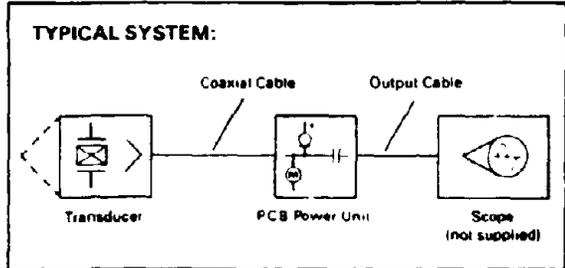(7) Calibration furnished to 5%, optional calibration available to 10% frequency range
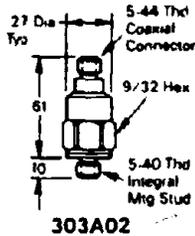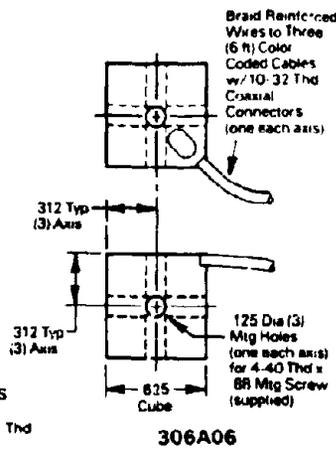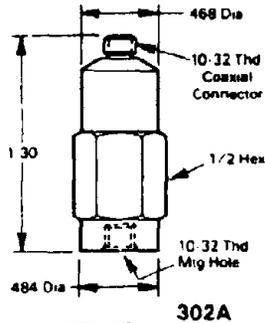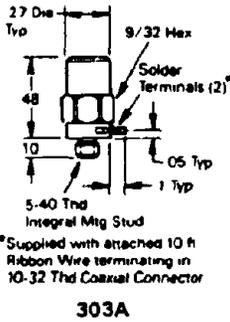
(8) Field repairable solder terminal connection

OPTIONS. Special sensitivities, ranges, electrical isolation, hermetic sealing, low and high pass filtering, case pressure insensitivity, lower frequency response, longer discharge TC, biased and limited output, lower noise, lower temperature electronics to -440°F or higher temperature to +325°F, conventional charge output, solder connector adaptor and thermal jackets.

N.B.S. TRACEABLE CALIBRATION CERTIFICATE IN COMPLIANCE WITH MIL-STD-45662 FURNISHED WITH EACH ACCELEROMETER

65

MOTION
shock & vibration

PCB
PIEZOTRONICS

DIMENSIONAL DATA
ACCELEROMETERS
with built-in amplifiers
Series 300

TRANSDUCER CONFIGURATION DIMENSIONS:

See specific data sheet for transducers not shown.

27 Dia Typ
9/32 Hex
Solder Terminals (2)*
48
10
05 Typ
1 Typ
5-40 Thd Integral Mtg Stud

*Supplied with attached 10 ft Ribbon Wire terminating in 10-32 Thd Coaxial Connector

303A

468 Dia
10-32 Thd Coaxial Connector
1 30
1/2 Hex
10-32 Thd Mtg Hole
484 Dia

302A

MOUNTING STUDS
Mod 081805
10-32 Thd to 10-32 Thd (supplied)
Mod M081805
10-32 Thd to M6 x 0 75 Thd (optional)
27

Braid Reinforced Wires to Three (6 ft) Color Coded Cables w/10-32 Thd Coaxial Connectors (one each axis)
312 Typ (3) Axis
312 Typ (3) Axis
625 Cube
125 Dia (3) Mtg Holes (one each axis) for 4-40 Thd x 88 Mtg Screw (supplied)

306A06

5/8 Thd, Two Pin Electrical Connector
15/16 Hex
2 5
1 8
1 0 Dia
1/4-28 Thd Mtg Hole

27 Dia Typ
5-44 Thd Coaxial Connector
9/32 Hex
61
10
5-40 Thd Integral Mtg Stud

303A02

SOLDER CONNECTOR ADAPTOR Model 070A09 (accessory)
Sig/Pwr
gnd
10-32 Thd

328B
37

MOUNTING STUDS
Mod 081820
1/4-28 Thd to 1/4-28 Thd (supplied)
Mod M081820
1/4-28 Thd to M6 x 0 75 Thd (optional)

10-32 Thd Coaxial Connector
310 Dia Typ
5/16 Hex
68
10-32 Thd Integral Mtg Stud
Dim "A" = 14 for Models 305A03 305A04 305A05
Dim "A" = 29 for Models 305A 305A02
A

305A 305A02

575 Dia
10-32 Thd Coaxial Connector
1 5
5/8 Hex
10-32 Thd Mtg Hole
580 Dia

307A

MOUNTING STUDS
Mod 081805
10-32 Thd to 10-32 Thd (supplied)
Mod M081805
10-32 Thd to M6 x 0 75 Thd (optional)
27

TYPICAL SYSTEM:

Coaxial Cable   Output Cable

Transducer   PCB Power Unit   Scope (not supplied)

75 Dia
3/4 Hex
1 32
10-32 Thd Coaxial Connector
10-32 Thd Mtg Hole
35

30BB

27
MOUNTING STUDS
Mod 081805
10-32 Thd to 10-32 Thd (supplied)
Mod M081805
10-32 Thd to M6 x 0 75 Thd (optional)

RTV Protective Coat
23 Dia
30
6 ft Cable with 10-32 Thd Coax Connector

309A

TYPICAL KIT:

Includes accelerometer, 10 ft input cable, power supply, and 3 ft output cable in a convenient carrying case.

Designate and order as follows

For battery power kit add 'K' to accelerometer Model No (e.g. K302A).

For line power kit add 'KL' to accelerometer Model No (e.g. KL302A)

Special length cables to 50 ft supplied in kits at no additional charge, if specified.

Typical Kit

MINIATURE 2 GRAM

QUARTZ ACCELEROMETER

Series 303A

PCB
PIEZOTRONICS

MOTION
shock & vibration

with built-in microelectronics & 10 mV/g sensitivity

Measure shock and vibration in applications requiring small size, low mass or very high frequency response.

Series 303A Quartz Accelerometers function to transfer shock and vibratory motion into high-level, low-innnedance (100 ohm) voltage signals compatible with readout, recording or analyzing instruments. These tiny sensitive (10 mV/g) sensors operate reliably over wide amplitude and frequency ranges under adverse environmental conditions.

They are structured with permanently polarized compression-mode quartz elements and a microelectronic amplifier housed in a lightweight metal case. Three different case and connector configurations give you a choice in mounting and cabling. The built-in electronics operate over a coaxial or two-conductor cable; one lead conducts both signal and power. Solder terminal versions are normally supplied with a ribbon wire cable (10 ft. long; Model 007B10) attached. Model 303A02 requires Model 002G coaxial cable with a Micro 5-44 connector on one end.

Test results of the behavior of the Model 303A are presented below. Note especially the sharp clean signals free of cable noise and the exceptionally high frequency response. Because of the low mass, Series 303A sensors measure n.otion of many light structures without appreciably changing the structure or behavior of the test object during the measuring transaction.

Frequency Response (mounted)

sensitivity

frequency →   50K
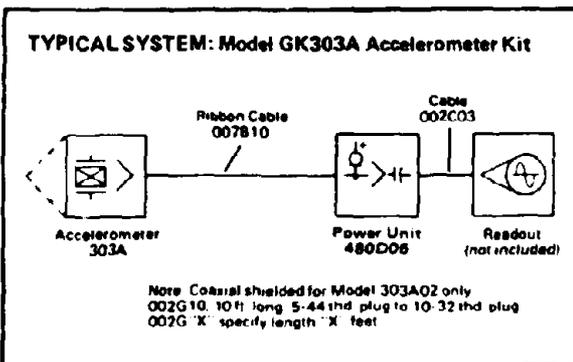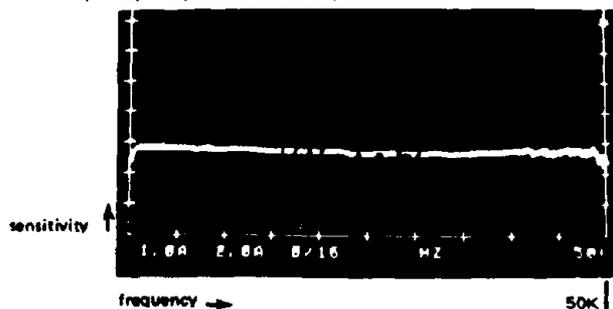
303A

Supplied with attached
10' ribbon wire terminating
in 10-32 Micro connector
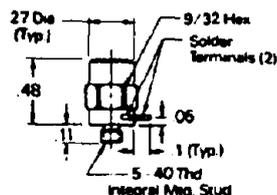
303A02
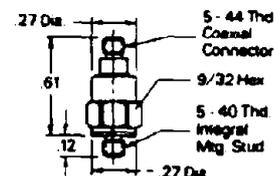
303A03

Supplied with attached
10' ribbon wire terminating
in 10-32 Micro connector
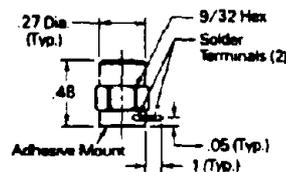
SPECIFICATIONS: Model No.		303A & 303A03
Range (for ±5 output)	g	±500
Resolution	g	0.01
Sensitivity (nominal)	mV/g	10
Resonant Frequency (mounted)	kHz	70
Frequency Range (±5%)	Hz	1 to 10000
Discharge Time Constant	sec	1
Linearity	%	1
Output Impedance	ohm	100
Outer t Bias (nominal)	V	11
Overload Recovery	microsec	10
Transverse Sensitivity (max.)	%	5
Strain Sensitivity	g/μin/in	0.05
Temperature Coefficient	%/°F	0.03
Temperature Range (operational to +250°F)	°F	−40 to +200
Vibration	g	±1000
Shock (protected)	g	2000
Size (hex. x height)	in	0.28 × 0.48
Weight (approx.)	gm	2
Connector (solder terminals)		2
Case Material		s.s.
Seal		epoxy
Excitation Voltage	V	+18 to 24
Excitation Current (constant)	mA	2 to 20

Notes:

Model 303A02 has a 5-44 micro-connector. Other specifications are the same.

Options include 080A15 adhesive mounting base, 080A16 three-axis mounting adaptor (10-32 thread) and triaxial Model 303A06.

TYPICAL SYSTEM: Model GK303A Accelerometer Kit

Accelerometer 303A   Power Unit 480D06   Readout (not included)

Note: Coaxial shielded for Model 303A02 only
002G10, 10 ft long 5-44 thd plug to 10-32 thd plug
002G "X" specify length "X" feet
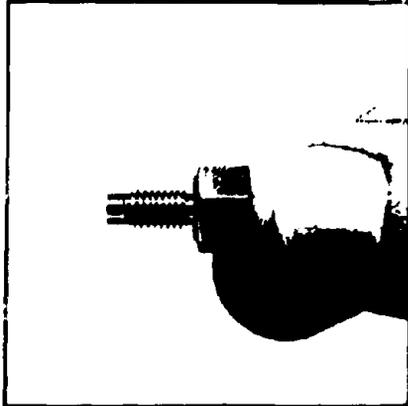
Entran

EPX Series
Miniature Threaded Pressure Transducers

- **5 PSI to 5000 PSI (.35 BAR TO 350 BAR)**
- **10-32 UNF OR M5 METRIC THREADED MOUNT**
- **O-RING SEAL, STAINLESS STEEL CONSTRUCTION**
- **STATIC & DYNAMIC PRESSURE**

Entran's EPX Miniature threaded pressure sensors represent a unique approach in transducer design based on mounting ease and flexibility of use. Developed with a wide range of applications in mind, the EPX offers an optimum combination of characteristics which permit static and dynamic pressure measurements when small size and ease of attachment are of prime importance.

The EPX has a stainless steel diaphragm construction which eliminates the cracking and shattering problems normally associated with silicon diaphragm transducers. Available in either a 10-32 UNF or M5 metric thread, with built in O-ring seal, it allows simple mounting to test objects or systems. A custom thread length option enables matching the EPX directly to existing requirements and allows specification of the exact transducer length between 0.250" and 1.500"(6.3mm and 40mm), instead of machining the fixture to match the transducer. Another standard option incorporates two varieties of "Welded" diaphragm seals for use in water or corrosive fluids compatible with 304SS. In short, it is possible to customize the transducer directly from this data bulletin without the necessity of complicated special order procedures.

The EPX is a semiconductor strain gage device which combines a fully active Wheatstone bridge with state of the art transducer design. The semiconductor elements are bonded directly to the stainless steel diaphragm thereby providing extremely high frequency response coupled with extremely low sensitivity to extraneous accelerations, vibrations and shocks. Its high output enables the EPX to drive most recorders and data monitoring systems directly, without amplification and costly signal conditioning. The semiconductor circuitry is fully compensated for temperature changes in the environment.

The EPX is ranged for the pressures which are commonly experienced in research, testing and control, from 5 to 5000 psi (.35 to 350 bar) (for ranges higher than 5000 psi or 350 bar, contact Entran directly.) Typical uses have varied in scope from missile engine and skin pressure studies, automotive fuel line and brake hydraulic tests, ship model impact and perturbation studies to chemical process control and general university laboratory research.

Entran
Specification   **EPXS-185**

SPECIFICATIONS
EPX-10

RANGE psi	5	10	15	25	50	100	250	500	1000	2500	5000
OVERRANGE psi	50	50	50	50	100	200	500	1000	2000	4000	6000
SENSITIVITY mV/psi (nom.)	3	2.5	2	2	1.5	1.2	.5	.25	.12	.05	.025
mV F.S.	15	25	30	50	75	125	125	125	125	125	125
[1] RES. FREQUENCY nom.	45KHz	50KHz	65KHz	65KHz	75KHz	80KHz	120KHz	150KHz	200KHz	300KHz	450KHz
COMB. NON-LIN. & HYST.	±1%F.S.					±¾%F.S.	±½%F.S.				
TEMP. SHIFT ZERO	±1mV/100°F			±2%F.S./100°F		±1½%F.S./100°F					
SENS.	±2%/100°F (±3½%/100°F for EPX6-)										
g SENS. nom. %F.S./g	.015	.007	.0048	.003	.0016	.0012	.0006	.0004	.0003	.0002	.0001

[1] Useful Frequency Range is 20% of Resonant Frequency

SPECIFICATIONS COMMON TO ALL RANGES AND THE METRIC SERIES

	EPX-	EPX6-	CUSTOM OPTIONS[2] (see Selection Manual)
EXCITATION	10 VDC	6 VDC	3 to 15 VDC or VAC
IMPEDANCE INPUT	1200 Ω nom. typ. (350 Ω min.)	700 Ω nom. typ. (350 Ω min.)	200 Ω to 2500 Ω nom.
OUTPUT	350 Ω nom.	350 Ω nom.	120 Ω to 500 Ω nom.
REPEATABILITY	±0.25%	±0.25%	±0.1%
RESOLUTION	INFINITE		
COMPENSATED TEMP.	70°F to 170°F (21°C to 77°C)		Ranges within: −100°F to 450°F / −73°C to 230°C
OPERATING TEMP.	−40°F to 250°F (−40°C to 121°C)		−100°F to 500°F (−73°C to 260°C)
ZERO OFFSET (70°F / 21°C)	±10mV	±10mV	±1%F.S.

[2]Custom options may alter other specifications and are not necessarily available on all models and in all combinations.
Contact Entran directly for your specific requirements.

WIRING DIAGRAM

METRIC SERIES
EPX-M5

RANGE	bar	0.35	0.7	1	1.5	3.5	7	15	35	70	150	350
OVERRANGE	bar	3.5	3.5	3.5	3.5	7	14	30	70	140	240	420
SENSITIVITY (nom)	mV/bar	40	35	30	30	20	17	8	3.5	1.7	.8	.35
	mV F.S.	15	25	30	45	75	125	125	125	125	125	125
†RES. FREQUENCY nom.		45KHz	50KHz	65KHz	65KHz	75KHz	80KHz	120KHz	150KHz	200KHz	300KHz	450KHz
COMB. NON-LIN. & HYST.		±1%F.S.				±¾%F.S.		±½%F.S.				
TEMP. SHIFT	ZERO	±1mV/50°C				±2%F.S./50°C		±1½%F.S./50°C				
	SENS.	±2%/50°C (±3½%/50°C for EPX6-)										
g SENS. nom. %F.S./g		.015	.007	.0048	.003	.0016	.0012	.0006	.0004	.0003	.0002	.0001

†-Useful Frequency Range is 20% of Resonant Frequency

MOUNTING STYLES

EPX-10 or EPX-M5 } External Compensation Module

EPX-10I or EPX-M5I } Internal Compensation (Module within Housing)

EPX-10U EPX-10W or EPX-M5U EPX-M5W } External Compensation Module

EPX-10IU EPX-10IW or EPX-M5IU EPX-M5IW } Internal Compensation (Module within Housing)

FOR WATER OR CORROSIVE FLUIDS COMPATIBLE WITH 304SS

For water usage less than 8 hours continuous, the standard epoxy seal is adequate. For water usage more than 8 hours or in fields which normally swell epoxy, select Entran's "Braze Weld", option "U". For use in corrosive media compatible with 304SS, Biomedical or Food Processing work, select Entran's "Seam Weld", option "W". "U" weld cannot be used above 2500 psi (150 bar) and has a temperature limit of 300°F (150°C).

CUSTOM SCREW LENGTH OPTION

Standard EPX units are supplied with DIM "X", the distance from sealing surface to diaphragm, as 7/16" (11.1mm). You can order any length you desire between 0.250" and 1.50" (6.3mm and 40mm). Just add the custom "X" dimension to the model number as shown on the back page and consult the price list for the additional charge. All EPX-10 units have "X" expressed in inches and EPX-M5 units have "X" expressed in mm.

LEAD WIRES & COMPENSATION MODULE

All units are supplied with 24"(61cm) of Teflon insulated shielded cable with the Compensation Module located 18" (46cm) from the transducer (except for internal compensation units). The module is supplied with each unit wired into the leads. If you desire longer lead lengths or module location other than 18" (46cm), please consult price list and specify at time of order.

TO ORDER AN "EPX" SERIES PRESSURE TRANSDUCER

1. Select Desired Input Voltage Option EPX or EPX6.
2. Select Desired Mounting Style in Standard or Metric Versions.
3. Select Welded Version If Required.
4. Select Pressure Range.
5. Specify Pressure Reference Gauge, Absolute, Differential or Sealed Gauge.
   If Not Specified, Unit Will Be Supplied as Gauge.
6. Specify Custom Thread Length If Required.

EPX	— 10	U	—	100	G	—	(XX)

SERIES   STYLE   WELDED OPTION   PRESSURE RANGE   REFERENCE: CUSTOM THREAD LENGTH

REFERENCE:
G-Gauge (Relative)
D-Differential (Wet/Dry or Dry/Dry)
S-Sealed Gauge
A-Absolute
(Gauge if not specified)
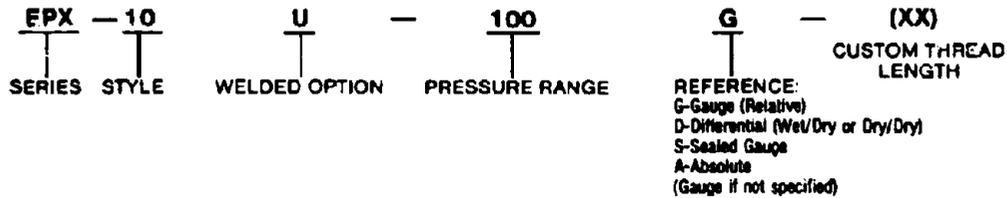
Examples:

EPX-10-50	Standard 10-32 UNF Thread, 50 PSIG range.	
EPX-M5-3.5	M5 Metric Thread, 3.5 bar range, relative (gauge).	
EPX-10U-100A-(.37)	Braze Welded 10-32 UNF Version with 100 PSIA range and custom thread length of "X" = 0.37".	
EPX-M5W-7-(10)	Beam Welded M5 Metric Thread with 7 bar range and custom thread length of "X" = 10mm.	
EPX6-10I-5000	EPX with 6V Excitation option on 10-32 UNF Threaded Housing with Internal Compensation Module, 5000 psig.	

ACTUAL SIZE

TYPICAL INSTALLATION

EPX-10 or EPX-M5

10-32 UNF
or
M5 METRIC

BUILT-IN
O-RING

10-32 UNF
or
M5 METRIC

REFERENCE
PRESSURE PORT

REFERENCE
PRESSURE PORT
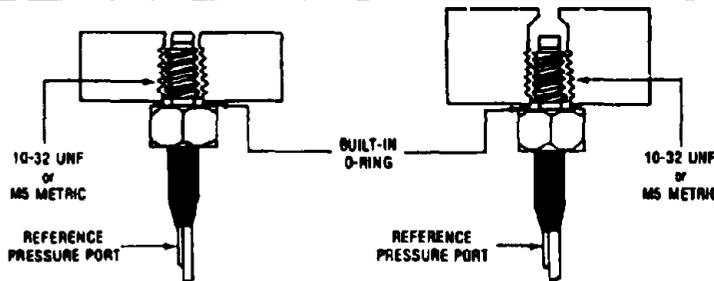
INSTALLATION TORQUE

Pressure Range	In-Lbs.	Meter-Newton
0 to 50 psi 0 to 3.5 bar	5	0.6
60 to 500 psi 4 to 35 bar	10	1.2
600 to 5000 psi 40 to 350 bar	15	1.8

Lamina

Medium Speed – High Torque – Miniature Size
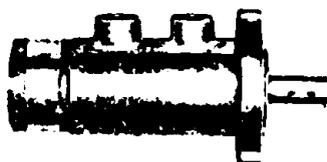
Built by Lamina under license from Char-Lynn, these motors use the Orbit principle to obtain reduced speed on the output shaft with increased torque. Best operation is in a moderate speed range with 4 to 6 GPM. Will run with equal torque in either direction and is instantly reversible.

Body length 5-1/8" on Model A-100; slightly shorter on other models. Front mounting flange 3" diameter. Four mounting holes 11/32" diameter spaced on a 5-hole pattern, with one hole omitted in line with the oil ports. Bolt circle 2.375" diameter. Shaft 5/8" diameter x 1-1/8" long with Woodruff keyway. Motor body diameter 1-3/4". Oil ports are straight thread with O-ring seal, 9/16-18. For foot mounted motors see listings below.

Motor shaft will tolerate full side loading but thrust loading should be kept as low as possible. Recommended fluid, Type A transmission fluid or hydraulic oil with viscosity not less than 100 SSU (at 100°F). Four displacements are offered in the chart below. Choose the one which most closely matches load speed and torque requirements.

Choose From Five Displacements

MODEL No.	A-25F	A-37F	A-50F	A-62F	A-100F
Displacement, Cubic Ins./Rev.	0.81	1.21	1.62	2.03	3.25
Maximum Oil Flow (Supply)	All models 8 GPM				
Maximum Operating Pressure	All Models 1500 PSI				
No Load RPM @ 8 GPM	2272	1517	1137	909	568
Full Load RPM @ 8 GPM	2107	1385	1038	843	519
No Load RPM @ 1 GPM	284	169	142	113	71
Full Load RPM @ 1 GPM*	74	21	16	29	8
Maximum HP @ 8 GPM*	All models 3 HP				
Maximum Torque, Inch-Lbs.*	100	150	200	250	400
Approximate Weight	All models 3¼ Lbs.				
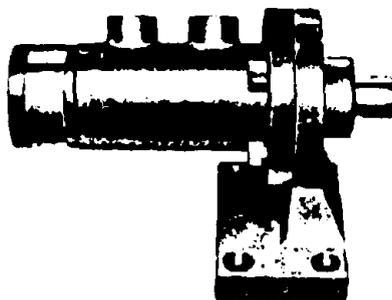
Foot Mounted Lamina Motors

To order above motors for foot mounting, add "M" after model number, as A-62FM, etc. Bracket must be specified at time of original order; it cannot be added to flange mounted motors in the field.

Centerline height 3¼"; Shaft extends 1¼" beyond bracket. The weight of motor and foot bracket 6¼ lbs.

ORBIT MOTOR PRINCIPLE

Lamina motors listed above and Char-Lynn ORBIT motors on Pages 96D to 96J obtain their desirable slow speed, high torque characteristic by operating on the ORBIT* principle. This gives a built-in speed reduction without the use of external gear boxes or speed reducers. The result is a slow shaft speed at a greater efficiency than could usually be produced by a high speed hydraulic motor operating through an external speed reducer.

*Note: ORBIT is a registered trademark of Char-Lynn Division, Eaton Corp.

MOOG
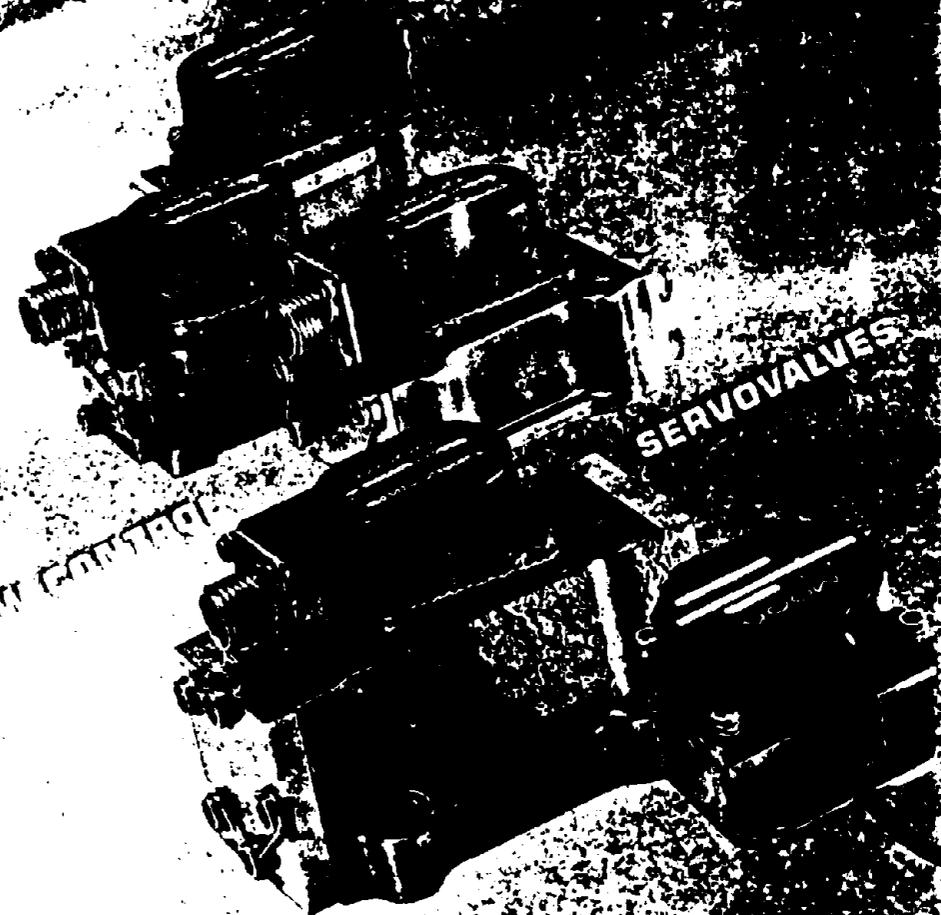
SERVOVALVES

TYPE 30 FLOW CONTROL

TYPE 30 SERVOVALVES

■ flow control ■ double nozzle
■ two stage ■ mechanical feedback

FIVE BASIC SIZES

		PORT CIRCLE DIAMETER		MAX. RATED FLOW		VALVE WEIGHT	
		in.	mm	gpm at 3000 psi	liters/min at 210 bars	lbs	kg
TYPE 30 SERVOVALVES	Series 30	0.480	12.19	3.1	12	0.42	0.19
	Series 31	0.625	15.88	6.8	26	0.81	0.37
	Series 32	0.780	19.81	14	54	0.81	0.37
	Series 34	0.780	19.81	19	73	1.10	0.50
	Series 35	1.000	25.40	44	170	2.13	0.97

THIS CATALOG CONTAINS

■ general information on Type 30 servovalves
■ information on standard valve designs

STANDARD DESIGNS

■ are assembled from standard parts
■ offer choice of

□ rated flow
◻ rated pressure
◻ rated current (coil resistance)
◻ internal coil connection
◻ electrical connector or cable
◻ connector or cable location
◻ seal compound

■ give standard performance (per Moog specification)
■ eliminate non-recurring start-up costs
■ minimize lead-time; certain models carried in stock

POSITION SERVO –
THE USUAL APPLICATION OF FLOW CONTROL SERVOVALVES
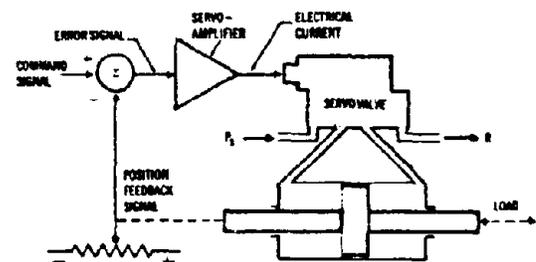
■ servovalve supplied with constant pressure P_s (e.g., 3000 psi or 210 bars)
■ servovalve controls flow to and from piston end chambers in response to electrical signal
■ piston drives load
■ position feedback signal obtained from pot, LVDT, DCDT, etc.
■ difference between position command signal and position feedback signal is error signal
■ error signal is amplified to drive servovalve
■ load moves to reduce error to near zero

MAGNET
MOTOR COIL
UPPER POLEPIECE
ARMATURE
FLEXURE TUBE
FLAPPER
NOZZLE
FEEDBACK WIRE
HYDRAULIC AMPLIFIER FILTER
SPOOL
BUSHING

DESIGN FEATURES

FLOW METERING SLOT
BUSHING
SPOOL
HYDRAULIC AMPLIFIER FILTER
INLET ORIFICE
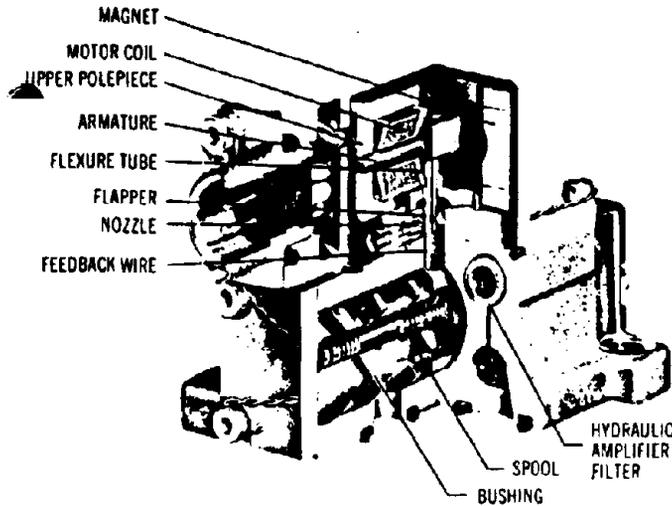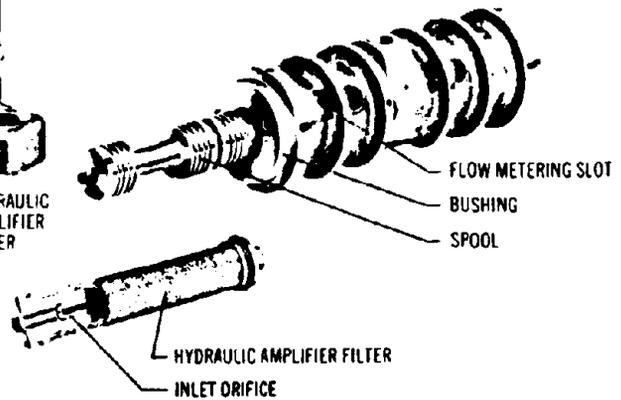
MATERIALS used in Standard Type 30 Servovalves

Body*, end caps and accessories	17-4 PH stainless steel
Spool and bushing*	440-C stainless steel
Filter (35 micron absolute)	Sintered stainless steel wire mesh
Flexure tube	Beryllium copper
Polepieces and armature	4750 Nickel-iron steel
Magnets	Alnico VI
Feedback wire	440-C stainless steel
Torque motor cover	Anodized aluminum alloy

▲The Series 30 has an integral bushing and body made from 440-C.

QUALITY CONTROL
- conforms to MIL-Q-9858A
- valves assembled and tested in clean room (per FED STD 209 Class 100,000)

ENVIRONMENTAL CAPABILITY OF STANDARD DESIGNS*
- meet or exceed MIL-V-27162 (SAE ARP 490C)
- suitable for Type I and II missile and aircraft hydraulic systems per MIL-H-8775C
  - ☐ Type I —65°F to +160°F (—54°C to +71°C)
  - ☐ Type II —65°F to +275°F (—54°C to +135°C)
- meet or exceed following as tested per MIL-STD-810B
  - ☐ high temperature — normal performance at +275°F fluid and ambient with MIL-H-5606 fluid
  - ☐ low temperature — normal performance at 0°F fluid and ambient with MIL-H-5606 fluid
  - ☐ extreme low temperature — valves will respond to input commands at —65°F fluid and ambient on MIL-H-5606 fluid
  - ☐ altitude — normal performance to 100,000 feet altitude
  - ☐ random vibration — will withstand 25 g rms (5 to 2000 Hz) 30 minutes per axis
  - ☐ sinusoidal vibration — will withstand sweep from 16 g at 25 Hz, to 35 g at 2000 Hz, 30 minutes per axis
  - ☐ acceleration — will withstand 50 g any axis
  - ☐ shock — will withstand 6 msec sawtooth, 100 g peak, any axis
  - ☐ salt spray, fungus, humidity, sand and dust — will withstand all exposures per MIL-STD-810B
  - ☐ useful life — >10 years with normal overhaul
  - ☐ cyclic life — >10⁸ cycles with normal wear

*Type 30 Servovalves are not necessarily limited by the environments listed.
Special designs are available that considerably extend these capabilities.

- rugged, stainless steel body
- one-piece bushing with EDM flow slots*
- bushing slip-fit in body bore
  - ☐ eliminates bushing land O-rings
  - ☐ bushing easily removed for cleaning or replacement
- O-ring sealed spool stops
  - ☐ eliminates pressure loading of ends of bushing
- spool bushing tolerances for diametral clearance held within 20 microinches (½ μm)
- 20 μm nominal filter (35 μm absolute) for pilot flow
- symmetrical, double nozzle hydraulic amplifier
  - ☐ provides consistent performance over wide temperature range
- hydraulic amplifier integrated into main valve body
  - ☐ eliminates several O-rings
- torque motor in environmentally sealed compartment
- frictionless, flexure tube supported armature/flapper
  - ☐ isolates hydraulic fluid from torque motor
- balanced, double coil, double air gap torque motor
  - ☐ reduces temperature centershift
  - ☐ minimizes external magnetic fields
  - ☐ reduces sensitivity to external magnetic materials or fields
- motor coils have resilient potting
  - ☐ cushions coils during thermal and vibration extremes
- mechanical feedback with simple cantilever spring
  - ☐ rolling ball contact with spool minimizes wear
  - ☐ feedback removable without damage to valve
- locating pin in base of valve prevents improper installation

*EDM = electric discharge machined
Series 30 does not have a bushing (slots are EDM'd in valve body)

Design features of Type 30 Servovalves are covered by U. S. Patents 3,023,782; 3,228,423 and 4,017,706 together with corresponding patents in several foreign countries

SPECIAL DESIGNS
major programs usually warrant a special servovalve design

■ special Type 30 valves are designed to satisfy customer specification
  □ provides optimum servovalve configuration
  □ allows customer configuration control

■ may be
  □ special performance
  □ special environments
  □ special testing
  □ special bias
  □ special quality control
  □ special mounting
  □ special porting
  □ special connector
  □ special materials
  □ special handling

■ any change from standard is a special

■ special designs carry a unique model number, parts list, test procedures, etc.

■ special Type 30 designs that require a non-standard manifold pattern will be assigned a Series 33 model number

UNUSUAL ENVIRONMENTS

● high temperature — to 400°F (200°C) fluid and ambient with Viton seals (limited life)
  — to 650°F (350°C) fluid and ambient with metallic seals, ceramic insulated magnet wire, and special magnetic materials

■ high accelerations — to 400 g with special mass balanced armature/flapper and stainless steel flexure tube

■ high vibration — to 100 g rms (20-2000 Hz) with silicon damping fluid and armature motion restraint

■ high shock — to 5000 g with damping fluid and stainless steel flexure tube

■ nuclear radiation — to 2 x 10⁶ rads (c) with standard teflon insulated magnet wire; higher radiation levels with ceramic insulation and metallic seals

■ external pressurization — to hundreds of psi with special motor cap or with motor cavity vented to pressure equalized return
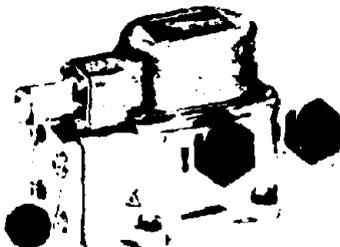
SPECIAL 33 SERIES FOR SHORT DUTY DURING HIGH ACCELERATION
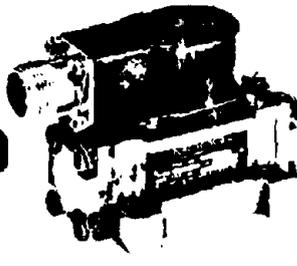
SPECIAL 32 SERIES FOR OPERATION TO 600°F. OIL AND AMBIENT

SPECIAL BODY CONFIGURATIONS

■ diamond port pattern Series 30

■ plug-in electrical connector mounted flush with manifold surface

■ plug-in hydraulic manifold with annular hydraulic ports

SPECIAL 33 SERIES WITH PLUG-IN MANIFOLD AND THREE-WAY SPOOL
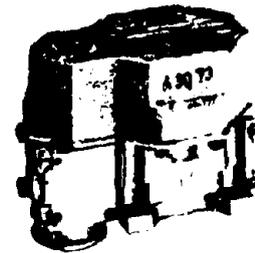
SPECIAL 30 SERIES WITH PLUG-IN ELECTRICAL CONNECTOR

SPECIAL FLUIDS

■ other hydraulic fluids including water

■ most fuels, propellants, and oxidizers

■ pneumatics

SPECIAL SPOOL DESIGNS

■ three-way spools having single control port

■ spool stops for limiting maximum flow
  □ limit flow usually held to ±10%

■ special spool null cuts
  □ prescribed amounts of underlap or overlap, symmetrical or unsymmetrical

■ graphite impregnation of spool for pneumatic and water applications

■ non-linear slot width
  □ different flow gain for each valve polarity as used with some three-way actuators
  □ stepped width slots for dual flow gain

OPERATION

UPPER POLEPIECE
ARMATURE
PERMANENT MAGNET
LOWER POLEPIECE
COIL
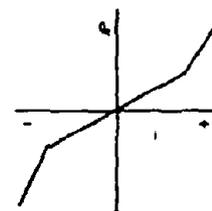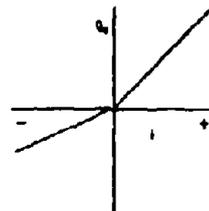
TORQUE MOTOR

- charged permanent magnets polarize polepieces
- dc current in coils causes increased force in diagonally opposite air gaps
- magnetic charge level sets magnitude of decentering force gradient on armature

TORQUE TO ROTATE ARMATURE

PERMANENT MAGNET FLUX

PERMANENT MAGNET ATTRACTIVE FORCE

COIL FLUX
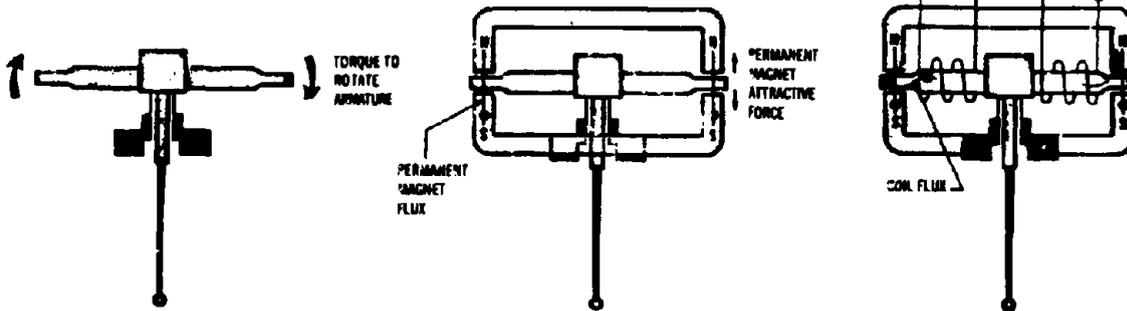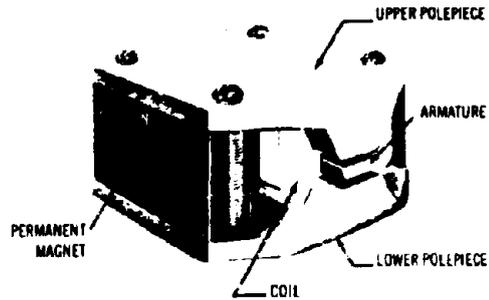
HYDRAULIC AMPLIFIER

- armature and flapper rigidly joined and supported by thin-wall flexure tube
- fluid continuously flows from pressure P_s, through both inlet orifices, past nozzles into flapper chamber, through drain orifice to return R
- rocking motion of armature/flapper throttles flow through one nozzle or the other
- this diverts flow to A or B (or builds up pressure if A and B are blocked)

ARMATURE
FLAPPER
INLET ORIFICE
FLEXURE TUBE
NOZZLE
DRAIN ORIFICE
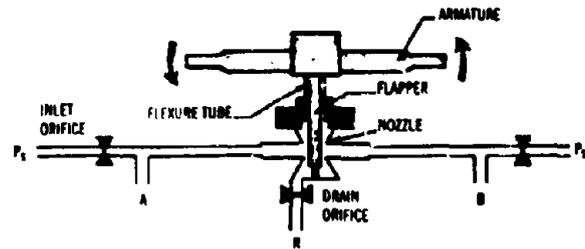
VALVE SPOOL

- spool slides in bushing (sleeve), or directly in a body bore for Series 30
- bushing contains rectangular holes (slots) or annular grooves that connect to supply pressure P_s and return R
- at "null" spool is centered in bushing; spool lobes (lands) just cover P_s and R openings
- spool motion to either side of null allows fluid to flow from P_s to one control port, and from other control port to R

SPOOL AT NULL

FEEDBACK SPRING
SPOOL
BUSHING

C2
C1

SPOOL DISPLACED TO LEFT

C2
C1
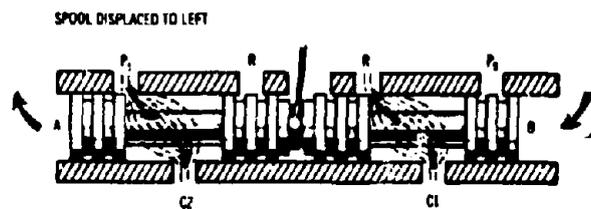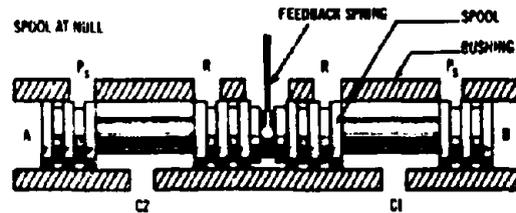
OPERATION

- electrical current in torque motor coils creates magnetic forces on ends of armature

- armature and flapper assembly rotates about flexure tube support

- flapper closes off one nozzle and diverts flow to that end of spool

- spool moves and opens P_S to one control port; opens other control port to R

VALVE RESPONDING TO CHANGE IN ELECTRICAL INPUT
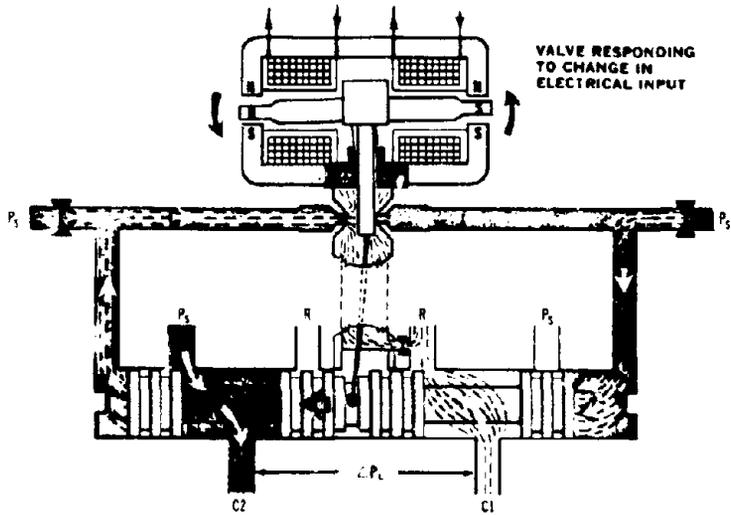
- spool pushes ball end of feedback spring creating restoring torque on armature/ flapper

- as feedback torque becomes equal to torque from magnetic forces, armature/flapper moves back to centered position

- spool stops at a position where feedback spring torque equals torque due to input current

- therefore spool position is proportional to input current

- with constant pressures, flow to load is proportional to spool position

VALVE CONDITION FOLLOWING CHANGE

FLOW TO ACTUATOR
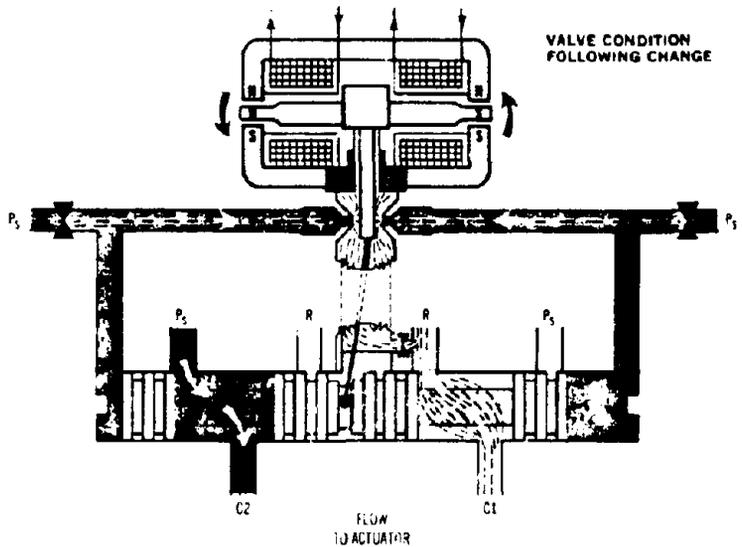
TYPICAL TYPE 30 SERVOVALVE SIZES

	Thousandths of an inch	Micro meters
air gap spacing (each gap)	10 to 15	250 to 380
maximum armature motion in air gap	±3	±75
maximum torque on armature	±0.3 in-lbs	±0.03 N·m
inlet orifice diameter	5 to 10	125 to 250
nozzle diameter	10 to 20	250 to 500
nozzle flapper maximum opening	2.4 to 3.0	60 to 75
drain orifice diameter	10 to 15	250 to 380
spool stroke	±10 to ±20	±250 to ±500
spool/bushing radial clearance	0.03 to 0.06	0.7 to 1.5
bushing/body radial clearance	0.08	2.0

ELECTRICAL CHARACTERISTICS

STANDARD COIL CONFIGURATIONS

CODE FOR PART NUMBER OF STANDARD VALVE	P	S	D	I
INTERNAL COIL CONFIGURATION	PARALLEL COILS	SERIES COILS	DIFFERENTIAL COILS	INDIVIDUAL COILS
PINS (IF CONNECTOR)	B A	B A	B A C	B A D C
COLORS (IF CABLE)	grn red	grn red	grn red blu	grn red yel blu
SERIES COILS	Not possible	B+ A—	B— C—	B+ Tie A,D C—
PARALLEL COILS	B+ A—	Not possible	Not possible	Tie B,D+ Tie A,C —
DIFFERENTIAL COILS	Not possible	Not possible	for A+ when current A to B <A to C for A— when current B to A>C to A	Tie A,D for A,D+ when current A to B<D to C for A,D— when current B to A>C to D
SINGLE COIL	Not possible	Not possible	B+ A— or A+ C—	B+ A— or D+ C—

ELECTRICAL CONNECTOR

- standard configurations (see table below)
  - ☐ Bendix pigmy
  - ☐ 18 inch long cable

- special Type 30 connectors
  - ☐ connector type and location per specification
  - ☐ flush manifold plug-in connector available

COIL CONFIGURATION AVAILABLE	CONNECTOR TYPE	CODE FOR PART NUMBER
P. S, D, I	*PC-02E-8-4P 4 pin screw	4 PC
P, S, D	*PC-02E-8-3P 3 pin screw	3 PC
P, S, D, I	*PT-02E-8-4P 4 pin bayonet	4 PT
P, S, D	*PT-02E-8-3P 3 pin bayonet	3 PT
I	4 wire cable, 18" long	**4 CA
D	3 wire cable, 13" long	3 CA
P, S	2 wire cable, 18" long	2 CA

*Bendix Corporation part number
**only choice for Series 30

DITHER

- servovalve performance normally measured without dither
  - ☐ dither current may be applied to Type 30 Servovalves
  - ☐ usually will improve servovalve and actuator threshold
  - ☐ will increase spool null leakage

- dither characteristics
  - ☐ usually 100 to 400 Hz (selected to suit system)
  - ☐ peak-to-peak dither amplitude may be as high as ±20% servovalve rated current without degradation of valve life
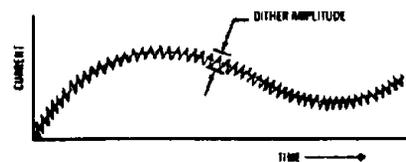
RATED CURRENT

■ choice of coil resistance and coil connections determine valve rated current (see table below)

■ other coil resistance and rated current combinations can be supplied for special valves
  □ lower rated current can be specified for standard coils, but with corresponding degradation of valve performance

■ triple rated current can be supplied indefinitely with no damage to servovalve

COIL IMPEDANCE

■ composed of
  □ dc coil resistance
  □ ac coil resistance
  □ apparent coil inductance

■ dc coil resistance
  □ nominally equal for both coils, but may vary ±10% as coils are wound for desired number of turns
  □ will vary with temperature (approximately 0.002 ohms/ohm °F)

■ ac coil resistance
  □ represents work done in moving armature
  □ becomes significant above 200 Hz

■ coil apparent inductance
  □ includes coil self inductance plus mutual inductance of other coil
    — mutual coupling of coils is approximately 50%
  □ will vary considerably with motion of armature (back emf's)
    — affected by valve supply pressure, signal amplitude, and signal frequency
    — may become capacitive at higher frequencies
    — usually specified at 50 Hz with normal operating conditions
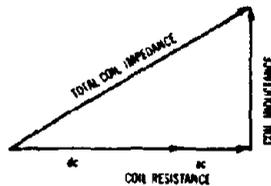
QUIESCENT CURRENT

■ may be present with push-pull operation of three and four wire coil configurations
  □ signal input current $i = i_1 - i_2$
  □ quiescent current $i_Q = \dfrac{i_1 + i_2}{2}$
    when $i_1 = i_2$
  □ quiescent current i_Q should be $i_R > i_Q > \dfrac{i_R}{2}$

  □ small null shift and gain change may occur with changes in quiescent current amplitude and polarity

SERVOAMPLIFIER

MODEL 82-300 SERVOCONTROLLER

■ should provide dc current into torque motor coils
  □ irrespective of coil inductance and resistance
  □ requires current feedback amplifier
  □ large shunt capacitance at output of amplifier may produce undesirable resonance with servovalve coil impedance

■ current feedback amplifier
  □ eliminates apparent servovalve gain change due to changes in coil impedance
  □ minimizes phase lags due to coil inductance

■ standard servovalve drive amplifiers available from Moog
  □ rack mounted laboratory units
  □ low cost industrial units
  □ aerospace units on special order

	P Parallel Coils			S Series Coils			D Differential Coils			I Individual Coils		
	Ohms	i_R	Ma.	Ohms		Ma.	Ohms		Ma.	Ohms		Ma.
0040	20	0.10	50	80	0.36	25	40	0.19	50	40	0.12	50
0080	40	0.18	40	160	0.66	20	80	0.34	40	80	0.22	40
0130	65	0.30	30	260	1.1	15	130	0.58	30	130	0.37	30
0200	100	0.59	20	400	2.2	10	200	1.1	20	200	0.72	20
0500	250	1.1	15	1000	4.1	7.5	500	2.1	15	500	1.3	15
1000	500	2.6	10	2000	9.7	5	1000	5.0	10	1000	3.2	10
1500	750	3.4	8	3000	12.5	4	1500	6.4	8	1500	4.1	8

NOTE: Resistance values at 68°F (20°C) ± 10% tolerance
  Inductance values are typical for 50 Hz, servovalve pressurized. Inductance is not normally measured on individual servovalves.
  *Inductance values per coil with differential operation (Class A push-pull).

HYDRAULIC CHARACTERISTICS

SUPPLY PRESSURE

■ 500 psi to 4000 psi for standard designs
  □ valves are set up and tested at supply pressure specified
  □ valves can be used at other supply pressures, but some null shift may occur

■ valves supplied for pressures below 500 psi should be specially designed
  □ Type 30 Servovalves are usable with supply pressures as low as 50 psi
  □ servovalve performance, especially threshold and dynamic response, is degraded with low supply pressure

PROOF AND BURST PRESSURES

■ proof pressure capability:
  □ at supply and control ports $= 1.5\ P_s$
  □ at return port $= 1.0\ P_s$

■ burst pressure capability:
  □ at supply and control ports $= 2.5\ P_s$
  □ at return port $= 1.5\ P_s$ or 5000 psi maximum

RETURN PRESSURE

■ may vary widely with minimal valve null shift

■ should never exceed supply pressure to avoid back flowing hydraulic amplifier

RATED FLOW

■ each valve Series covers a range of rated no-load flow to the maximum shown (for MIL-H-5606)

NO-LOAD RATED FLOW CIS / RATED VALVE PRESSURE DROP PSI
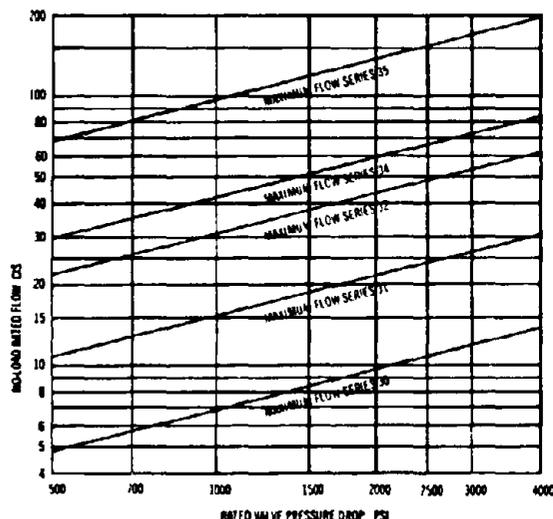
STANDARD SEALS, FLUIDS, AND TEMPERATURES

O-RING ELAS-TOMER	RECOMMENDED FLUIDS	RECOM-MENDED TEMP. RANGE*	TEST FLUID	CODE FOR PART NO.
Buna N	Petroleum Base Fluids such as MIL-H-5606, MIL-H-6083, DTE, Regal, Brayco MIL-H-83283 (synthetic hydrocarbon) Freon 12, 13, 14, 113, 114 Silicone Fluids	—65°F to +275°F	MIL-H-5606	BUN
Fluorocarbon Rubber (Viton)	Petroleum Base Fluids such as Type A Transmission Fluid, JP-4, JP-5 Superrefined Mineral Oils Silicone Fluids Silicate Ester Fluids such as MIL-H-8446, MLO-8200, OS-45, M2V Industrial Phosphate Ester Fluids such as Cellulube, Pydraul, Pyroguard Di-Ester Base Fluids such as MIL-L-7808, Houghton Safe Tri-Ester Base Fluids such as Trichloroethylene	—20°F to +400°F	MIL-H-5606	VIT
Ethylene Propylene Rubber	Aircraft Phosphate Ester Fluids such as Skydrol, Hyjet, Aerosafe Freon 22, 31, 32, 115 Hydrazine** UDMH Water, steam, air	—65°F to +300°F	Hyjet IV	EPR

* Operating temperature range may be further restricted by fluid.
** Standard Type 30 Servovalves are recommended for short term use with this fluid. Special designs with all stainless steel wetted parts are available.

FLOW-LOAD CHARACTERISTICS
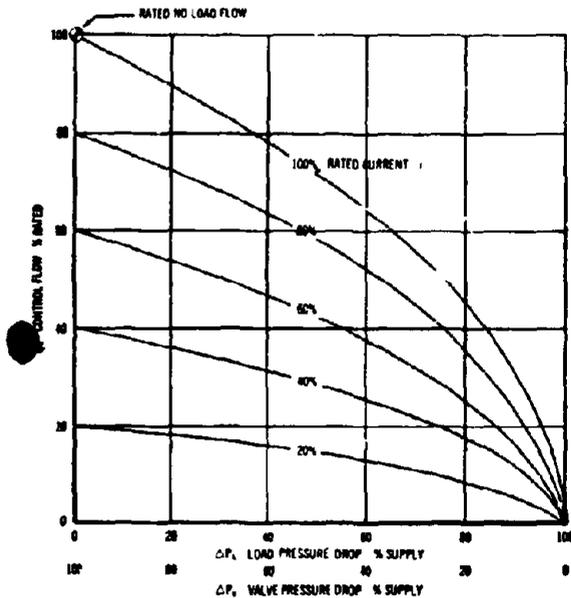
- nominal flow to load

$$Q_v = K i \sqrt{\Delta P_v}$$

where Q_v = valve flow to load
           K = servovalve sizing factor
           i = input current
           ΔP_v = valve pressure drop

$$\Delta P_v = (P_s - R) - \Delta P_L$$

where P_s = supply pressure
           R = return pressure
           ΔP_L = load pressure drop

- some flow saturation will occur with servovalves having maximum flow capacity
  - ☐ saturation causes droop at the high end of the flow curve
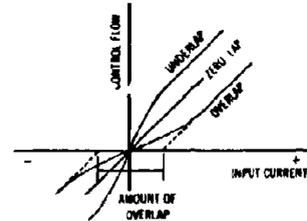
SPOOL LAP

- standard servovalves have zero lap within limits of flow linearity shown on Page 12

- prescribed amounts of under or overlap can be specified on special order
  - ☐ underlap (or open center)
    - — increases flow gain at null
    - — reduces valve pressure gain at null
    - — increases valve null leakage
  - ☐ overlap
    - — reduces flow gain at null
    - — reduces null leakage flow
    - — degrades pressure gain (into a load)
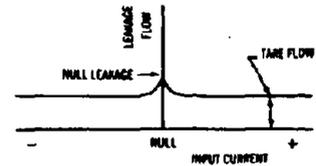
INTERNAL LEAKAGE

- includes first stage hydraulic amplifier flow, spool null leakage flow, and bushing laminar leakage flow
  - ☐ spool null leakage flow is essentially zero when spool is off-null
  - ☐ servovalve internal leakage excluding spool null leakage is called **Tare Flow**

- hydraulic amplifier flow largely determines servovalve frequency response
  - ☐ lower flow degrades response

- spool null leakage flow is related to maximum valve flow (slot width) and null cut
  - ☐ table gives internal leakage of standard Type 30 Servovalves (with MIL-H-5606)
  - ☐ special low leakage versions of the miniature Series 30 available with <0.25 cis at 3000 psi

MAXIMUM LEAKAGE OF STANDARD TYPE 30 SERVOVALVES

VALVE SERIES	TARE LEAKAGE FLOW CIS at 1000 psi	at 3000 psi	SPOOL NULL LEAKAGE FLOW (% rated flow at rated pressure)
30	< 0.20	< 0.35	< 4
31	< 0.25	< 0.45	< 4
32	< 0.28	< 0.50	< 3
34	< 0.35	< 0.60	< 3
35	< 0.45	< 0.75	< 3

STATIC PERFORMANCE

CONTROL FLOW

■ servovalve control flow to the load is nominally proportional to electrical input current
  ☐ standard production acceptance test limits for no load flow shown below
  ☐ limits do not include servovalve hysteresis or null bias
  ☐ limits at ±100% input for maximum flow designs may be +10, −20% due to non-linearities caused by flow saturation
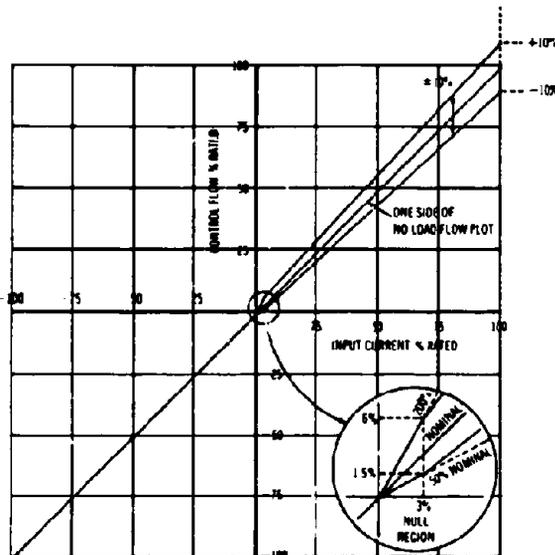
■ control flow non-linearity is greatest in null region
  ☐ may be from 50% to 200% nominal gain within range of of ±3% electrical input for standard null cut
  ☐ can be held to closer limits on special order

■ maximum valve flow may be 120 to 140% rated flow with oversignal
  ☐ spool stops to limit maximum flow can be provided on special order

■ control flow characteristic may change with fluid temperature
  ☐ a +100°F temperature rise may cause control flow to increase as much as 3% due to fluid viscosity effects
  ☐ at very high temperatures (over 400°F) a +100° temperature rise may cause control flow to decrease by 3% due to magnetic effects

PRESSURE GAIN
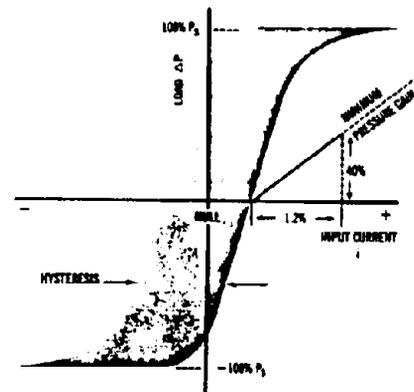
■ blocked load ΔP_L changes rapidly from $-P_s$ to $+P_s$ in null region
  ☐ minimum pressure gain will be 0.4 P_s/1.2%i_R for standard servovalves
  ☐ maximum pressure gain may be three times higher
  ☐ pressure gain will degrade with spool null edge wear

■ special pressure gain requirements may interact with desired flow gain at null, spool null leakage, and nominal control port pressures at null

HYSTERESIS

■ maximum hysteresis for standard Type 30 Servovalves with normal operation conditions is <3%
  ☐ hysteresis may increase to 4% at −30°F
  ☐ hysteresis limit for special high temperature servovalves (>400°F) is <4%

THRESHOLD

■ maximum threshold for standard Type 30 Servovalves with normal operating conditions is <½% (without dither and with supply pressure greater than 1000 psi)
  ☐ with P_s below 1000 psi threshold limit is <1%
  ☐ threshold limit should be doubled at −30°F
  ☐ with dither, threshold→0%
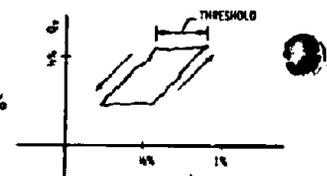
NULL BIAS
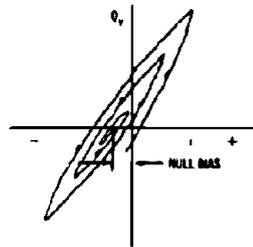
- electrical input current to give valve null includes both temporary null shifts and permanent changes in null bias

- null bias is measured under standard valve operating conditions (pressures, temperatures, etc.)

- null bias measurements exclude valve hysteresis

- initial servovalve null bias on standard valves (as shipped) is less than 2% rated input

- long-term null bias after exposure to environments and use can be expected to be < 5%

NULL SHIFT

- change in null bias with environment and operating conditions will vary from unit to unit, but is generally:

	NULL SHIFT
TEMPERATURE	
50°F to 150°F	< 2%
0°F to 200°F	< 4%
ACCELERATION	
TO 40 G SPOOL AXIS	< 0.3%/G
TO 40 G TRANSVERSE AXIS	< 0.1%/G
SUPPLY PRESSURE	
60% TO 110%	< 4%
QUIESCENT CURRENT	
50% TO 200% RATED CURRENT	< 6%
BACK PRESSURE	
0% TO 20% OF SUPPLY	< 4%

- special mass balanced torque motor design available for < 0.06%/g to 400 g

- null shifts are not normally measured during production acceptance testing

- tighter null shift specifications can be imposed by providing 100% valve testing under critical environment

SPOOL DRIVING FORCE

- hydraulically created force to move spool for standard Type 30 Servovalves is >1 lb/% input (with 3000 psi supply)
  - □ spool driving force gradient will be >0.6 lb/% input for operation at 1000 psi supply

- maximum spool driving force for standard servovalves at 3000 psi supply is

SERIES	MAX. SPOOL FORCE (POUNDS)
30	55
31	55
32	110
34	140
35	160

- special servovalves with lower hydraulic amplifier flow will have higher spool driving force gradients (but lower dynamic response)
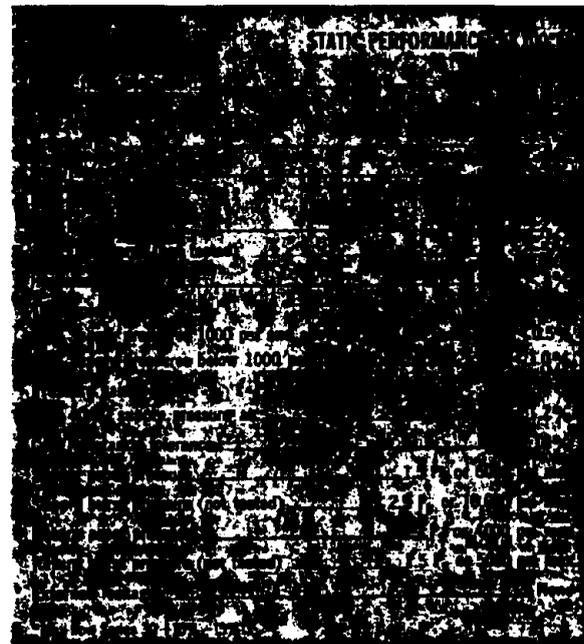
SUMMARY

DYNAMIC PERFORMANCE

FREQUENCY RESPONSE

- will depend upon signal amplitude, supply pressure, and internal design configuration

- plot below shows typical responses for standard Type 30 Servovalves
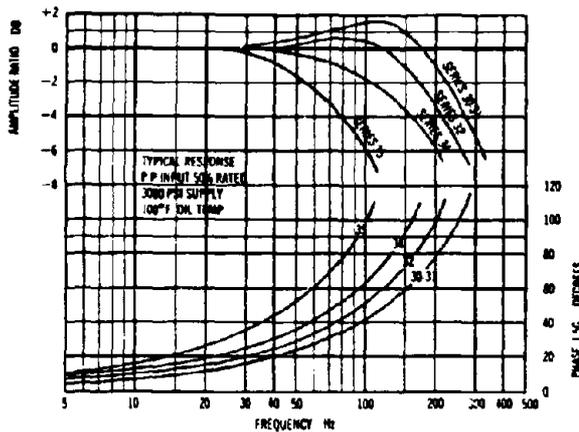
- for system design these characteristics can be approximated by

Valve Series	Equivalent First Order Time Constant sec.	Equivalent Second Order Natural Frequency Hz	Damping Ratio
30	0.0015	200	0.5
31	0.0015	200	0.5
32	0.0020	160	0.55
34	0.0029	110	0.6
35	0.0035	90	0.9

- frequency response of specially designed valves can be improved by
  - ☐ increased hydraulic amplifier leakage flow
  - ☐ shorter spool stroke (larger slot width)
  - ☐ use of stub shafts on spool ends
  - ☐ higher rated current (stiffer feedback wire)

STEP RESPONSE

- time response to step input of current depends on valve design parameters

- approximate transient response of standard Type 30 Servovalves operating at 3000 psi is

Valve Series	Approximate Response Time to 90% Output sec.
30	0.0025
31	0.0075
32	0.0045
34	0.007
35	0.012

INTERNAL DYNAMICS OF TYPE 30 SERVOVALVES

See Technical Bulletin 103 for a discussion of servovalve dynamic characteristics and response measuring techniques.
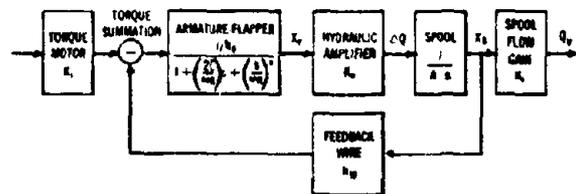
TYPICAL PARAMETERS FOR SERIES 31*

I = torque motor current .... ± 10 ma

x_f = flapper displacement at nozzles .... ± 0.0012 in max.

x_s = spool displacement .... ± 0.015 in max.

$\triangle Q$ = hydraulic amplifier differential flow ...... ± 0.18 in^3/sec

Q_v = servovalve control flow .... ± 4 gpm

K_1 = torque motor gain .... 0.025 in-lbs/ma

K_2 = hydraulic amplifier flow gain .... $150 \frac{in^3/sec}{in}$

K_3 = flow gain of spool/bushing .... $1030 \frac{in^3/sec}{in}$

A = spool end area .... 0.026 in^2

k_f = net stiffness on armature/flapper ..... 115 in-lbs/in

k_w = feedback wire stiffness .... 16.7 in-lbs/in

b_f = net damping on armature/flapper ..... $0.016 \frac{in-lbs}{in/sec}$

I_f = rotational mass of armature/flapper ..... $4.4 \times 10^{-6} \frac{in-lbs}{in/sec^2}$

$\omega_N = \sqrt{\frac{k_f}{I_f}}$ natural frequency of first stage ..... 814 Hz

$\zeta = \frac{1}{2}\frac{b_f}{k_f}\omega_n$ damping ratio of first stage 0.4

$K_v = \frac{K_2}{k_f}\frac{k_w}{A}$ servovalve loop gain ..... 840 sec^{-1}

* Consult factory for parameters of other series valves.
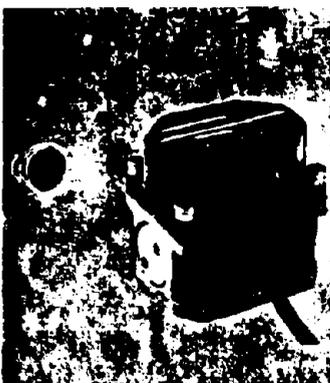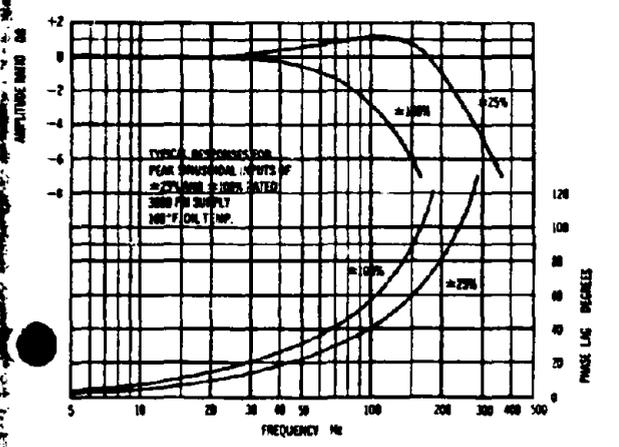
14

STANDARD SERIES 30 SERVOVALVES

- specify rated control flow in cu within limits of...
  - use two digits and decimal point as indicated
  - specified flow will be provided for test fluid (see page 10)
  - lower rated flows available on special order
- specify supply pressure from 500 to 4000 psi
  - lower and higher pressures available on special...

INSTALLATION DETAILS

Weight ..... 0.42 lbs. (0.19 kg)

NOTE: CABLE SHOWN OVER RETURN PORT (CODE R)
DIRECTION CAN BE OVER PRESSURE PORT (CODE P)

NAME PLATE

0.57 (14.5) MAX
0.88 (23.4)
0.80 (20.3) MAX
0.80 (20.3) MAX

1.54 (39.1) MAX

0.125 (3.18)
CABLE, 18 INCH (45 CM) MINIMUM LENGTH

0.438 (23.63)
0.469 (11.91)

0.87 (22.1) MAX RAD
CONTROL PORT C1
RETURN PORT
0.230 (6.10)

PORTS 0.160 (4.06) MAX
C BORED 0.200 I D x 0.310 O x 0.030 DEEP
(5.08 x 7.87 x 0.76)
0.516 (13.1)
1.032 (26.21)
0.56 (16.8) MAX
0.56 (16.8) MAX

0.152 (3.86) MA 4 MOUNTING HOLES
PRESSURE PORT
0.240 (6.10)
0.240 (6.10)
0.240 (6.10)
CONTROL PORT C2

Dimensions in parentheses are in millimeters

Typical frequency response for Standard Series 30 Servovalves shown below

AMPLITUDE RATIO DB
+2
0
-2
-4
-6
-8

TYPICAL RESPONSE FOR PEAK SINUSOIDAL INPUTS OF ±25% AND ±100% RATED
3000 PSI SUPPLY
100°F OIL TEMP.

±100%
±25%
±100%
±25%

120
100
80
60
40
20
PHASE LAG DEGREES

5   10   20   30 40 50   100   200 300 400 500
FREQUENCY Hz

500	6.5	4.5
1000		3.5
1500		
2000		

MOOG

F120

DESCRIPTION

The MOOG rack mount module cage F127-101, is especially designed to accomodate the MOOG plugboards F120. The case will fit the standard 19'' équipment racks or cabinets.

The cages are completely assembled with card guides installed for ten modules ; one location reserved for the power supply unit.

The motherboard provides large buses for the supply voltages and GND,

The pin of each receptacle is also wired to a terminal strip to allow easy access for external wiring.

The MOOG modules allow access to all controls and test points on the front panel of each module.

SPECIFICATIONS

- standard 19'' a rack mount module cage
- 1 plugboard location reserved for power supply (DIN 41612 style H)
- 9 plugboard locations (DIN 41612 style C)
- terminal strip for main power supply connection (220/110 VAC)
- terminal strip for output voltage of power supply ± 15 V, ± 24 V, and 5 V (option)
- 9 (32 pins) terminal strips, one for each plugboard location
- any supply failure triggers a relay
- contact rating : 0,25 A/24 VDC
- weight : 2,5 kg

MOTHER BOARD FRAME   F127-101

INSTALLATION

Side view

250

40   170
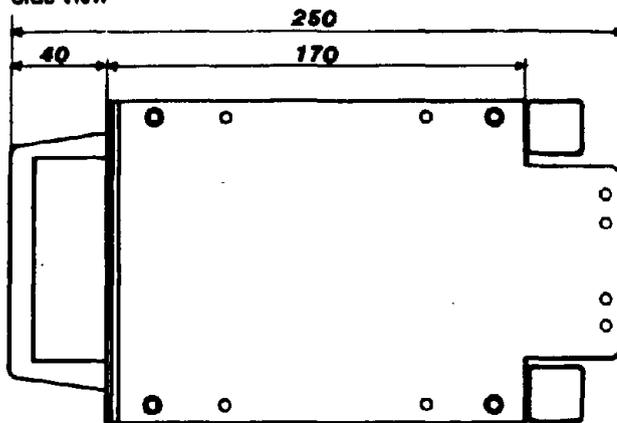
Dimensions in millimeters

Front view

483

465

MOOG

– ∞ + 24V
– ∞ + 15V

+ 24V
+ 15V
– 24V
GND
ON
OFF
– 15V

F 128-201

132,5

67,1

37,5
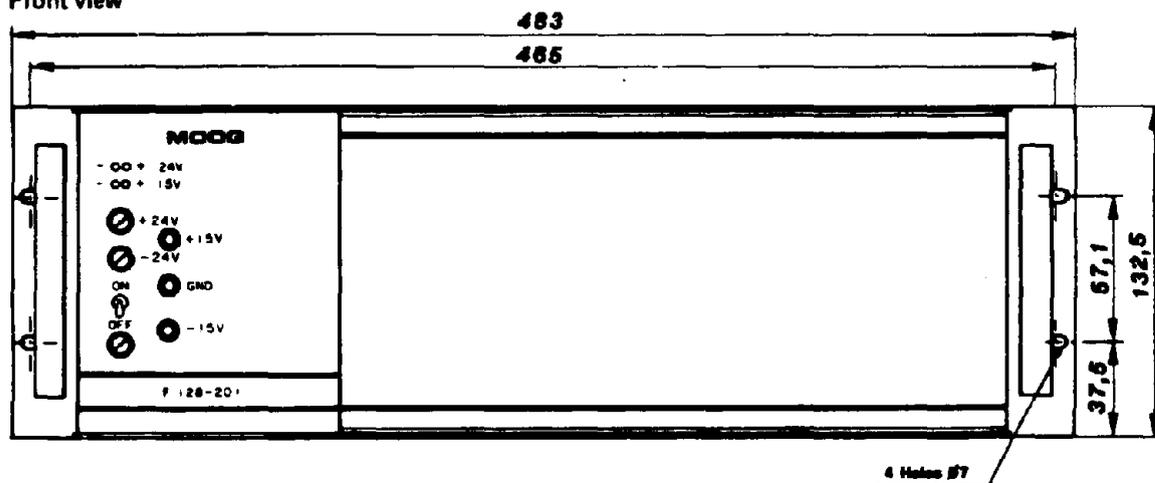
4 Holes Ø7

CONNECTING

Rear view
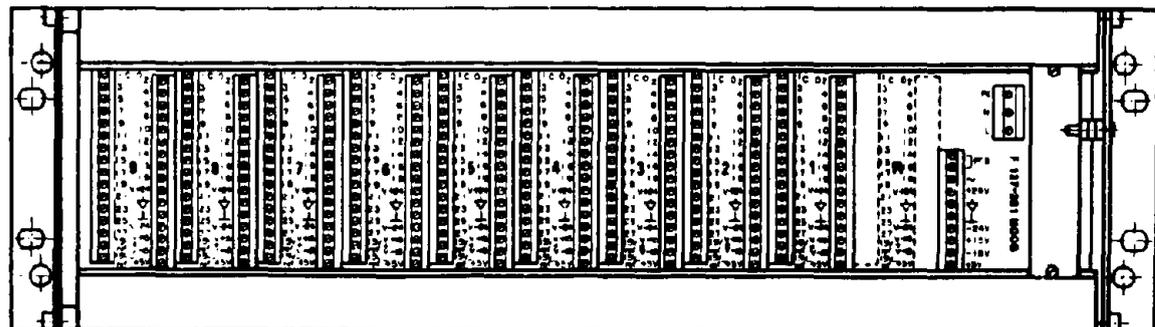
DESCRIPTION

The stabilized, IC regulated power supply F128-201 was designed to operate the cards of the series F120. The entire power supply conveniently mounts in the chassis F127-101. Front panel controls include ON/OFF switch, jacks to measure the regulated 15 V and 5 V (option) outputs and LED indicators to show the presence of all voltages.

SPECIFICATIONS

Input voltage range	: 210-230 VAC/50 Hz
	option 105-115 VAC/60 Hz
Output voltage	: + 15 VDC stab : 1.0 A max
	− 15 VDC stab : 1.0 A max
	+ 24 VDC not stab : 1.2 A max
	− 24 VDC not stab : 1.2 A max
	+ 5 VDC stab : 1 A max. (optional)
	note : the total source current of
	15 and 24 volt supplys may
	not exceed 1.2 Amp.
24 VDC fuses	:
Primary fuse	: 1 Amp (slow blowing)
Power failure detection	: 15 V or 5 V supply failure triggers a
	relay contact rating : 0,25 A/24 VDC
Weight	: 1,7 kg
Test points	: jacks provided on front panel for
	GND, + 15 VDC, − 15 VDC,
	(and 5 VDC) outputs.
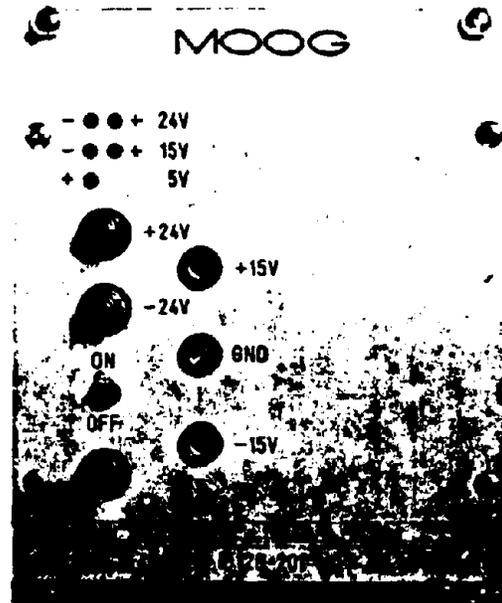
BLOCK DIAGRAM
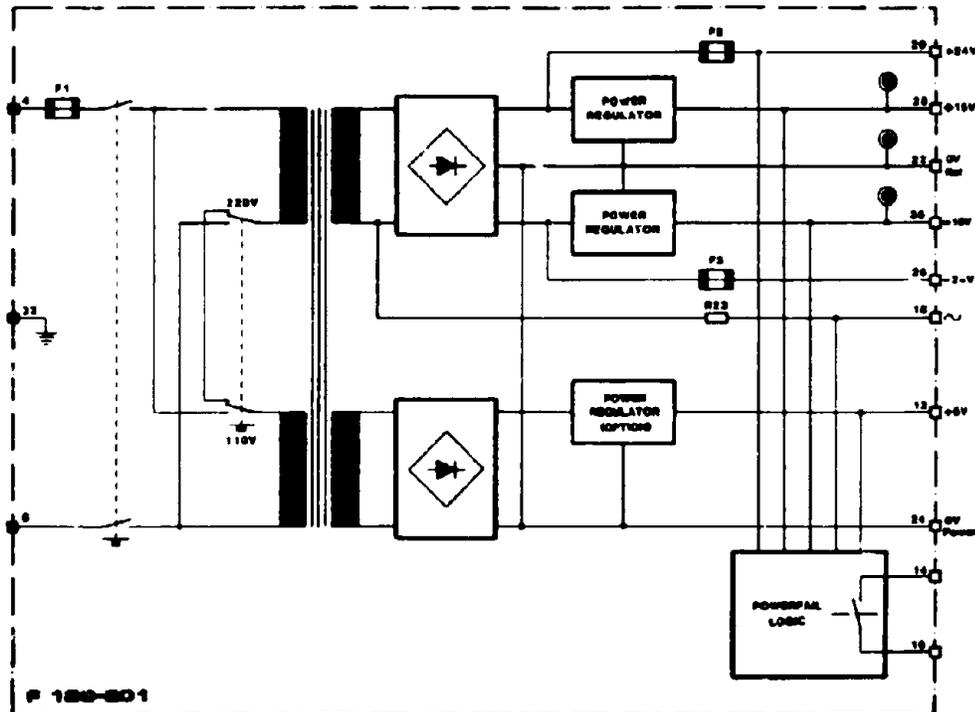
DESCRIPTION

The servoamplifier card F122-202, is a differential amplifier with PID control action, which supplies a servovalve with a current proportional to the difference of control and feedback voltage.

Controls to adjust amplifier gain, input sensitivity, null offset as well as test points for input and output signals are located on the front panel to allow fast and easy set up of an electrohydraulic control system.

SPECIFICATIONS

power supply ±15 VDC	input sensitivity : signal Vin 7 + P7
±24 VDC	signal VR7 + P9 (0 … 100%)
input voltage range : pin 3	zero offset : P1 (± 750 mV)
pin 9	proportional gain : P2 (fine) R6 (coarse)
output capability : max current	differential gain : P6, P8
max voltage	integral gain : P5
proportinal gain 5 to 200 mA/V	dither : amplitude P4
linearity ± 3% of full	frequency P3
temp. coefficient 0.1 mV/deg	
between 10 to 50	test points :
(input resistance and gain of 50 mA)	input voltage : signal of terminal 3 at jack Vin 3
freq. reponse - 3 dB	signal of terminal 7 at jack Vin 7
with inductive load	signal of terminal 9 at jack VR7
dither : amplitude 0 to 20 mV	(signal taken after potentiometer divider)
frequency	voltage image of
External control relays	servovalve current : jack Isv (voltage drop accross
consumption 15 mA/24 VDC	R 31 proportional to servovalve
contact rating 2 amp/24 VDC	current, R 31 = 20 Ohm ± 1%)

USE

The servoamplifier F122-202 is used in electrohydraulic control devices to directly drive a servovalve. This card can accomodate servovalves of up to 50 mA rated current.

FUNCTION

The servoamplifier F122-202 consists of :
- a comparator with adjustable gain
- three amplifiers, comprising PID control action with gain of each amplifier independend of the others
- an output driver stage including an operational amplifier and two power transistors (this output stage employs current feedback to reduce the effects of coil impedance variations due to temperature).

INITIAL INSTALLATION

The setpoint voltage is applied to pin 3 of the servoamplifier. This voltage can be taken from a potentiometer supplied by ± 15 V or by any type of voltage generator as eg. the card F124-202.

The feedback voltage is then applied to pin 7. This voltage is generally derived from some sort of transducer (as eg. position, force) and must be of opposite polarity to the setpoint voltage. Potentiometer P7, is used to adjust the input sensitivity.

An additional input is available, wich can accomooate the higher voltages encountered when using a tachometer for measuring speed.

Adjust the loop gain using P2 to the largest possible value to achieve best system performance regarding accuracy and speed. Increasing loop gain is limited by the system becoming unstable.

If integral control action is required, close jumper « I » to activate the integrator and adjust gain using P5 on front panel. Integral control allows to eliminate static erros in control loops. To disable integral control, jumper points 8 and 10 (relay K1 can be used).

To achieve differential control, close jumper « D » and adjust gain using P6 and P8. Differential control improves system reponse and damping which has a stabilizing effect on the control element.

To activate the dither, close jumper « DITHER ». Use P4 to adjust dither amplitude and P3 for dither frequency.

To use voltage output close jumper « U ».
To use current output close jumper « I ».

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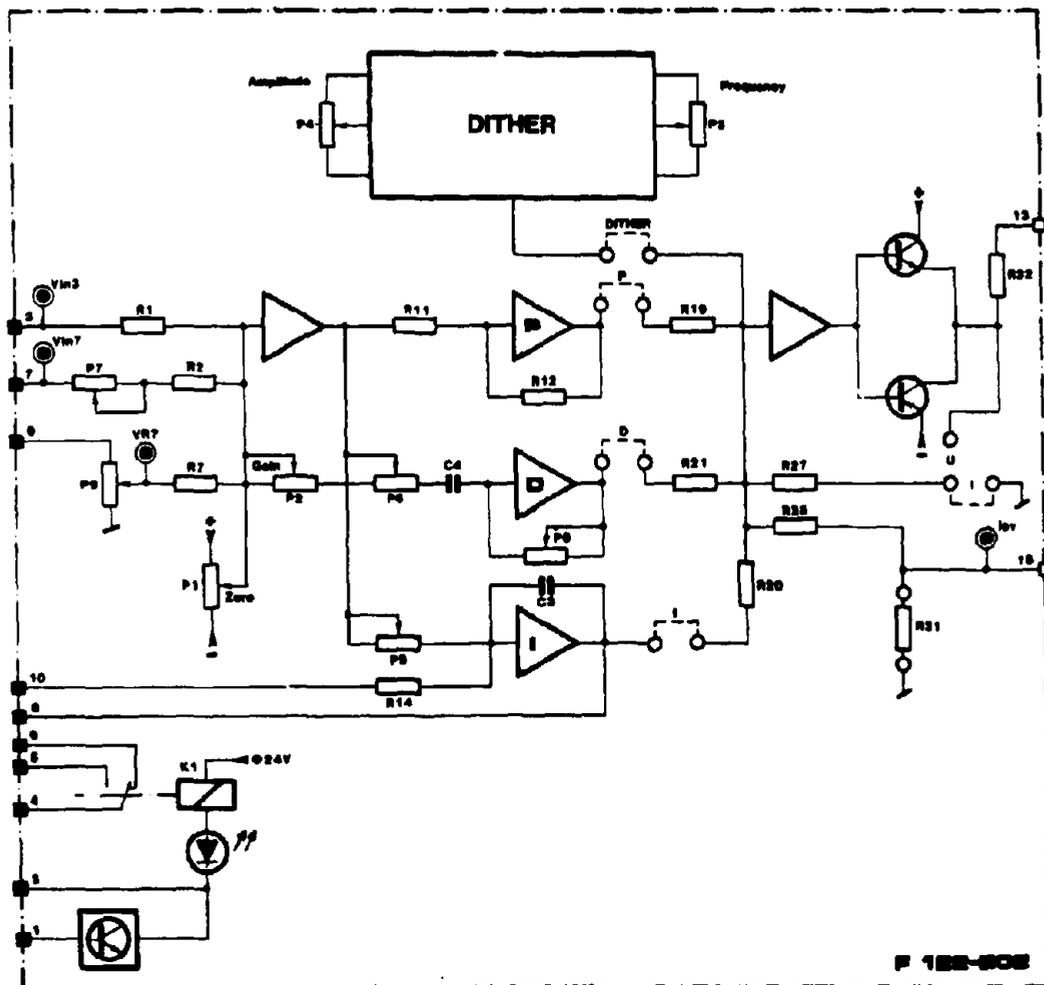
DESCRIPTION

The card F123-202 allows to check twelve different voltages prewired on the mother board terminals. This card facilitates system adjustments, modifications or maintenance.

SPECIFICATIONS

Power supply	: ±15 V stab. (100 mA each)
Input Voltage ranges	: 0 – 999 mV
	: 0 – 10 V
Resolution	: 1 mV
Input Selection	: switch on front panel
Range Selection	: automatic

USE

This card allows permanent checking or adjustment of up to twelve different system parameters requiring frequent calibration or verification.

BLOCK DIAGRAM

RANGE LOGIC UNIT

SW 1

F 123-202

DESCRIPTION

The card F123-203 generates a ramp of adjustable slope. The output voltage of the card is ramped from it's current value to the voltage applied to the card input. The slope can be either ascending or descending, depending on whether the input voltage is greater or less to the input voltage. When the ramp is completed, the output voltage is equal to the input voltage in amplitude and sign.

SPECIFICATIONS

Power supply   : ± 15 VDC stab
Input voltage   : ± 10 VDC max
Output voltage   : ± 10 VDC
                     Load impedance
Slope range   : 1,5 to 100 V/sec
                     this range may be modified as follow
                     - by increasing the value of R10 for
                     ascending slope and for R15 for de-
                     creasing slope
                     - by increasing the value of capacitor
                     C8

Adjustments :
Ascending slope   : P2 on front panel or ext. potentio-
                    meter
Descending slope : P3 on front panel or ext. potentio-
                    meter
Null shift   : P1 on front panel
Rounding at end
of slope   : P4 (by jumper)

Test points :
Input voltage   : jack Vin 3 on front panel
Output voltage   : jack Vout 9 on front panel
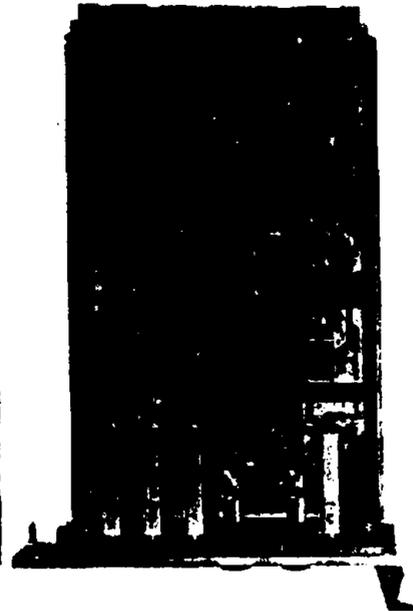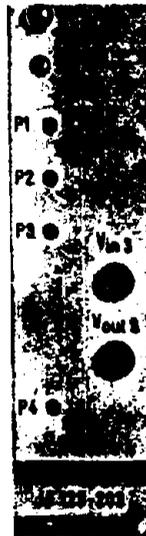
USE

The card F123-203 can be used in a speed control device to allow precise control of acceleration and deceleration. In a position controlling device it allows movement at a constant and adjustable speed.

BLOCK DIAGRAM
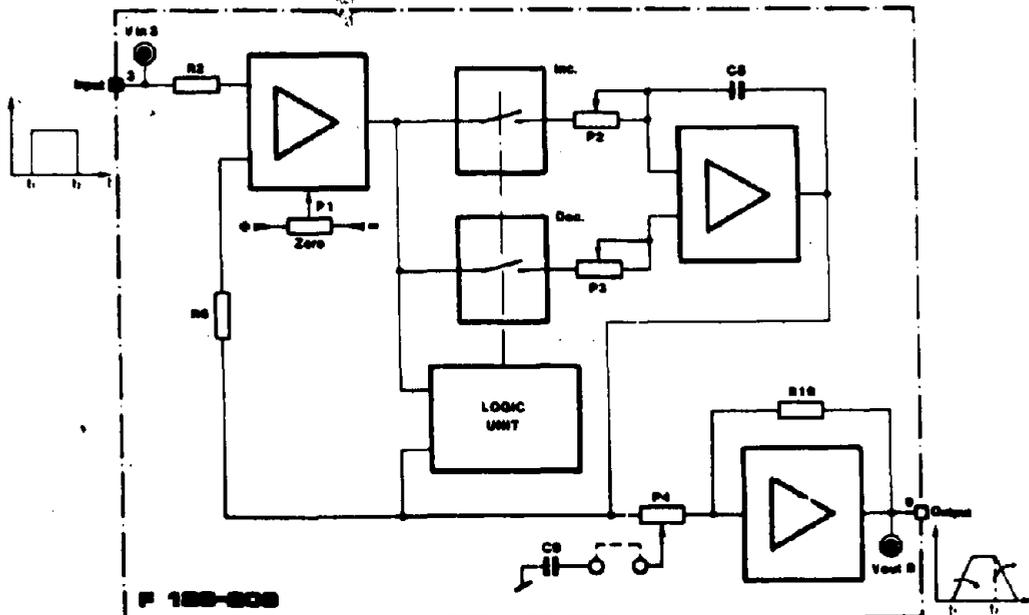
F 123-203

DESCRIPTION

The card F123-204 is a position demodulator intended for use with a LVDT position encoder. The oscillator circuit generates a sinusoidal waveform to drive the primary coils of the LVDT. The demodulator stage uses the signal returned by the secondary coils to generate a DC voltage directly proportional to the displacement of the core within the LVDT.

SPECIFICATIONS

Power supply : ± 15 VDC stab. (25 mA each)

Oscillator output :
- Voltage : 4,5.. - 11 Vpp (500 mA)
- Frequency : 0 to 2500 Hz

Demodulator output :
- Voltage : 0 ... ± 10 VDC adjustable
- Load impedance : 2 kOhm min.

Adjustments
Oscillator :
- Frequency adjustment : P1
- Output voltage : P2
Demodulator :
- Output voltage : P3
- Zero offset : P4

Test points
- Oscillator output voltage : jack Vout 13
- Demodulator input voltage : jack Vin 5
- Demodulator output voltage : jack Vout 11

USE

This card is used in electrohydraulic control loops using LVDT position encoders to supply a continuous voltage in direct proportion to sign and amplitude of the positioning element.

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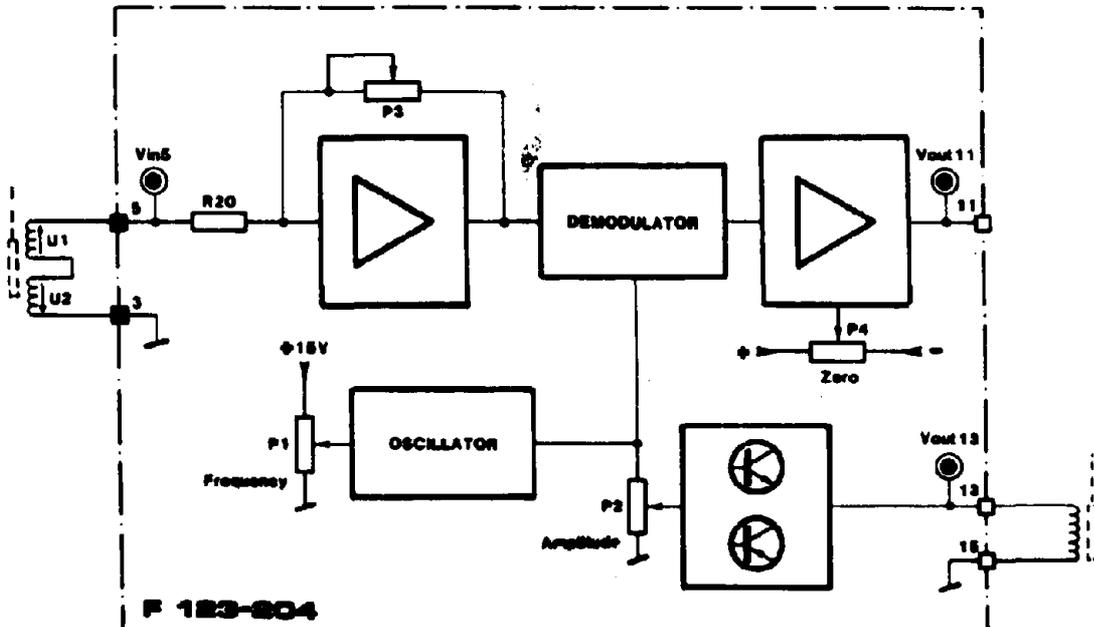
DESCRIPTION

The F123-205 card can compare two variable analog voltages or can be used as a threshold detector regarding a preset voltage.

The two relays K1 or K2 will be switched according to the result of the comparison.

K1 and K2 = OFF for $U_1 = U_2$
K1 = ON, K2 = OFF for $0 < (U_1 - U_2)$
K1 = OFF, K2 = ON for $(U_1 - U_2) < 0$

Two adjustable trigger threshold, adjustable hysteresis effect and adjustable delay time are employed to provide reliable switching characteristics.
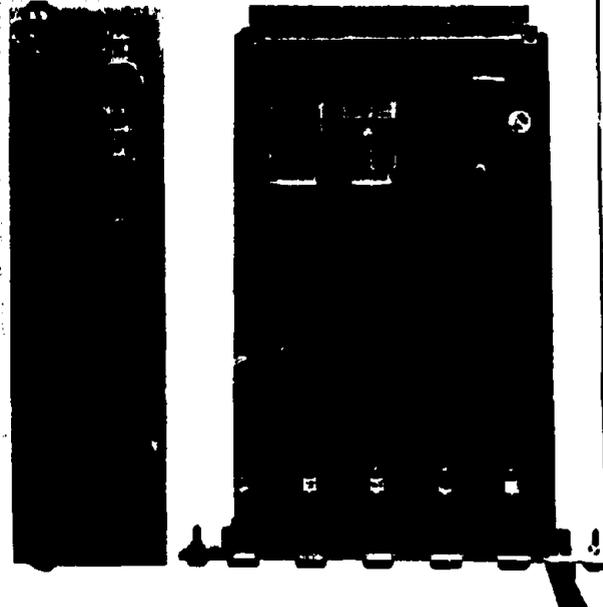
SPECIFICATIONS

Power supply : ± 15 V
Input voltage range : ± 10 V
Input resistance : ≥ 100 kΩ
Relay outputs : 2 SPDT contact
                contact rating 2 ?
Time constant : 0 to 10 seconds
Trigger threshold : 0 to ? V

Adjustments
 — Preset voltage : P1 (fine)
 — Hysteresis : R 25 (fine)
 — Threshold : P3 (for K1)
 — Delay time : P2 (fine) range
   constant

 — Delay time annulation : by ?

Test points
 — Preset voltage : Jack V?
 — Error : Jack TP 1
 — Error delayed : Jack TR 2
 — Threshold : Jack K1 : TP?
                K2 : TP?

Indicators
Two LED indicators on front panel show ON/OFF state of relays.

USE

The F123-205 card has been specially adapted to the use in analogic circuits where it is necessary to monitor the amplitude of a voltage that represents eg. position or speed.

The output of this card can then be used to give an alarm when a preset maximum value has been exceeded or to signal a sequence controller that the set point has been reached.

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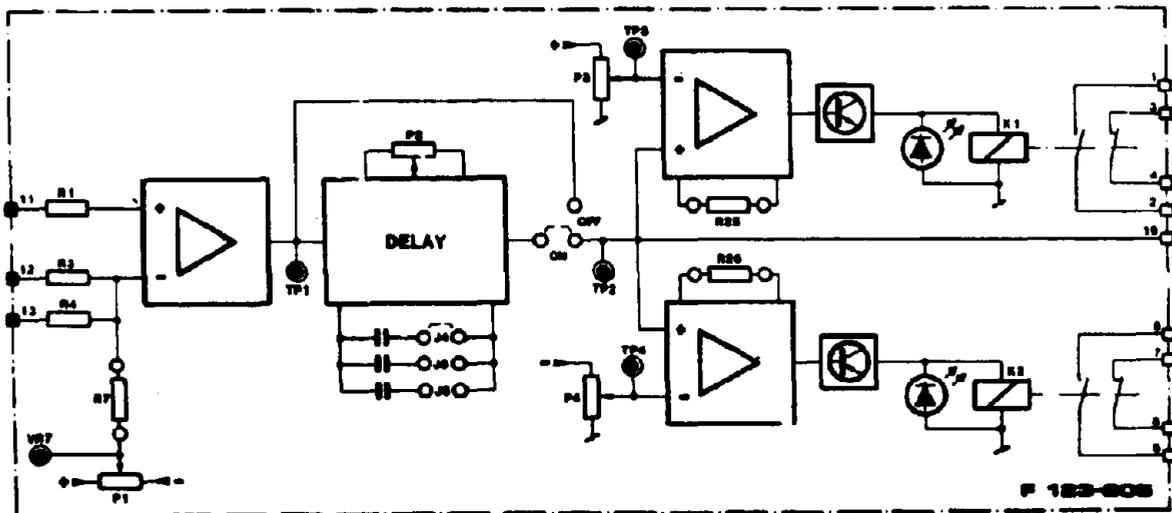
F 123-205

DESCRIPTION

The F123-206 card generates a sinusoidal, triangular or square wave voltage of adjustable amplitude and frequency.

This card can be used as a controller for position, speed, force, for example in automated test stands and fatigue test stands or wherever a set point voltage is required that varies according to sinusoidal, triangular or square wave functions.
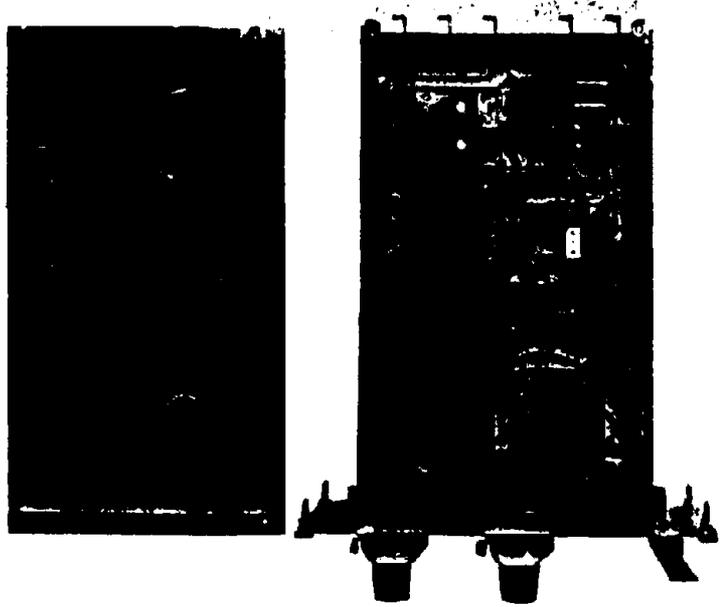
SPECIFICATIONS

Power supply	: ± 15 V stab. (30 mA each)	Test points	
Output signal	: sinus, triangular, square wave voltage	Output voltage	: jack Vout 4
		Adjustments	
Amplitude	: ± 10 V pp max.	Waveform selection	: switch
Load impedance	: > 2 kOhm	Frequency range	: switch
Frequency range	: range A : 0,1 to 1,5 Hz	Frequency adjustment	: 10 turns potentiometer with vernier scale
	range B : 1 to 15 Hz	Output amplitude	: 10 turns potentiometer with vernier scale
	range C : 10 to 150 Hz	Null shift adjust	: P1
	range D : 100 to 2000 Hz	Waveform symmetry	: P3

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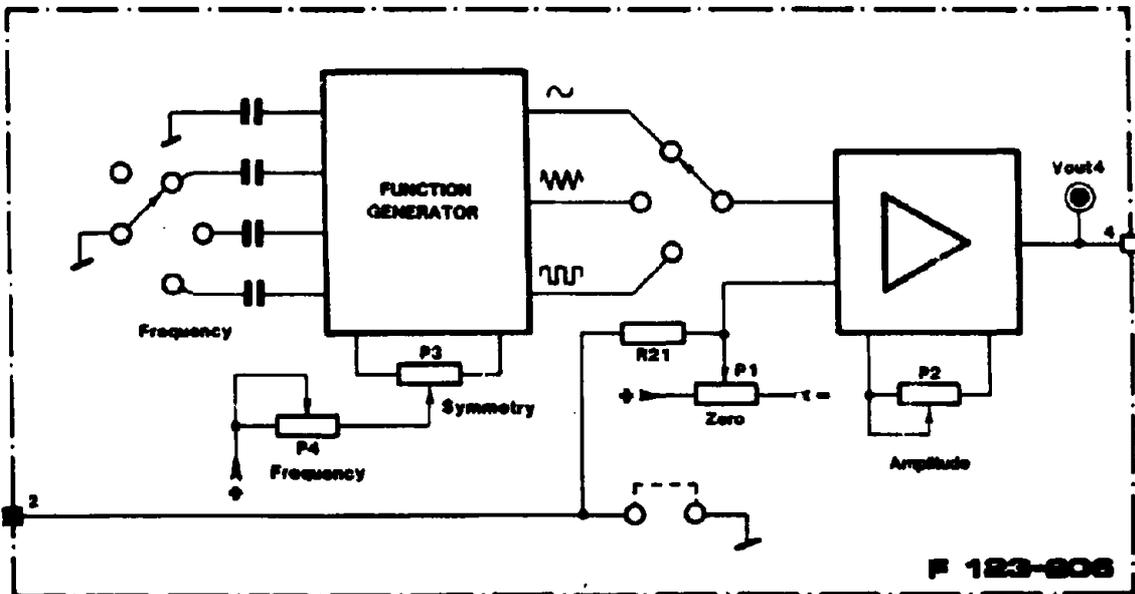
DESCRIPTION

The card F123-207 consists of :

- a potentiometer and resistor voltage divider which allows to adjust an output voltage between + 15 V and −15 V. This output voltage can also be switched by a relay externally controlled.

- A relay with a high contact rating to switch external devices.
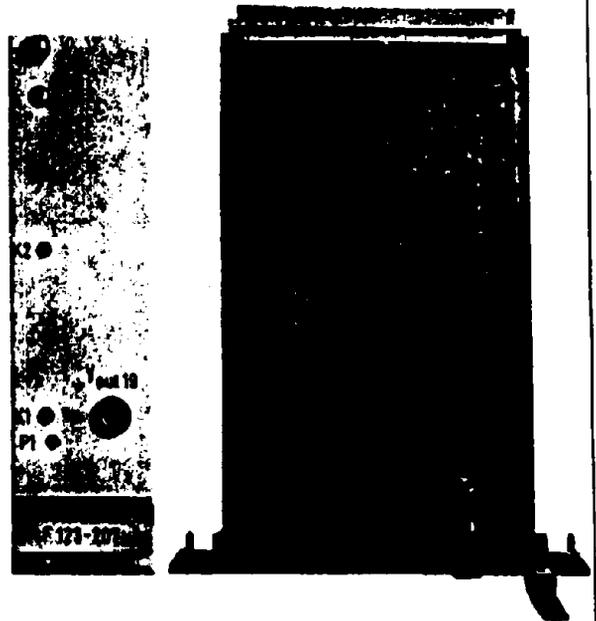
SPECIFICATIONS

- Power supply : ± 15 VDC stab.
  + 24 VDC
- Relays   K1 - consumption = 15 mA at 24 VDC
                    contact rating = 0.25A/24 VDC
                K2 - consumption = 25 mA at 24 V
                    contact rating = 2A/24 VDC
                    contacts = 4 x SPDT

Adjustments :
- Output voltages : P1 (R1, R2, R3)
- Function selection :
  . jumpers select input voltage for potentiometer
    + 15 V (J2), −15 V (J3) or ext. voltage (J1)
  . jumpers select output voltage :
    J4 — output is potentiometer output direct
    J5 — output is potentiometer switched by relay
    J6 — output is input voltage switched by relay

Test points :
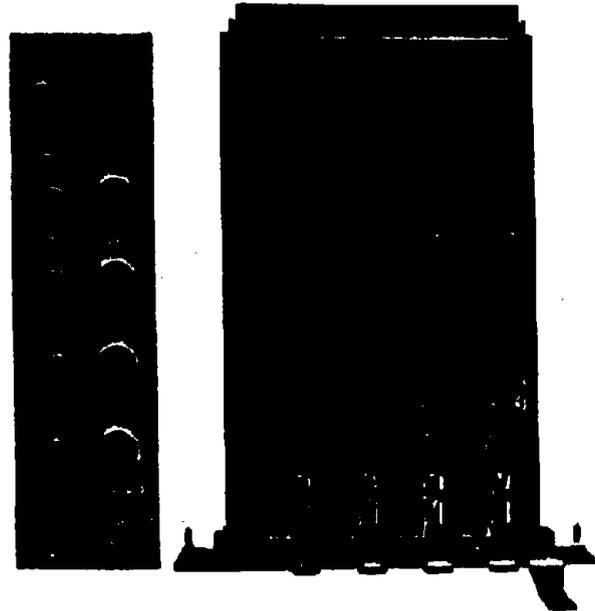Potentiometer output voltage : jack Vout 10

BLOCK DIAGRAM
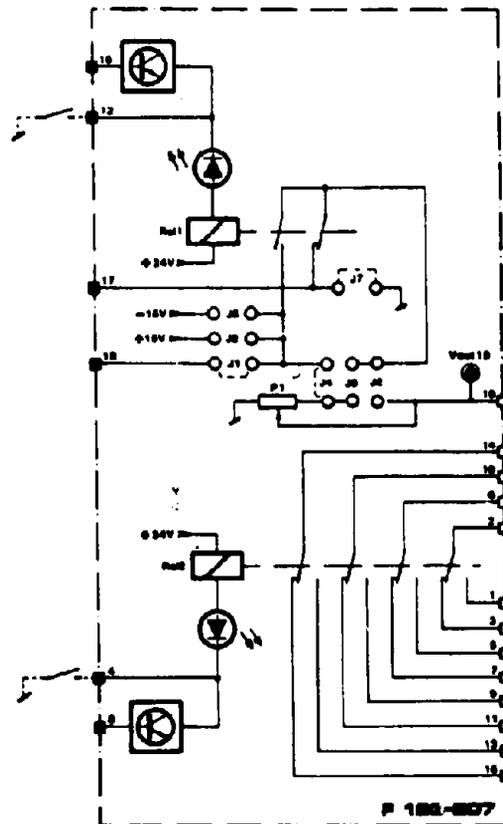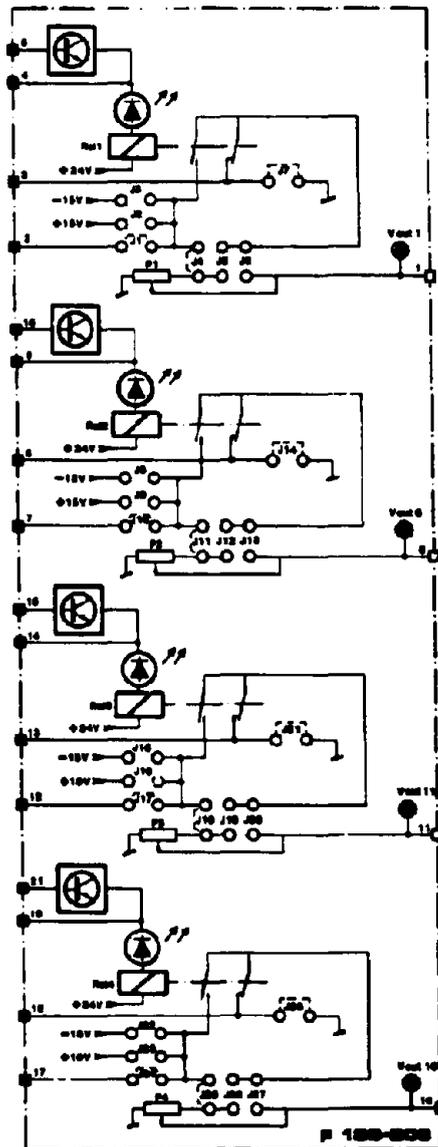
DESCRIPTION

The card F123-208 allows four set point voltages to be set between −15 VDC to + 15 VDC. These voltages can be used as set point values for a control device such as position or velocity controller. The set point voltages can also be switched by relays externally controlled.

BLOCK DIAGRAM

SPECIFICATIONS

Power supply : ± 15 VDC Stab
±24 VDC

Relays : consumption = 15 mA/24 VDC
contact rating = 2 A/24 VDC

Adjustments :
Output voltages :
potentiometer 1 - P1 (R1-R2-R3)
potentiometer 2 - P2 (R6-R7-R8)
potentiometer 3 - P3 (R11-R12-R13)
potentiometer 4 - P4 (R16-R17-R18)

Function selection :
jumpers select input voltage for each potentio-
meter + 15 V, −15 V or ext. voltage
jumpers select each output voltage :
J4, J11, J18, J25
output is potentiometer output direct
J5, J12, J19, J26
output is potentiometer switched by relay
J6, J13, J20, J27
output is input voltage switched by relay

Test points :
Output voltage :
potentiometer 1 : jack Vout 1
potentiometer 2 : jack Vout 6
potentiometer 3 : jack Vout 11
potentiometer 4 : jack Vout 16

DESCRIPTION :

The card F123-209 supply sensors like strain gages pressure transducers and load cells with necessary supply voltages and amplifies their output signal.

The supply voltage can be monitored via a remote sense line to provide temperature compensation and compensation of line voltage drop when using excessively long connections between sensor and conditioner card.

Three selectable amplifier ranges allow adaptation to a wide range of sensors.

The output from the sensor can be passed through a low pass filter to provide a smoother signal.
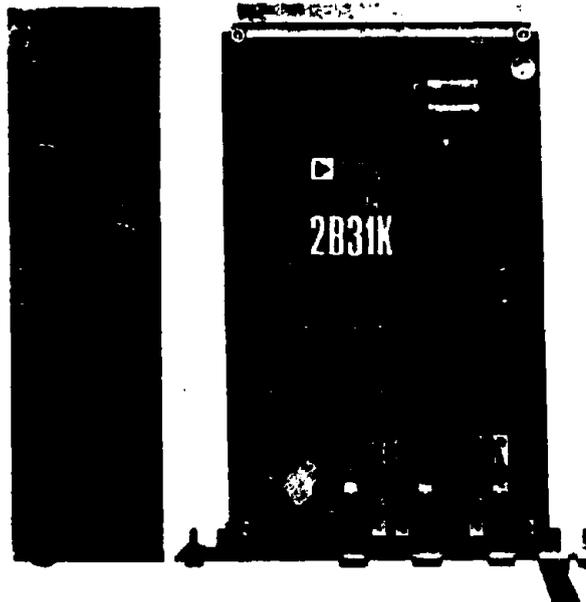
SPECIFICATIONS

Power Supply	± 15 VDC		P1 on front panel
Sensor Supply voltage	adj. 9 ... 11 VDC		4 - wire sensor - close jumpers J4, J5
Output Voltage	adj. 0 ... ± 10 VDC		6 - wire sensor - open jumpers J4, J5
Bridge Resistance	60 ... 1000 ohms		P2 on front panel
Sensor Ranges	a) 0,75 to 1,5 mV/V		close J1 to select 0,75 to 1,5 mV/V
	b) 1 to 2 mV/V		close J2 to select 1 to 2 mV/V
	c) 1,5 to 3 mV/V		close J3 to select 1,5 to 3 mV/V
Filter cut-off Frequency	2 to 5000 Hz		R7, R8, R9 (see table 1)

Test points :
Gage Zero Offset : Jack Vout
Unfiltered Output : Jack Vout 19
Filtered Output : Jack Vout 17

resistor R2
Select via jumper J6, J7
activate calibration via switch on
front panel (or ext. contact)

USE

The card F123-209 is used with pressure transducers or similar gages integrated into a control loop using a servo amplifier card like the F122-202.

INITIAL INSTALLATION

- Select the power supply mode of the sensor (4 wires or 6 wires for remote sensing) and adjust the supply voltage to 10 VDC.
- Adjust the sensor zero offset at null pressure.
- Adjust the zero offset of the output stage to read 0 VDC.
- Adjust the output level of the amplifier at max. pressure, or use the calibration resistor.
- Eventually readjust the zero setting of the output stage.

TABLE 1
Filter cut off Frequency

| F (HZ)	R9 (KΩ)	R8 (KΩ)	R7 (KΩ)
2	∞	∞	∞
5	1270	2050	383
10	523	806	154
50	40	137	26,7
100	44,2	68,1	13,3
500	8,66	13,3	2,61
1000	4,32	6,65	1,3
5000	0,866	1,33	0,261

BLOCK DIAGRAM

F 123-209
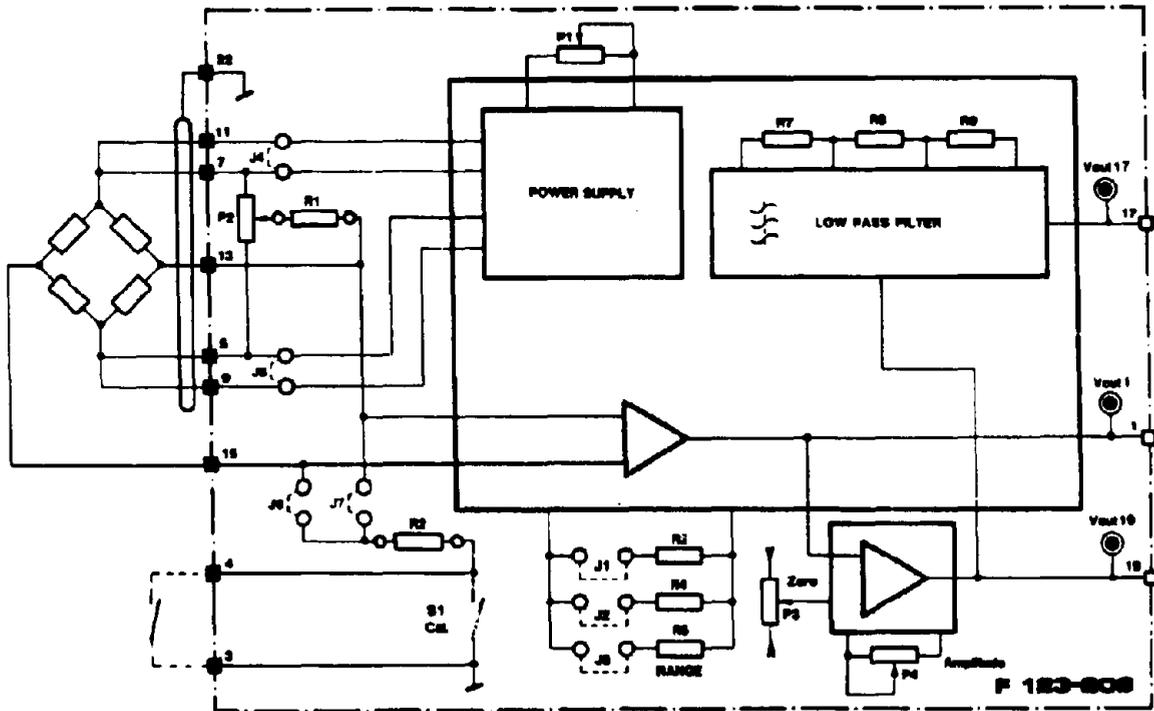
DESCRIPTION

The F123-210 card contains a number of auxiliary analog functions often needed to implement an analog control scheme. This card makes available two amplifiers with 4 inputs each and a precision rectifier.

BLOCK DIAGRAM

F 123-210
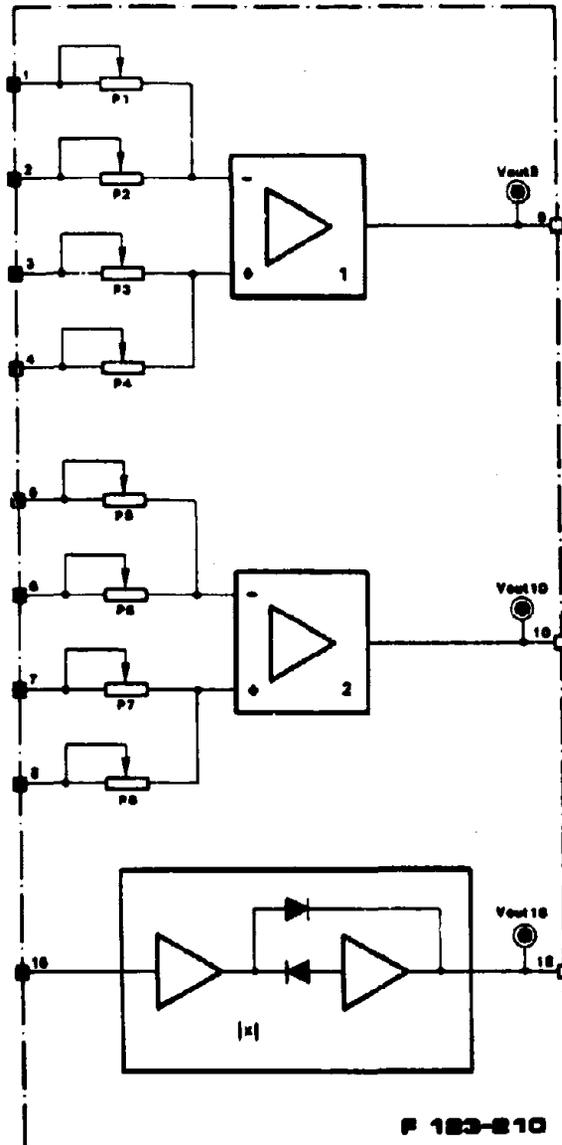
SPECIFICATIONS

Power Supply : ± 15 VDC stab (15 mA each)
Output Voltages : ± 10 V max
Input Impedance : amp 1 = 100 kohm
: amp 2 = 100 kohm
: rectifier = 20 kohm
Load Impedance : 2 kohm min
Gains : amp 1 and amp 2 = ± 1
: rectifier = +1

Adjustments
- amp 1 gain : P1, P2, P3, P4
- amp 2 gain : P5, P6, P7, P8

Test Points
- ampli 1 output : jack Vout 9
- ampli 2 output : jack Vout 10
- recitifier output : jack Vout 18

USE

The amplifiers on this card can be used as either adders, subtractors or inverters needed to complete a control scheme beeing realized with MOOG standard cards of the series F 120.

The precision rectifier can be used to obtain the magnitude (absolute value) of a voltage, which can then be compared to a fixed voltage for threshold detection using MOOG card F123-205.
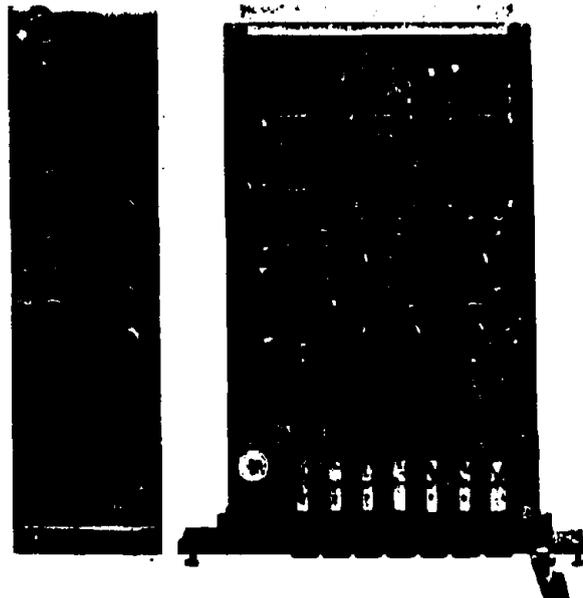
DESCRIPTION

The card F123-211 supplies two sensors like strain gages, pressure transducers and load cells with the necessary supply voltages, amplifies their output signal and delivers an electrical signal proportional to their sum or their difference such as $\Delta = \pm \alpha P1 \pm \beta P2$ α and β being comprised within zero and one.

Three selectable amplifier ranges allow adaptation to a wide range of transducers.

Trimming potentiometers and test points are located on the front panel.

SPECIFICATIONS

Power Supply	: +/- 15 VDC
Sensor Supply Voltage	: adj. 0 to 10 Vdc
Output Voltage	: adj. 0 to +/- 10
Bridge Impedance	: 350 ohms
Sensor Ranges	: 0.75 to 1.5 mV/V
	: 1 to 2 mV/V
	: 1,5 to 3 mV/V

Test Points

Output transducer 1	: P1 and − P1
Output transducer 2	: P2 and − P2
Output divided	: α and β
Output summation	: Δ

Adjustments

Sensor Supply Voltage	: P1
Sensor Zero Offset	: P6 and P7
Sensor Output Voltage	: P4 and P5
Sensor Gain	: close J1-J4 to select 0,75 to 1,5 mV/V
	: close J2-J5 to select 1 to 2 mV/V
	: close J3-J6 to select 1,5 to 3 mV/V

USE

The card F123-211 is suitable to be used in a control loop using servo-amplifier card like F122-202 in conjunction with two pressure transducers, each output being used differentially or independently.
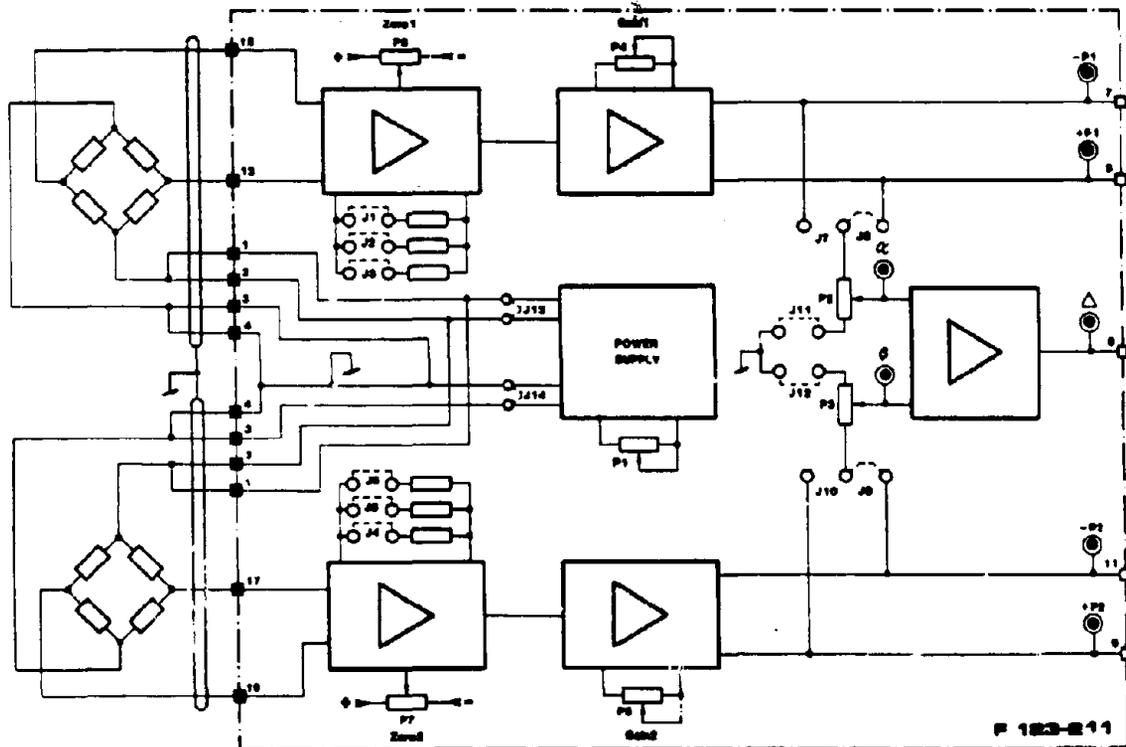
INSTALLATION

— Connect the sensors (+ excitation : 1-2, − excitation 3-2)
— Select the proper sensor gain
— Adjust the sensor supply voltage by P1
— Adjust each zero offset with a null pressure by P6 and P7
— Adjust each output level by P4 and P5
— If differential output is used, select the mode
  (close jumper J11 and J12)
  and adjust the dividers α and β.

BLOCK DIAGRAM

F 123-211

DESCRIPTION

The card F124-201 is a function generator which converts a function stored in PROM memory in digital form to an analog voltage. This allows any type of function once defined and stored into PROM to be used in a control system.

The stored function can be regenerated as either a continuous voltage-time waveform (X-T mode) using a clock frequency or as a voltage-voltage converter (X-Y mode) using an external input voltage.

Several cards F124-201 can be coupled to generate more complexe syncronized waveforms.
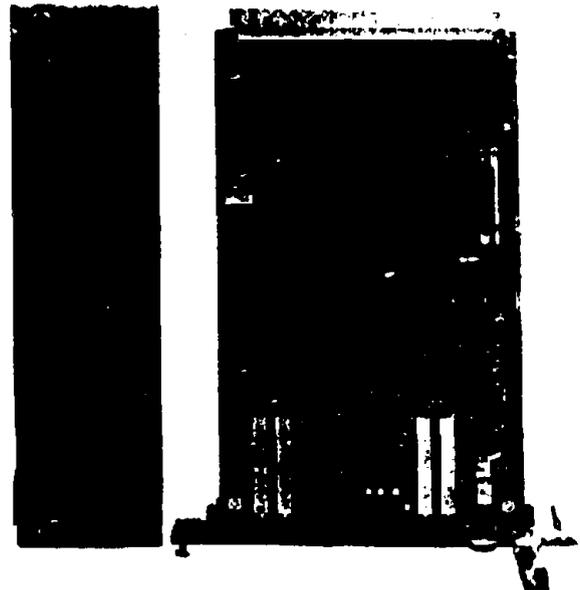
SPECIFICATIONS

Power supply : ± 15 VDC stab (60 mA each)
    + 5 VDC stab 15 mA
    (internal J 19 external J 20)

Prom :
• type 2704 close J12-J13-J16-J18 (512 samples)
• type 2708 close J11-J13-J16-J17 (1024 samples)
• type 2716 close J11-J14-J15-J17 (1024 samples)

Function resolution :
• 8 bit resolution (0-255)

X-T mode :
• function in PROM regenerated using
   . range 1 : 0,2 to 2 seconds
   . range 2 : 2 to 20 seconds
   . range 3 : 20 to 200 seconds
   . range 4 : external clock frequency (5 to 10 KHz)
     (Vpp = 5 V)

X-Y mode :
• function in PROM used as voltage-voltage converter (input voltage range : 0 to + 10 VDC or -10 to + 10 VDC)

Output voltage :
0 to + 10 VDC or -10 to + 10 VDC

Single/Continuous :
in X-T mode, stop after one pass thru PROM or generate continuous waveform

Stop :
in X-T mode, stops clock generator (waveform)

End of cycle :
in X-T mode a + 5 V pulse of 5 ms duration signals end of pass thru PROM

Test points :
output full voltage : jack Vout 8
output adjustable voltage : jack Vout 10

Adjustments :

Frequency :
P1 on front panel
(range by ext. switch or jumpers on circuit board)
Amplitude :
P5 on front panel
Mode selection :
   output voltage   : see table 1
   input voltage   : see table 1
   X-T, X-Y   : see table 1
   single/continuous : switch or ext. signal
   stop X-T   : switch or ext. signal

USE

Especially designed to be used in automatic control of differents types of hydraulic devices, the card F124-201 is used as a programmable function generator between classic designs (sine, triangle and squarewave generators) and more evolved systems using microprocessors.

TABLE 1

	S1	S2	S3	S4	S5	S6	S7	S8
Input 0/+10V	–	–	–	–	–	–	0	–
Input –10V/+10V	–	–	–	–	–	–	1	–
Output 0/+10V	–	–	–	–	–	–	–	0
Output -10V/+10V	–	–	–	–	–	–	–	1
X - T Mode	0	0	0	1	1	–	–	–
X - Y Mode	1	1	1	0	0	1	–	–
Internal clock	–	–	–	–	–	1	–	–
External clock	0	0	0	1	1	0	–	–

NOTE :

– In X-T mode always apply + 15 VDC to pin 6.

– If an external switch to select frequency range is not used, jumper pin 1 to pin 3,5 or 7.

BLOCK DIAGRAM

F 124-201

DESCRIPTION

The card F124-202 is a ramp generator with output amplitude and slope being adjustable. The card can produce either symetrical triangle or sawtooth waveforms. Pressing START does one cycle ; ie. ramps the output voltage to the set point value and then back to the initial value. The ramp can be stopped at any time by pushing the « STOP » button and then resumed by pressing START again.

To stop the output voltage at the set point value close jumper J3. The descending ramp will start by an external signal (pin 15) or by pushing the START button.

SPECIFICATIONS

Power supply :
± 15 VDC stab (40 mA)
+ 5 V stab — 10 mA (internal)
Output voltage : 0 ... 10 V
Load impedance : 2 kOhm min.
Frequency range : 1 ... 80 ...

Test points :
Ramp output voltage

Adjustments :
Ramp peak voltage :
10-turn potentiometer
Slope :
10-turn potentiometer
selected by toggle switch

Output polarity :
toggle switch
Slope sign :
toggle switch
Waveform :
toggle switch selects triangle/sawtooth
Reset cycle :
push button or ext. contact resets ramp gene-
rator

push button or ext. contact freezes ramp

push button or ext. contact starts ramp
... ramp after STOP

USE

The card F124-202 is an automatic voltage generator for control of speed, position, pressure or force where very slow set point voltage ramps are required.

BLOCK DIAGRAM
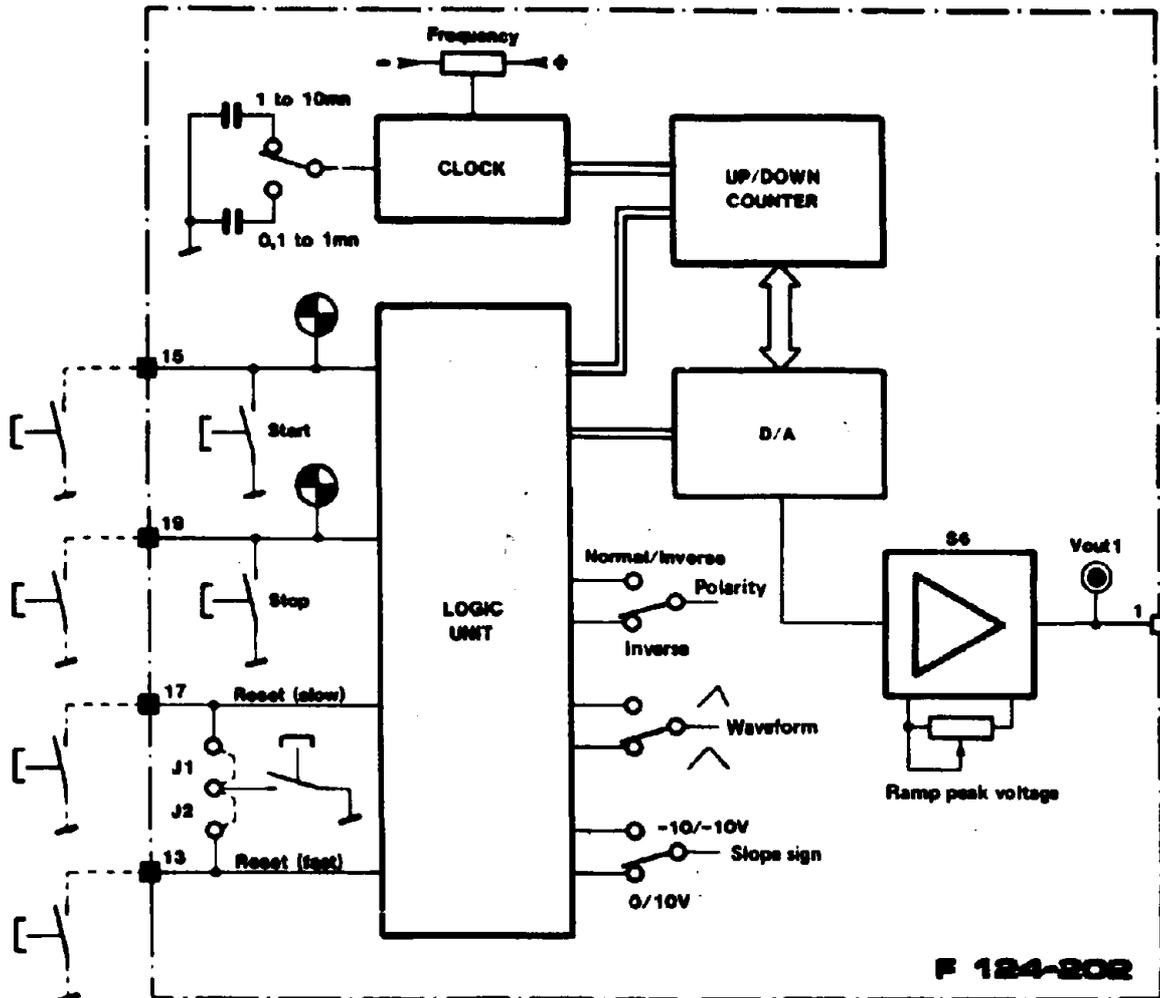
F 124-202

DESCRIPTION

The card F124-203 consists of an up/down counter which can be preset via front panel BCD switches.

This card then counts the number of preset pulses, either digital (TTL or CMOS compatible) or analog signals via a schmitt-trigger input.

When the final count is reached (0 when counting down from preset value, or preset value when counting up from 0) a relay and an analog switch are toggled.

The up/down counter is started, stopped and reset either using the front panel push buttons or via external signals.

SPECIFICATIONS

|	
Power Supply	: ± 15 V stab (110 mA exc.)
Input sensitivity	: pulse (pin 13) 1,5 V/ap ... ... wave/form signal (pin 15) ... ... to 10 Vpp
Frequency	: 5 Khz max.
Counter capability	: 0 to 999999
Contact rating	: relay 250 mA/24 ... ... : Résistance 800 ohm ... ... 0 ... 10 V (only pos...)
Adjustments :	
Trigger levels	: P1 and P2
Start	: push button in front panel ... ... ternal contact
Stop	: push button in front panel ... ... ternal contact
Reset	: push button in front panel ... ... ternal contact
Up/Down Selection:	switch in front panel
Present Value	: BCD. coded switches on front panel (6. decimals)
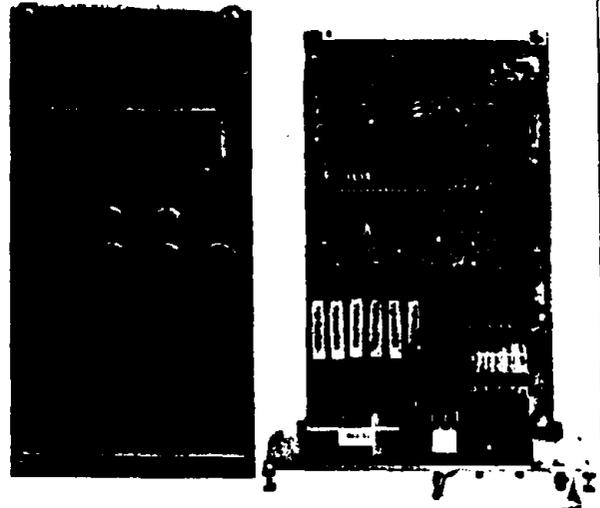
BLOCK DIAGRAM
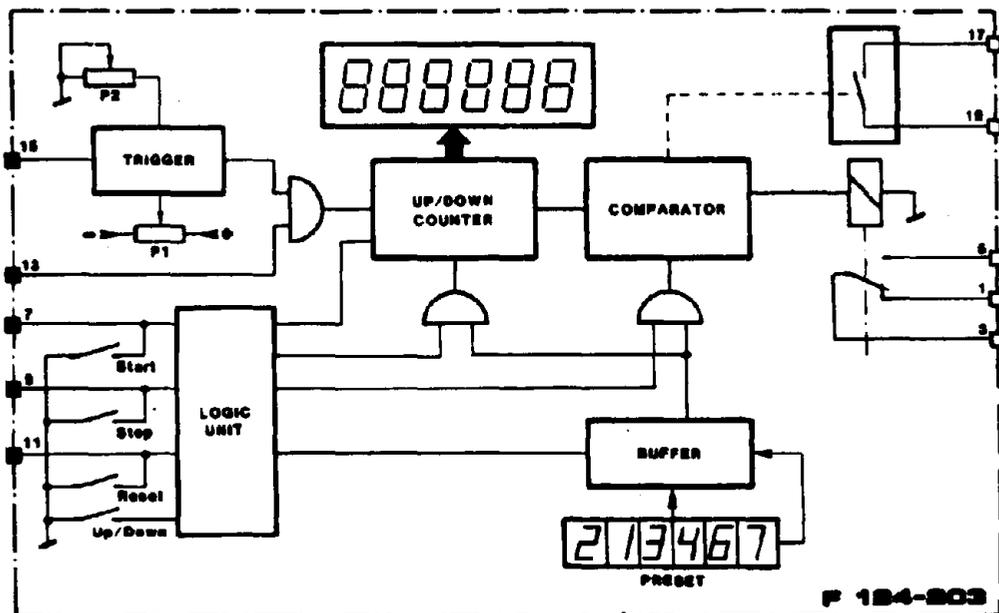
USE

The card F124-203 is especially designed to be used in automatic sequence control, wherever the number of cycles to do are predetermined or the number of cycles executed are to be counted, as for example in fatigue test stands.
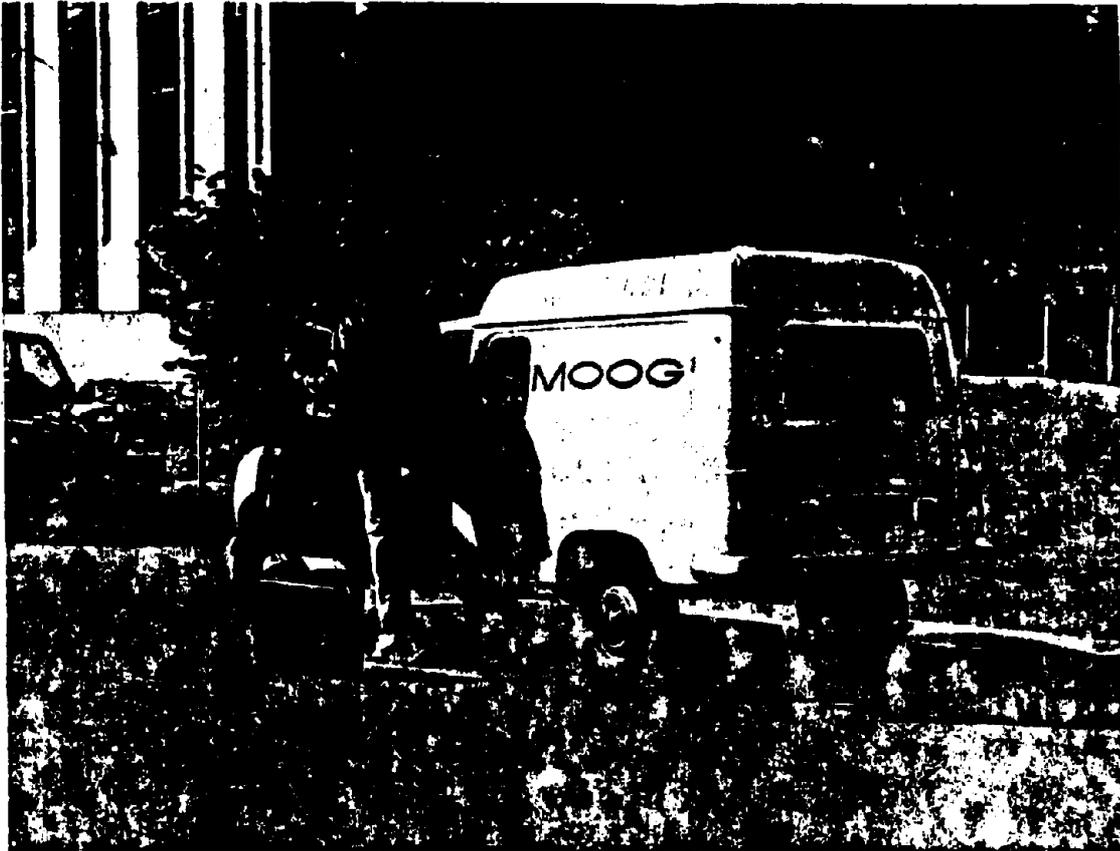
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Moog
Robot Wrist
An Electrohydraulic Actuator

MOOG
PLANTS AND
OFFICES
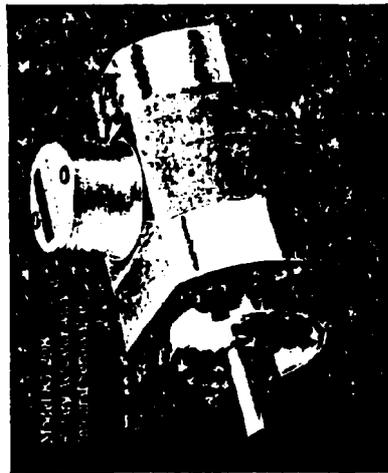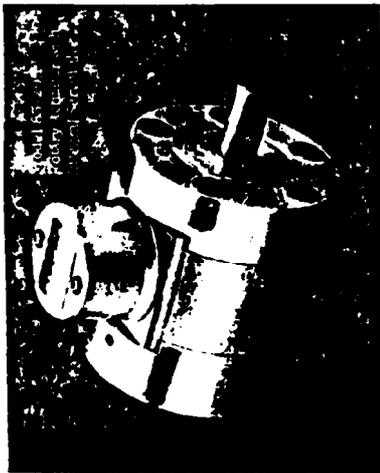
MOOG

Rotary Actuator Specifications	Moog Model 85-285	Moog Model 85-286
Displacement	0.18 IN³/RAD	0.35 IN³/RAD
Rotation	270 Degrees	270 Degrees
Supply Pressure	2500 PSI Max	2500 PSI Max
Stall Torque at 1000 PSI	100 IN-LB	200 IN-LB
Operating Torque at 1000 PSI	50 IN-LB	100 IN-LB
No Load Velocity at 1000 PSI	1000 DEG/SEC	800 DEG/SEC
Servovalve Parameters		
• Flow at 1000 PSI	1.0 GPM	1.5 GPM
• Coil Resistance	100 OHMS	100 OHMS
• Rated Current	20 MA	20 MA
Actuator Weight	2.6 LBS	4.5 LBS

Moog®
Robot Wrist

FEATURES:

- High Speed. High Performance
- Compact, Sturdy, Light Weight
- Long Life, No Leakage
- No External Plumbing or Cables
- Intrinsically Safe Design
- Easy Installation and Maintenance

The Moog 3-Axis Electrohydraulic Actuator offers a compact, totally integrated, long life wrist assembly for the Robot Industry. Completely enclosed hydraulic and electrical connections eliminate external hydraulic hoses and electrical cables. Life is extended beyond the limitation imposed by dynamic seals by eliminating the high pressure dynamic seals.

The unique electrohydraulic servo package design is sturdy and light weight. Rotary motion for each axis is provided by a single vane laminar actuator* with an integral servovalve. The structural brackets, designed for strength and light weight, house the rotary position transducers (servopot, resolver or encoder) for closed loop control of each axis.

Maintenance is simplified by the modular nature of the wrist assembly which allows replacement of an actuator unit without disturbing the other two axes.

The assembly is installed by manifold mounting the base of the assembly. Both electrical and hydraulic interfaces are provided at the base manifold with a single electrical connector and o-ring sealed hydraulic supply and return ports.

Internal hydraulic porting is the key to the clean and compact design. Each rotary actuator housing in the Moog wrist has two o-ring sealed manifold ports for hydraulic supply and return connections. These ports supply fluid to the integral servovalve and through the actuator shaft and mounting bracket to the actuator of the next outboard axis.*

The intrinsically safe servovalve coils and rotary transducers used in the design make the wrist safe for operation in hazardous environments of spray painting and adhesive dispensing applications.

Moog 3-Axis Robot Wrists permit operating speeds in excess of 1000 degrees per second. Two types of wrist motion are available. In Type "A" the yaw actuator provides bend motion about an axis parallel to the base. In Type "B" the yaw actuator provides a rotate motion about an axis perpendicular to the base. A bend motion about the pitch axis and a rotate motion about the roll axis are common to both wrist types. The yaw and pitch axes rotate 220 degrees and the roll axis rotates 270 degrees. Moog wrist assemblies with corresponding payloads, weight and size are tabulated below.

*Patent applied for

3-AXIS ELECTROHYDRAULIC ACTUATION

Type "A"

Type "B"
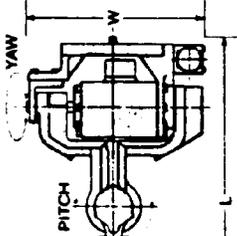
Moog Wrist	Wrist Type	Payload @ 1000 PSI	Payload @ 2000 PSI	Wrist Weight	Wrist Size L,W (inches)
1	A	4 LB	7 LB	17 LB	17.9
2	A	9 LB	18 LB	20 LB	17.9
3	B	9 LB	18 LB	20 LB	21.8
4	A	14 LB	28 LB	36 LB	20.12
5	B	14 LB	28 LB	36 LB	24.12

Wrists 2 and 4 assume yaw motion in the horizontal plane (as depicted).

Pages C112-C116 have been removed.

Due to copyright restrictions, pages C112-C116 have been
omitted

Pages C117-C121 have been removed.

Due to copyright restrictions, pages C117-C121 have been omitted.

REVIEW OF DOWNHOLE GEOTECHNICAL TOOL

prepared for

SWEET AND AIKEN, INC.
13810 Champion Forest Drive
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by

EQUIPMENT DEVELOPMENTS CO.
8911 Rocky Lane
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713 937 6873
July 12, 1985

REVIEW OF DOWNHOLE GEOTECHNICAL TOOL

GENERAL

This report is a review of the electronic portion of a downhole geotechnical tool intended for use in ma ing soil measurements during offshore geotechnical drilling. It consists of a mechanical section designed to be run inside the drill string, land and lock in the drill string and extend a tubular member into the soil at the bottom of the borehole. The tubular member is then moved or oscillated in a rotary manner by a hydraulic powered servo mechanism. The response of the tube is then measured electronically by transducers. These electrical signals along with signals from other transducers are transmitted to the surface via electrical conductors.

OPERATION

The use of the tool is envisioned to be operated from a normal geotechnical drilling vessel and is to be wireline operated. Since the tool is hydraulically powered, it will be connected to a surface control console by an umbilical containing at least a single hydraulic conductor along with the necessary electrical conductors for power, control, and data. During operation, the umbilical should be stored on a powered winch and payed out over an appropriately sized sheave to minimize chances for damage to the umbilical. The winch can be air or electrically powered. It should contain a live hydraulic swivel and an electrical slip ring assembly to allow operation of the tool without having to disconnect the umbilical.

Because of the hostile offshore environment, the operation of the overall system should be kept as simple as possible simplicity should be considered through out the design process. As shown in sketch 1, the successful operation of the tool will necessitate the following major components or subsystems:

> Surface hydraulic control console.
> Surface electronics/data acquisition/data recorder console.
> Powered winch and sheave system.
> Umbilical.
> Downhole tool including sensors and downhole electronics package.

ACTUATOR

The actuator is defined as a hydraulically operated rotary motor coupled to a servo valve using either torque or angular displacement feedback. This system is technically feasible, but will require great care in the design, and ultimately during operation, to insure cleanliness of the hydraulic fluid. Adequate filtration should be designed into the system. There are two ways to handle the hydraulic system: either a return to surface

1

system using a pressure and a return line in the umbilical, or a waste to sea system that requires only a single pressure line in the umbilical for the actuator. For this tool a waste to sea system is the recommended approach but care must be taken in the design of the vent components to insure that sea water and contaminants do not intrude into the hydraulic fluid. In a waste to sea system, the issue of environmental contamination must be considered. Other considerations for the selection of the hydraulic fluid include specific gravity relative to sea water, viscosity, compatibility with components, and corrosion potential on hydraulic components. One of the water based fluids should be considered.

As an alternative to the hydraulic actuator, an electrical torque motor should be considered for the tool provided that a small size adequately powered motor can be located. A manufacturer such as Globe is a potential supplier for the motor. Although more electrical conductors would be required in the umbilical, the problems associated with maintaining cleanliness of the hydraulic fluid would be eliminated along with the power hydraulic fluid conductor in the umbilical.

Command inputs to the servo motor will be sine wave motion and step inputs. Frequency will be 0.1 sec/cy to 50 sec/cy at amplitudes ranging from approximately plus and minus 0.01 degrees to plus and minus 15 degrees.

SENSOR ENVIRONMENT

The sensors or transducers defined for use in the geotechnical tool are to be located inside the tool body and operated in a dry nitrogen environment at just above ambient sea pressure. Since the tool will be operated at water depths up to 1500 feet, the internal pressure will reach approximately 700 psi. This is a technically feasible approach and presents several advantages as far as the transducers are concerned. Intrusion of seawater and contaminants into the tool body will be minimized. The main seal which separates the seawater from the tool body cavity will be pressure balanced thereby minimizing frictional drag that will be seen as an error in some of the measurements. Although seal friction drag can probably be cancelled during calibration, variations in the quantity during operation cannot be accounted for in the calibration.

The approach of using a one atmosphere chamber for the sensors should not be considered; but, using a pressure balanced chamber filled with a dielectric fluid such as Dow Corning fluid number 200 is a good approach and should be considered for this tool. The attached sketch 2 illustrates this concept.

In either case, the selected sensors and transducers must be carefully reviewed with the vendors to insure that they can operate in the elevated ambient pressure environment and that they will maintain their performance characteristics over the

2

defined pressure and temperature ranges.

SENSORS

The following is a list of the sensors defined for use with the geotechnical tool.

A single torque sensor will be required. The Lebow strain gage unit which has been selected is an excellent choice provided that it can perform at the elevated pressure environment and that it will fit into the allocated space envelope.

A single angular displacement transducer will be required. The Trans-Tek unit which has been selected should prove to be an excellent choice if, like the torque sensor, it can operate in the enviro^ment and it will fit into the tool.

An angular accelerometer has been defined and will be made by mounting a pair of linear PCB quartz accelerometers on equal length arms 180 degrees apart. This approach should prove satisfactory if calculations show that the accelerometers have the proper range and sensitivity to operate over the frequency and amplitude ranges. Of course operation at the elevated pressure must be checked with PCB engineering.

Three pressure sensors have been defined, two of which will be differential types. The PCB pressure sensors selected are not satisfactory for this application because the PCB quartz sensors are not capable of measuring static pressures. Therefore another selection must be made such as those manufactured by Entran.

DOWNHOLE ELECTRONICS PACKAGE

The successful operation of this tool will require the use of a downhole electronics package for transducer signal conditioning, switching and control, amplification, distribution of command signals to the servo drive motor, and multiplexing of data signals to minimize number of conductors in the umbilical. It is a critical component and will be a custom design. There are no known off-the-shelf units to meet the needs of this tool. Unlike the cavity where the transducers and sensors are located, the electronics package should be of the one atmosphere design. Many electronic components will not operate satisfactorily at the elevated pressure of 700 psi. To do so would require special components to be manufactured and qualified over the pressure range. This would be more expensive and time consuming than designing and building the one atmosphere chamber. The design must consider the external pressure and must be carefully sealed to prevent intrusion of either seawater or nitrogen. If possible the seal should be a sealweld and tested for leakage prior to use downhole.

3

C12

DATA TRANSMISSION

Based on the known operational requirements and environment for
this tool, it is recommended that the data be transmitted from
the downhole electronics package via the umbilical to the surface
electronics console in analog form. This approach will minimize
the electronics component count in the downhole electronics pack-
age, thereby improving reliability. It will also give the equip-
ment operator and field engineer maximum flexibility in the use
and interpretation of the data. Digitization, other processing
and recording can then easily be done in the surface package
where space is not at such a premium.

An analysis of the sensors and required data shows that the
following number of electrical conductors will be required.

> 1. Electrical power--probably 12 or 24 volts DC.
>
> 2. Power common, signal ground, and shield.
>
> 3. Command signal--includes servo command encoded with
>   calibration ON/OFF signal.
>
> 4. Angle.
>
> 5. Torque.
>
> 6. Angular acceleration.
>
> 7. Pressure signals, 3 each--multiplexed together using a
>   frequency modulation technique. These are nearly sta-
>   tic signals and do not require wide bandwidth. Center
>   frequencies of 100 Hz., 500 Hz., and 2500 Hz. should be
>   satisfactory.
>
> 8. Spare.

Data transmission reliability will be improved if very low imped-
ence amplifiers having sufficient current capacity are used to
drive the umbilical even in the presence of minor salt water
shorts.

UMBILICAL

The electro-hydraulic umbilical will be a critical component and
must perform properly to insure long-term reliable operation in
the field. Initially, during field testing of the prototype
tool, the umbilical can be made from a multi-conductor hose
bundle strapped or taped to a well logging cable. The well log-
ging cable will also have a good tension strength member included
in the cable. Ultimately the umbilical should have the hoses and
electrical conductors all inside an overall jacket and strength
member. During operation, it will be necessary to store the
umbilical on a powered winch to insure that it does not become

entangled and kinked.

Do not have any connections or splices in the umbilical. Have a
single connection between the tool and the umbilical so that it
can be disconnected for repair or storage.   As much as possible
keep the entire system connected together to minimize the chances
for trash and contaminants to enter the hydraulic system.   Also
it is important to keep the hydraulic umbilical free of entrapped
air.

OTHER DESIGN CONSIDERATIONS

The following are a few tips to consider in the design that will
impact the reliability of the data obtained and the successful
operation of the tool in the hostile offshore environment.

Minimize the use of dissimilar metals which can cause electrical
potentials to be set up in the presence of the salt water elec-
trolyte.   A small sacraficial annode might have to be incorpor-
ated onto the tool.

Provide for easy repair and disassembly of the tool to the extent
possible.   Routing of electrical wires inside the tool body must
be done in a careful manner.

Conduct extensive calibration and testing prior to going to the
field. Try to work out as many bugs as possible first.

Keep in mind that the various stainless steel alloys that may be
used in the tool body are subject to galling, particularly in the
threaded areas.   Proper tolerencing and lubrication are impor-
tant.

Hydraulic control console

Electronics package

winch

umbilical

Connector

Electronic package

Double Tool
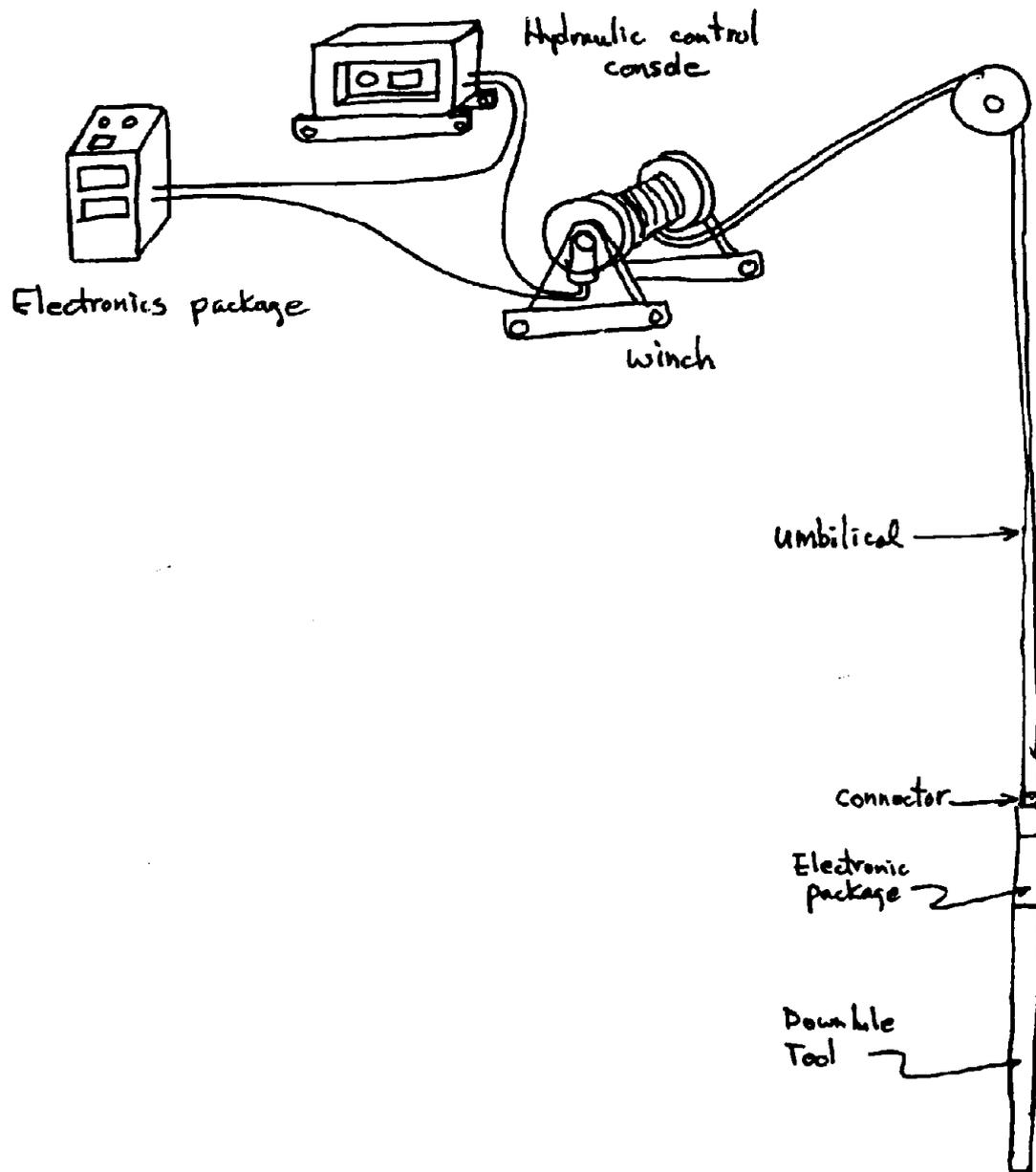
SKETCH 1: OVERALL SYSTEM SHOWING MAJOR COMPONENTS

sea water pressure

pressure compensating diaphragm

Dow
CORNING
DC 200
Fluid

Tool Body
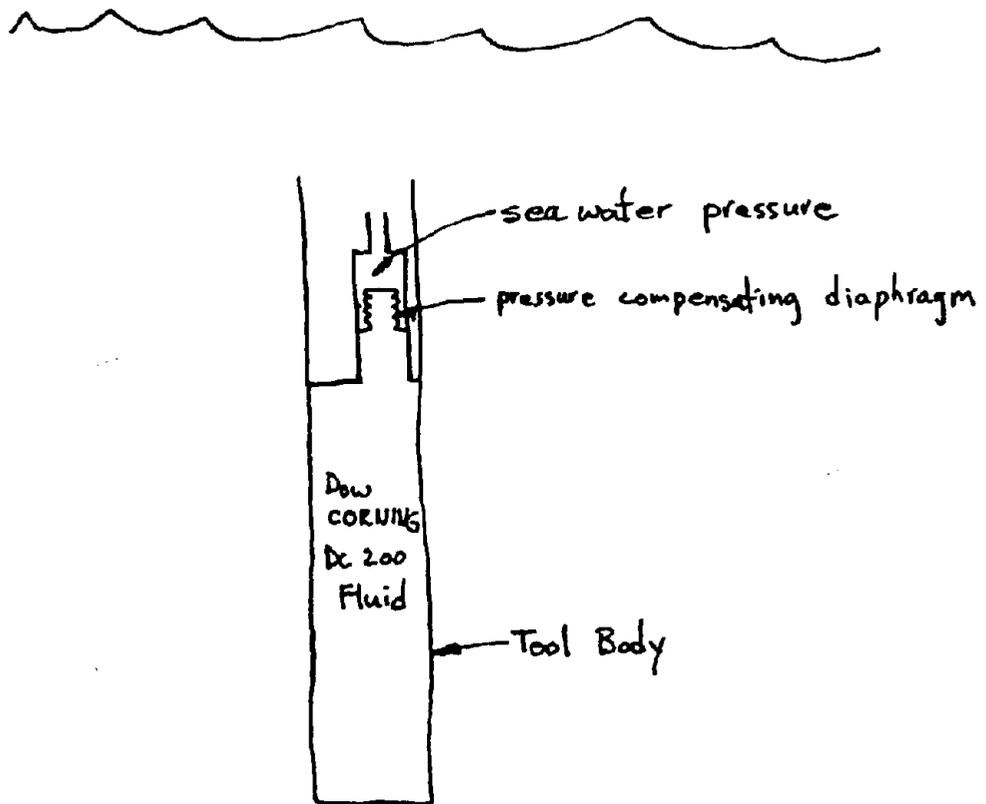
SKETCH 2:   PRESSURE COMPENSATION TECHNIQUE WITH DIELECTRIC FLUID

INSTRUMENTATION FOR LABORATORY TEST PHASE

GEOTECHNICAL TOOL

prepared for

SWEET AND AIKEN, INC.
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by

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July 19, 1985

INSTRUMENTATION FOR LABORATORY TEST PHASE
GEOTECHNICAL TOOL

INTRODUCTION:

This document discusses the instrumentation needed during a lab-
oratory test phase of a Geotechnical Tool designed to obtain
measurements on soils. During this phase the tool will be proto-
typed and connected to various instruments and computers. This
instrumentation set up is not intended to be taken to the field,
except for the Field Computer.

This information is furnished in response to your comments and
those of Ms. H∕enke following the issuance of our report dated
July 12, 1985.

Sketch 1 shows a block diagram of the various instruments and
thier interconnections.

FEATURES OF THE SYSTEM:

1.   Ease of set-up and operation.

2.   Flexibility to meet changing requirements during this
      phase of testing.

3.   Includes analog signal recording on hard copy strip
      chart record for quick observations.

4.   Analog to digital converter and computerized processing
      of data. Allows development of software on the comput-
      er that is suitable for inclusion in the final field
      system.

5.   Majority of components can be either purchased or
      rented.

DISCUSSION:

Although this laboratory instrumentation is more bulky and cum-
bersome than the components that will be eventually used in the
field system(except for the Field Computer), it provides a high
degree of flexibility that is needed during the laboratory test
phase. During this phase it is expected that exact test techni-
ques will evolve along with the software. The use of this system
will in no way degrade the eventual performance of the final
field unit which will include a downhole electronics package.
Likewise the degree of sensitivity and accuracy can be duplicated
with the final field unit if proper care is taken during the
design of the downhole electronics package, umbilical, and sur-
face electronics package.

1

At this time it is not expected that the oscilloscope and strip chart recorder will be required in the final surface electronics package.

FEASIBILITY OF DOWNHOLE ELECTRONICS PACKAGE:

Based on the known requirements for the Geotechnical Tool, the use of a downhole electronics package is feasible and well within the state of the art for such packages. They are routinely used in well logging tools and, to a certain degree, subsea geotechnical tools such as cone penetrometers. The unit will have to be a custom design to meet the specific requirements of this tool because of size, number of channels, and limited cable conductors, etc. No known off-the-shelf units can meet the specific requirements of this tool.

The package can be designed to fit into the Geotechnical Tool which has an ID of 2.8 inches and operate successfully in the offshore geotechnical environment in water depths up to 1500 feet. Although the exact length of the package cannot be determined at this time, it is expected to be approximately 2 to 4 feet long.

The cost for such a downhole electronics package also cannot be precisely determined at this time, but it is expected that a single unit might range between $25,000 and $40,000. It will probably have to be constructed on a cost plus basis for materials plus design and fabrication costs.

INSTRUMENTATION LIST:

The appended documents list the required laboratory instrumentation along with sources of supply, and costs both for purchase and rentals. An overall cost estimate is included.

It should be noted that the component called servo amplifier shown on sketch 1 is not listed nor costed in this document. It is peculiar to the hydraulic servo valve and actuator that will be used in the prototype tool.

SOFTWARE:

The cost for software development shown on the cost estimate is not intended to be a lump sum amount to cover the cost of all software that might eventually be needed by the laboratory system. It should however be considered reasonable for the development of certain acquisition routines peculiar to this test tool and allow for the training of the test engineer and or technician in further software developments as may be required.

David W. Hughes in Houston is a particularly talented and knowledgeable software development engineer. It is recommended that

he be retained to do this portion of the work.   His services can
either be arranged through EDC or directly.

Signal
Generator

Servo
Amplifier → Servo Valve

Geotechnical Tool →

feed back

strain gage
signal conditioners

Power Supply

← Torque

← Angular Displ.

← Pressure Trans.
← Pressure Trans.
← Pressure Trans.

Power
Supply ← Angular Accel.

Power
Supply ← Angular Accel.

COMPUTER
including
D - A, Terminal,
& Printer

Strip.
chart
Recorder

OSCILLOSCOPE
2 ch.
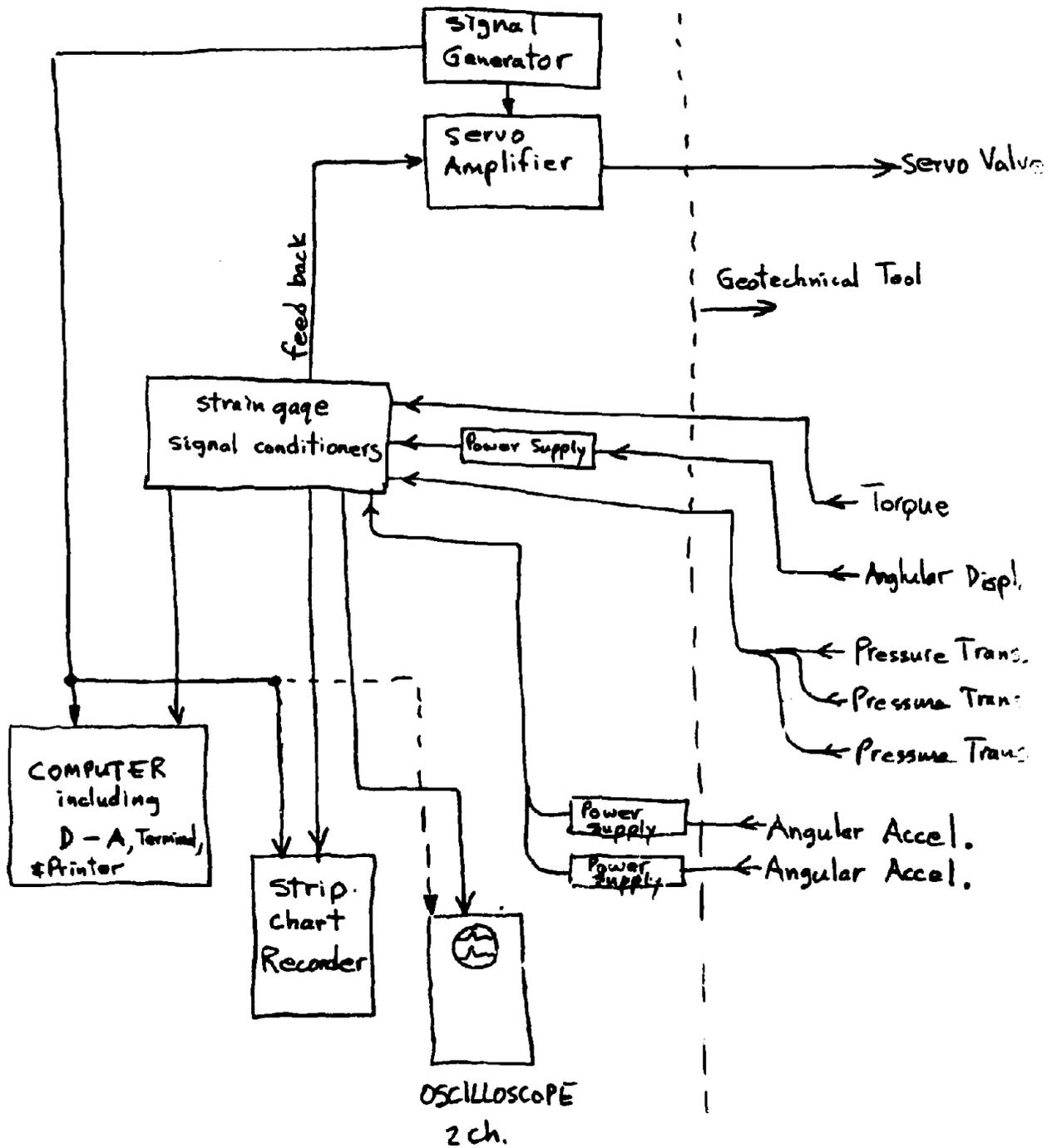
Sketch 1
(also Sketch A-1)

C134

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                    LABORATORY TEST PHASE      18 July 85
                    INSTRUMENTATION COST ESTIMATE     LJM

1.0  MAJOR INSTRUMENT COMPONENTS:              Purchase  Rent/mo.

     Power supply for Angle Trans.               $150

     Power supplies for angle accel.             $500

     Strain gage signal conditioners.           $2700      $350

     Field Computer                             $7200      $650

     Strip Chart Recorder. (used)               $3100      $720

     Oscilloscope.                              $2665      $235

     Signal Generator                           $4075      $440
                                         ---------------------------
                         Purchase Total $20390
                         Rental Total/mo.                  $2395
                                plus purchase              $650

2.0  LABOR COST FOR SET-UP

     Instrumentation Des.--50.00/hr., est. 40 hrs.     $2000.

     Instrumentation Technician--25.00/hr, est 150hrs= $3750.

     Software Eng.--50.00/hr.,est 150 hrs=             $7500.
                                                  ------------
                                                   $13250.
3.0  OTHER EXPENSES

     Shipping and Freight                              $500.

     Telephone                                         $150.

     Travel Expenses(if any) at cost +10%,
     local 21 cents/mi.
          500 mi.                   105
          Lodging,                 ------
          Air Travel               ------
                                  ---------
                                   105 + 10%        $116.

     Misc wire, cable, connectors, lab supplies       $600.
     and digital volt meter, etc.

     Strip Chart paper, floppy discs, printer paper,
     and other expendable supplies                    $600.
                                                  ----------
                                                   $1966.
```

INSTRUMENTATION LIST

Item: Power Supply

Nomenclature: D100 Transducer Fixed Voltage Power Supply

Description: Power for use with Trans-Tek Rotary Angular
Transducer. Plugs into 117VAC power and provides the correct
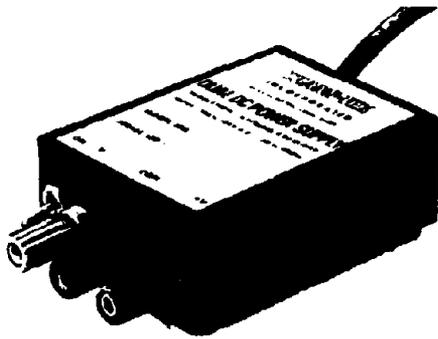stable voltage for the transducer.

Reason for Selection: Economical. Compatible with the transduc-
er. Manufactured by the same manufacturer as the transducer.

Source of Supply: Techanical Products, Inc.
                    1710 S. Dairy Ashford
                    Houston, Texas 77077
                    713 493 6520

Number Required: 1 each

Cost: $150.00 each, Purchase

DC SUPPLIES SERIES D100

TRANSDUCER FIXED VOLTAGE POWER SUPPLIES

☐ Compatible with Trans-Tek DC Excited Transducers
☐ Regulation: 01% Line, 05% Load
☐ Short circuit protected
☐ Line cord 3-wire

DESCRIPTION

Trans-Tek's power supplies are precise, regulated, preset voltage sources, ideally suited for DC excited transducers, discrete and IC function modules. All components are encapsulated to provide, reliable continuous operation to rated specifications in laboratory and factory environments.

They are equipped with an on-off switch, line cord, binding post terminals, rubber feet and two #4-40 tapped holes spaced 1.00 inch apart, located on the bottom side for permanent mounting. The holes are connected to earth ground via the power line cord ground terminal.

Dual voltage power supplies allow the user to choose the transducer excitation voltage; for example, Model D12 100 can be used to power a transducer with 12 VDC or 24 VDC. To excite the transducer with 12 VDC, connect the transducer +INPUT and −INPUT to the power supply +V and COM terminals, respectively (or to the power supply COM and −V terminals, respectively). To excite the transducer with 24 VDC, connect the transducer +INPUT and −INPUT to the power supply +V and −V terminals, respectively.

COMMON SPECIFICATIONS

Input: 105 to 125 V.A.C., 50 to 400 Hz*
Output Voltage tolerance: ±1% (Fixed)
Regulation: Line 0.01% max
　　　　　　Load 0.05% max
Temp. Coef. 0.02%/°C Typical
Ripple and Noise: 1mv RMS max
I/O Isolation: 50 MEGOHMS
Overcurrent Protection
　　Current Limiting: Either output to
　　common or other output, Indefinitely
Operating Temp. Range: −25° to +71°C

Storage Temperature: −40° to +85°C
Termination:
　　Line cord, 3-wire, 5 feet long, terminated in line plug with
　　earth ground.
Output Terminals: 3-Binding post, 6-way
　　Red - Positive Voltage
　　Black - Common
　　Violet - Negative Voltage
Dimensions: 2.5" Wide x 4.45" Long x 1.25" High
(83 x 113 x 32mm)
Approx. Weight: 1.25 lbs (575 gm.)

*210 to 250 VAC optionally available by adding K to model number, e.g. D12 100 becomes D12 100K

Output V.D.C	±12	±12	±15	±15
Current: MA	100	200	100	200

ORDERING

Orders should be made out to TRANS-TEK, INC., sent in care of your local Trans-Tek representative, or directly to Box 338, Route 83, Ellington, Conn. 06029 Tel. (203) 872-8351

INSTRUMENTATION LIST

Item: Power Supply

Nomenclature: PCB Piezotronics Model 482A Line Power Unit.

Description: Power supply for PCB Accelerometers. Plugs into
117VAC power and provides the correct stable excitation voltage
for the accelerometers.

Reason for Selection: Economical. Compatible with the
transducers. Manufactured by the same vendor as the
accelerometers.

Source of Supply: Technical Products, Inc.
                   1710 S. Dairy Ashford
                   Houston, Texas 77077
                   713 493 6520

Number Required: 2 each

Cost: $250.00 each, Purchase

Total Cost: $500.00

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LINE POWER UNIT

for voltage-mode transducers   **Model 482A**

PCB

PIEZOTRONICS

- powers transducers with built-in or attached amplifiers
- supplies power over signal lead
- eliminates bias on output
- monitors normal or faulty system operation
- provides adjustable constant-current excitation

For powering low-impedance piezoelectric transducers with built-in or attached amplifiers and coupling them to versatile readout instruments; and especially for driving long transducer cables.

Model 482A Line Power Unit powers the transducer over the signal lead and couples self-amplifying PCB transducers to oscilloscopes, recorders or other readout instruments. In the coupling process, it eliminates D.C. power bias from the output by means of a coupling capacitor. It also monitors normal operation of the system with a meter that detects such faults as cable and connector open or short circuits and transducer amplifier trouble. With a 24V D.C. regulator, Model 482A provides an output signal range linear to ± 10 volts. Constant current excitation adjustable from 2 to 20 mA (factory set at 12 mA) is supplied to transducer to improve linearity and cable driving capabilities. Higher magnitude supply currents drive longer cables and provide more available output current. With medium length cables, output current is generally available to within 2 mA of the supply current.

Like the transducer discharge circuit, the capacitive coupling circuit to the readout instrument also eliminates D.C. signal components at an exponential rate. When the readout load impedance is one megohm or more, the coupling time constant exceeds 10 seconds, which is sufficiently long for most applications.

For static calibrating or other purposes, Model 482A can be operated in a D.C. mode by taking the output signal directly off the transducer lead through a "T" connector or by shorting across the internal coupling capacitor. The resulting 11V D.C. bias on the output lead can be eliminated, if necessary, by a series battery or floating D.C. supply.

SPECIFICATIONS: Model No.		482A
Transducer Excitation	VDC	+22
Excitation Current (adjustable)[1]	mA	2 to 20
Voltage Gain		1
Coupling Capacitor	µF	10
Output Signal, F.S.	± volts	10
Output Current	mA	to 10
Noise, Wideband (pk-pk max.)	µV	500
Size	in	1.8x4.3x6
Weight	lb	2
Transducer Connector	micro	10-32
Output Connector		BNC
Power Cord (3-wire)	ft	6
Power Required (60 Hz)	V/A	100-125/0.3

OPTIONAL MODELS:	
Four Channel Version	482A04
Fixed Constant - Current Supply (2mA Diode)	482A07
Three Position Gain Switch (X1, 10, 20) (with 4mA Constant-Current Supply)	482A10

NOTE (1) Factory set at 4 mA.
(2) Output Cable not supplied. Order Model 012A03

BLOCK DIAGRAM:
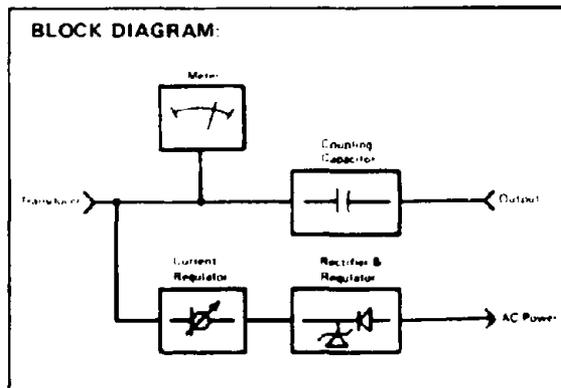
INSTRUMENTATION LIST

Item: Strain Gage Signal Conditioner

Nomenclature: Vishay Multi channel Signal Conditioner/Amplifier
2100 System. Configured with power supply and 6 channels of
signal conditioning.

Description: Unit accepts strain gage transducers and others,
amplifies the signals to appropriate levels for recording and
processing. Unit includes calibration capability.

Reason for Selection: Easy to set up and operate.   Economical.
Provides extra back-up channels for flexibility in laboratory
test program.   Universal type for use with wide range of other
instrumentation and computers needed in the lab test program.

Source of Supply: (Purchase)
                     Technical Products, Inc.
                     1710 S. Dairy Ashford
                     Houston, Texas 77077
                     713 493 6520

                     (Rental)
                     General Electric, Quick-Rental Service
                     900 South Loop West,Ste 180
                     Houston, Texas 77054
                     713 741 8137
                     1 800 GE RENTS

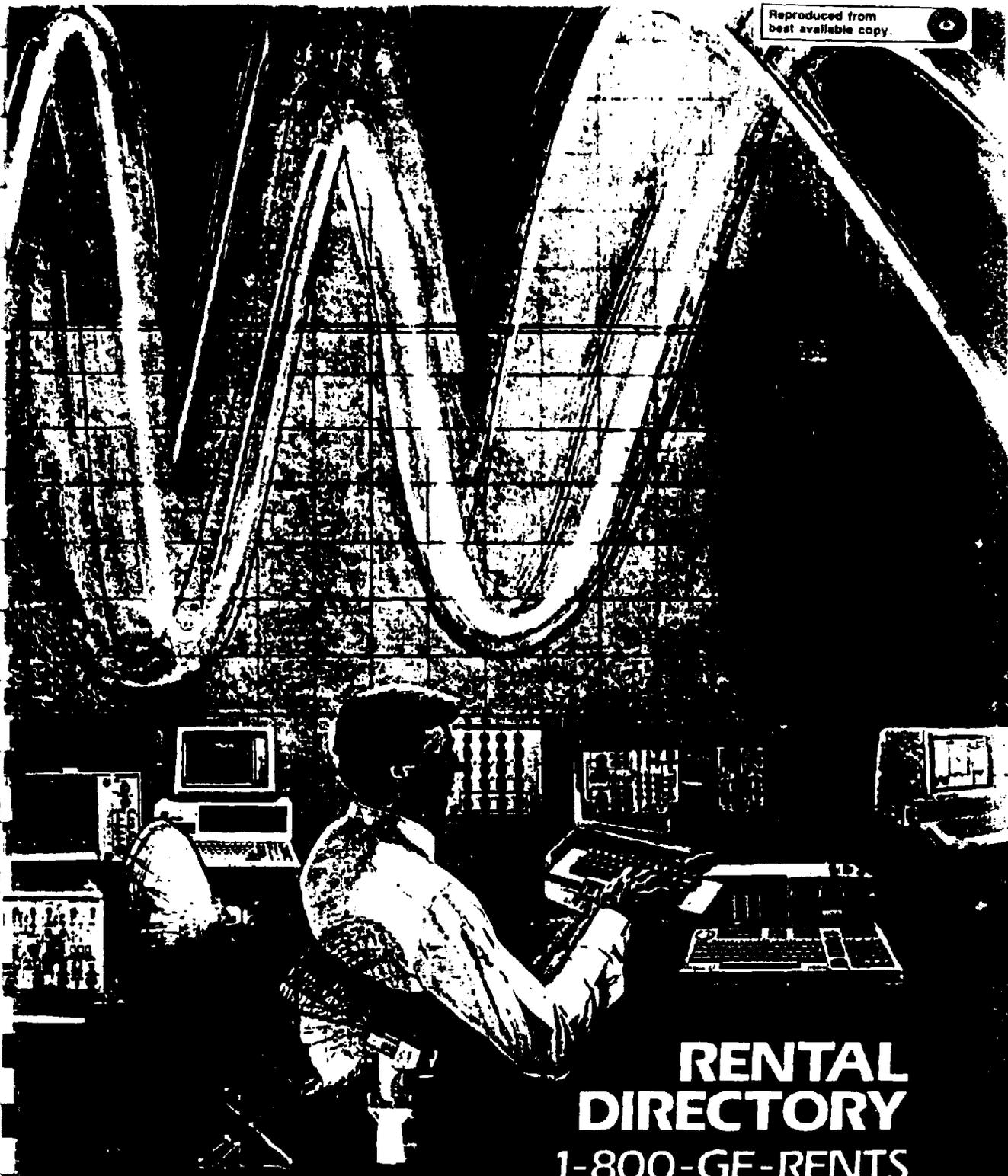Number Required: 1 Unit.

Cost: $2700.00 (estimated), each, Purchase.

        $350.00/month, Rental.

Pages C141-C146 have been removed.

Due to copyright restrictions, pages C141-C146 have been
omitted.

RENTAL
DIRECTORY
1-800-GE-RENTS

GENERAL ⚡ ELECTRIC

QUICK-RENTAL
service

Signal Conditioners (Continued)

Gould 13-4715-09 $ 60

Med. gain multi-span dc amp. Measurement range 400 μV FS to 100 V FS; in 20 steps; accuracy ±0.1% FS; zero suppression ±full scale.

Gould 13-4715-40 $ 110

Thermocouple amplifier type J. Range −150° to 1000° F or °C. Zero suppression.

Gould 13-4715-41 $ 110

Thermocouple amplifier type K. Range −200° to 1000° F or °C. Zero suppression.

Gould 13-4715-42 $ 110

Thermocouple amplifier type T. Range −200° to +400° F or °C. Zero suppression.

Gould 13-4715-43 $ 110

Thermocouple amplifier type E. Range 0° to +1000° F or °C. Zero suppression.

Gould 13-4715-44 $ 125

Thermocouple amplifier type R. Range +500° to +1760° F or °C. Zero suppression.

Gould 13-4715-45 $ 125

Thermocouple amplifier type S. Range +50° to +1760° F or °C. Zero suppression.

If you have any questions,
Just ask our rental specialists.

Honeywell Accudata 117-06 $ 210

Driver amplifier. Gain 0.01-10 in 10 steps, 0.01-25 with vernier, input ±100 mV to ±300 V, 6 channels.

Honeywell Accudata 118-1 $ 95

Strain gage excitation, control and amplification in a single unit: constant voltage or current gage excitation.

Honeywell Accudata 120 $ 70

Differential dc amplifier. Input 2.5 mV to 250 mV; output ±2.5 V, 65 mA, gain 10-1000 X, 8 steps, can be used with 105-2 above.

Honeywell Accudata 218-1 $ 140

Gage control amplifier. Selectable gage excitation voltages, three independent signal outputs, calibrated input suppression.

Soltec 1280 SM26 $ 70

High speed input module for use with SU-100 transient storage unit and 1286 recorder. Frequency response 5 kHz. Measurement range 50 mV to 200 V in 12 steps. ±100% zero adjust. Overall accuracy ±0.7% of full scale at 23°C.

Soltec 1280 VM86 $ 50

Multi-span dc plug-in module for use with 1286 recorder. Measurement range 1 mV to 200 V in 17 ranges. +100% zero adjust. ±400 zero suppression. Overall accuracy ±0.25% of full scale at 23°C.

Soltec 3410 4965 $ 115

Plug-in dc preamplifier for 3400 series recorder. 1 mV/cm to 20 V/cm in 19 ranges. Calibrated zero suppression.

Soltec SU-100 $ 600

32 k word (12 bit) 6 channel storage module allows transient/waveform data to be stored and output to 1286 recorder. Sampling frequency from 200 kHz to 200 Hz. Internal or external triggering, pre and post triggering and an AUTO mode that samples data, prints out data and resets system.

Vishay 2310 $ 155

Signal conditioning amplifier for strain gages, transducers, potentiometers and dc displacement transducers. All bridge completion built-in including 120 and 350 Ω dummies. Bridge excitation selectable 0.5-15.0 Vdc. Continuously variable and calibrated amplifier gain: 1.0-11,000. Electronic bridge balance. Three simultaneous buffered outputs: ±10 V @ 5 mA, tape output ±1.414 V @ 5 mA and galvanometer output ±10 V @ 70 mA.

Vishay 2350 $ 55

10-channel rack adapter for Vishay 2310 amplifiers

Vishay 2360 $ 55

Portable cabinet for 1 to 4 channels of Vishay 2310 amplifiers.

Watanabe AH 3101 $ 55

High gain amplifier. Range 4 mV FS to 500 V FS: vernier gain control.

Watanabe AL3101 $ 35

Low gain amplifier. Range 20 mV FS to 400 V FS. Vernier gain control.

Watanabe AS3101 $ 75

Differential amplifier with zero suppression. Range 20 mV FS to 80 V FS: calibrated zero suppression; selectable 3 pole input filter; vernier gain control.

LIGHT BEAM OSCILLOGRAPHS

Honeywell 1508 B $ 650

24-channel, 4 event, visicorder with dc to 25 kHz response, wide range (1200:1) dc servo record drive system, 42 speeds, 0.1-120 ips drive speeds, internal record take-up, accentuated tenth time line; requires galvanometers.

Honeywell 1858 $ 900

Up to 18 channels, expandable to 32 channels with Model 1870 amplifier housing. Fiberoptic CRT visicorder, dc to 5 kHz response at up to 7.2" trace amplitude, 1 mV to 300 V input sensitivity, 300 V common mode, 42 speeds from 0.1-120 ips, time interval marker, time lines recorder at 5 selectable intervals with 10-line accentuation, better than 0.5% FS accuracy. Requires plug-in signal conditioner next column.

INSTRUMENTATION LIST

Item: General Purpose Storage Oscilloscope.

Nomenclature: Tektronix T912 Dual trace, storage oscilloscope, or equivalent.

Description: General purpose oscilloscope that can be used for general viewing of signals and for set up and calibration of the instrumentation system. Allows for the quick and easy look see of specific signals.

Reason for Selection: Economical. General purpose oscilloscope.

Number Required: 1 each.

Source of Supply: General Electric Quick Rental Service
900 South Loop West, Ste. 180
Houston, Texas 77054
713 741 8137   Rent
1 900 GE RENTS

Tektronix, Inc.   Purchase
P. O. Box 4309
Houston, Tx 77210
713 933 3000

Cost: $235.00/month rental.

$2665.00 Purchase

QUICK-RENTAL
service

DESCRIPTION	RATE/MO.
Portable Oscilloscopes (Continued)	
Tektronix 475	**$ 440**

DC to 200 MHz, dual trace, 2 mV/div to 5 Vdiv; time base 0.01 μs/div to 0.5 s/div, 10×magnifier, calibrated sweep delay, 8 × 10 cm CRT, trigger view.

Tektronix 475A   **$ 475**

DC to 250 MHz at 5 mV/div, otherwise very similar to model 475.

Tektronix 475DM44   **$ 495**

Model 475 with 3½-digit multimeter and temperature probe, Δ delayed sweep with direct numerical readout.

Tektronix 485   **$ 750**

DC to 350 MHz, dual trace, 5 mV/div to 5 V/div; time base 0.5 s/div to 1 μs/div; 8 × 10 div display. 0.8 cm/div CRT.

Tektronix 2213   **$ 120**

DC to 60 MHz, dual trace, 2mV/div sensitivity; time base 0.05 μs to 0.5 s/div, delayed sweep. Lightweight and portable.

Tektronix 2215   **$ 150**

Same as Tektronix 2213, except allow simultaneous viewing of delayed and main waveform. Calibrated delayed sweep.

Tektronix 2235   **$ 175**

Two-channel, dc to 100 MHz bandwidth oscilloscope. Trigger view, dual time base, delayed sweep and X-Y operation.

Tektronix 2236   **$ 275**

Same oscilloscope specifications as the 2235 but with built-in 100 MHz, microprocessor-controlled counter/timer/multimeter. Totalize, dc volts, true rms volts, Δ time, frequency and period.

Tektronix 2335   **$ 265**

DC to 100 MHz, vertical channel deflections from 5 mV/div to 5 V/div, 10×magnifier to increase sweep rate to 5 ns/div, dual trace and delay sweep.

Tektronix 2336   **$ 275**

Same as 2335 with Δ time.

Tektronix 2337   **$ 315**

Same as 2335 with Δ time/DMM.

Tektronix 2445   **$ 330**

150 MHz bandwidth 4-channel oscilloscope. 1 ns/div sweep speed. 2 mV/div vertical sensitivity. CRT readout: scale factors, trigger level, voltage, time, frequency, phase and ratio measurements and mode indicators.

Tektronix 2465   **$ 480**

Same as 2445 but with 300 MHz bandwidth.

Tektronix SC501   **$ 115**

DC to 5 MHz, single trace, 10 mV/div to 10 V/div, time base 1 s/div; 5 × magnifier, 6 × 10 div, 0.203"/div CRT, requires TM500 power module.

DESCRIPTION	RATE/MO.
Tektronix SC502	**$ 235**

DC to 15 MHz; dual trace, 1 mV/div to 20 V/div, time base 0.5 s/div to 0.2 μs/div 10×magnifier, trigger view, enhanced automatic trigger. Requires TM 500 power module

Tektronix SC503   **$ 345**

DC to 10 MHz, dual channel bistable storage scope. 5 mV/div to 20 V/div; time base 2 s/div to 0.5 μs/div, 10 × magnifier. Variable enhancement and integration, auto-erase and X-Y capability. Requires TM500 power module.

Tektronix SC504   **$ 330**

DC to 80 MHz; dual trace; 5 mV/div to 10 V/div; time base 0.2 s/div to 0.2 μs/div; 10 × magnifier; X-Y capability. 8 × 10 div, 0.25"/div CRT; requires TM500 multicompartment power module.

Tektronix T912   **$ 235**

DC to 10 MHz, dual trace, bistable storage scope. 2 mV/div to 10 V/div; time base 0.5 s/div to 0.5 μs/div, 10 × magnifier, 8 × 10 cm CRT, 250 cm/ms stored writing rate.

Tektronix T922   **$ 150**

DC to 15 MHz, dual trace, 2 mV/div to 10 V/div, time base, 0.5 s/div to 0.2 μs/div, 10 × magnifier, 8 × 10 cm CRT, 110 Vac operation.

Tektronix T935   **$ 230**

DC to 35 MHz, dual trace, 0.5 s/div to 0.1 μs/div, 10 × magnifier, 2 mV/div to 10 V/div, 8 × 10 cm CRT, delayed sweep, 110 Vac operation.

Vu Data PS935/975   **$ 275**

Miniscope with DMM and counter, dual trace, dc to 50 MHz. Weighs less than 15 lbs. and battery operated.

Full Size

H-P 1980B-811   **$1750**

Oscilloscope measurement system includes 1980OA waveform measurement library, 1986OA digital waveform storage module, 1950A 100 MHz, 2-channel expansion module and 19811A plot/sequence ROM. HP-IB programmable, with 10 × 12 cm CRT display.

H-P 1950A   **$ 260**

Two-channel expansion module. Adds two 100 MHz vertical channels to the 1980. It features continuously calibrated variable deflection factors from 2 mV/div to 10 V/div.

Nicolet Explorer II 2090-2   **$ 740**

DC to 1 MHz, dual channel digital storage scope, complete with 206-2 plug-in amplifier.

Nicolet Explorer III 204-2   **$1200**

Same as Explorer II above except with pen output, diskette, binary digital I/O. Also includes 204-2, 2-channel plug-in and 2081 IEEE-488 interface.

INSTRUMENTATION LIST

Item: Signal Generator

Nomenclature: Hewlett Packard HP3325A Function Generator.

Description: General purpose signal generator to provide command signals to the servo actuator. Can command sinewaves, squarewaves, triangle waves, and ramps.

Reason for Selection: General purpose unit having the capability to meet the expected command functional requirements.

Number Required: 1 each.

Source of Supply: General Electric Quick Rental Service
                     900 S. Loop West, Ste. 180
                     Houston, Texas 77054
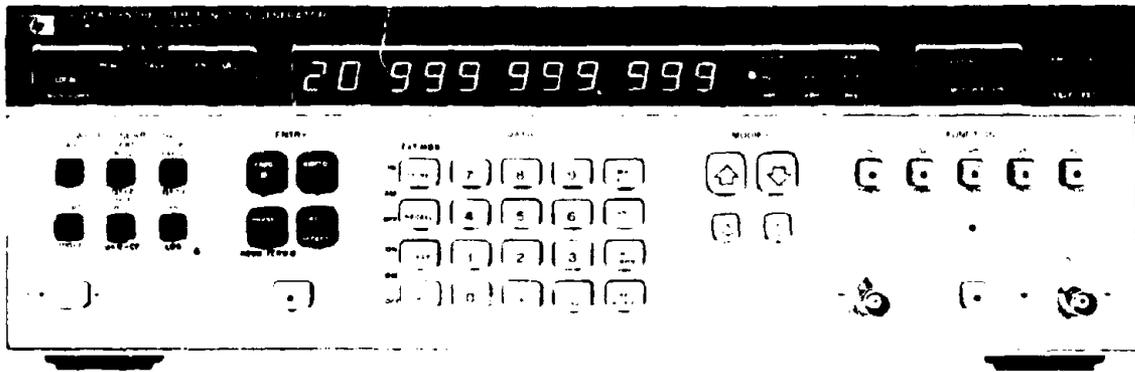   713 741 8137
                     1 800 GE RENTS   Rent

                     Hewlett Packard   Purchase
                     10535 Harwin Drive
                     Houston, Texas 77042
                     713 776 6400

Cost: $440.00/month Rental

        $4075.00 Purchase

Model 3325A

HP 3325A

The HP 3325A Synthesizer/Function Generator is an uncompromising, high performance synthesizer with 11 digit resolution, a function generator with precision waveforms, a wideband sweeper, and a fully programmable systems instrument

The HP 3325A is first with microhertz resolution below 100 kHz along with frequency coverage from .000001 Hz to 20.999 999 999 MHz. Signal purity, accuracy and stability are as good or better than earlier stand-alone HP synthesizers. Harmonics are 65 dB down below 50 kHz and you can externally modulate with AM and PM

The HP 3325A is also a high performance function generator providing precision waveforms with synthesizer accuracy and resolution. Squarewaves to 10.999 999 999 MHz have 20 ns rise and fall times. Triangles and ramps with .05% linearity are available up to 10.999 999 999 kHz. All waveforms can be dc and phase offset

A major contribution is wideband phase continuous sweep, covering up to the full frequency range of each waveform. Sweep log or linear, single or continuous without the phase discontinuities usually associated with synthesizers. Phase lock loop testing is made easier.

Make convenient swept frequency network measurement on filters, amplifiers or any passive or active network. Use the TTL marker to check the frequency of points of interest on a swept frequency display desired. Use the convenient "zoom" functions Δf × 2 and Δf ÷ 2 to quickly change the frequency span for the display desired

All necessary functions are programmable on the HP-IB, including frequency, amplitude, all functions, phase and dc offset, modulation, all sweep parameters, amplitude cal and self-test, making the HP 3325A a very versatile and powerful addition to automatic test systems. The isolated interface combined with floating outputs and inputs and talk mode make the HP 3325A easy to use in Automatic Test Systems

The phase of the output can be changed ± 720° with 0.1° resolution. The phase is advanced (or retarded) with respect to the output

phase. Two HP 3325A units can be phase locked together for dual phase output applications

DC offset is capable of ± 4.5 Vdc on the standard instrument. The high voltage option (Opt 002) allows ac voltages up to 40 Vpp and ac + dc up to ± 18 V total (ac peak + dc)

Ten storage registers can be programmed with ten different combinations of function/parameter settings from the front panel, stored and then recalled

The HP 3325A can display 11 digits of frequency and 4 digits of volts or millivolts from 1 mV to 10 volts peak to peak. Conversion to RMS or dBm is simple with the touch of a button

The HP 3325A provides unprecedented performance per dollar thanks to several major contributions from advances in HP technology. A single loop Fractional-N synthesis technique allows synthesizer accuracy with 11 digits of resolution and, as an added bonus, phase continuous frequency sweep. Fewer parts and integrated circuit technology make the difference. A unique method of triangle and ramp waveform generation provides excellent linearity. Add microprocessor control and Hewlett-Packard Interface Bus (HP-IB) operation and the result is more performance, flexibility and versatility on the bench or in automatic test systems than previously available, and at a lower cost

Specifications
Refer to the HP 3325A data sheet for complete specifications

Sine, Square, Triangle, negative and positive Ramps

Range
 Sine: 1 μHz to 20.999 999 999 MHz
 Square: 1 μHz to 10.999 999 999 MHz
 Triangle/ramps: 1 μHz to 10.999 999 999 kHz
Resolution: 1 μHz — 100 kHz
 1 mHz — 100 kHz
 Aging rate: ± 10⁻⁷/mo
Warm-up time: 20 minutes to within specified accuracy

Impedance: 50 Ω

Connector: BNC; switchable to front or rear panel, nonswitchable with option 002, except by internal cable change

Range: 1 mV to 10 V p-p in 8 amplitude ranges, 1-3-10 sequence (10 dB steps), into 50 Ω load.

Function	Sine		Square		Triangle/Ramps	
Units Displayed	min	max	min	max	min	max
peak-peak	1.000 mV	10.00 V	1.000 mV	10.00 V	1.000 mV	10.00 V
rms	0.354 mV	3.536 V	0.500 mV	5.000 V	0.289 mV	2.887 V
dBm (50 Ω)	-56.02	+23.98	-53.01	+26.99	-57.78	+22.22

Resolution: 0.03% of full range or 0.01 dB (4 digits)

(without dc offset, relative to programmed amplitude and accuracy)

Sinewave Amplitude Accuracy
1 mHz to 100 kHz: ±0.1 dB, ≤3 Vpp, ±0.2 dB, >3 Vpp
100 kHz to 20 MHz: ±0.4 dB, >3 Vpp, ±0.6 dB, 0.1 to 3 Vpp

Squarewave Amplitude Accuracy
1 mHz to 100 kHz: 1%, ≤3 Vpp, 2.2%, >3 Vpp
100 kHz to 10 MHz: 11.1%, ≤3 Vpp, 13.6%, >3 Vpp

Triangle Amplitude Accuracy
1 mHz to 2 kHz: 1.5%, ≤3 Vpp, 2.7%, >3 Vpp
2 kHz to 10 kHz: 5%, ≤3 Vpp, 6.2%, >3 Vpp

Phase noise: -60 dB for a 30 kHz band centered on a 20 MHz carrier (excluding ±1 Hz about the carrier) with high-stability option 001 installed.

Spurious: all non-harmonically related output signals will be more than 70 dB below the carrier (60 dB with dc offset), or less than -90 dBm, whichever is greater.

Sinewave harmonic distortion: harmonically related signals will be less than the following levels (relative to the fundamental) at full output for each range:

Frequency Range	Harmonic Level
0.1 Hz to 50 kHz	-65 dB
50 kHz to 200 kHz	-60 dB
200 kHz to 2 MHz	-40 dB
2 MHz to 15 MHz	-30 dB
15 MHz to 20 MHz	-25 dB

Rise/fall time: <20 ns, 10% to 90% at full output
Overshoot: <5% of peak to peak amplitude, at full output
Settling time: <1 µs to settle to within 0.5% of final value

Range: ±719.9° with respect to arbitrary starting phase or assigned zero phase
Resolution: 0.1°
Accuracy: ±0.2°

Range: dc only (no ac signal) 0 to ±5.0 V/50 Ω
dc + ac: Maximum dc offset ±4.5 V on highest range, decreasing to ±4.5 mV on lowest range
Resolution: 4 digits

Modulation depth at full output for each range: 0-100%
Modulation frequency range: dc to 400 kHz (0-21 MHz carrier frequency)
Sensitivity: ±5 V peak for 100% modulation

Sinewave Phase Modulation
Range: ±850°, ±5 V input
Modulation frequency range: dc-5 kHz

Sweep Time
Linear: 0.01 s to 99.99 s
Logarithmic: 2 s to 99.99 s single, 0.1 s to 99.99 s continuous
Maximum sweep width: full frequency range of the main signal output for the waveform in use, except minimum log start frequency <1 Hz

Phase continuity: sweep is phase continuous over the full frequency range of the main output

Reference input: for phase-locking HP 3325A to an external frequency reference signal from 0 dBm to +20 dBm into 50 Ω. Reference signal must be a subharmonic of 10 MHz from 1 MHz to 10 MHz.

Auxiliary frequency output: 21 MHz to 60.999 999 999 MHz, under range coverage to 19.000 000 001 MHz, frequency selection from front panel; 0 dBm; output impedance 50 Ω.

Sync output: square wave with V (high) >1.2 V, V (low) ≤0.2 V into 50 Ω.

X-Axis drive: 0 to >+10 V dc linear ramp proportional to sweep frequency, linearity, 10-90%, ±0.1% of final value.

Sweep marker output: high to low TTL compatible voltage transition at selected marker frequency.

Z-Axis blank output: TTL compatible voltage levels capable of sinking 200 mA from a positive source.

1 MHz reference output: 0 dBm output for phase-locking additional instruments to the HP 3325A.

* **10 MHz oven output:** 0 dBm internal high stability frequency reference output for phase-locking HP 3325A. (Opt. 001 only)

HP-IB Interface Functions: SH1, AH1, T6, L3, SR1, RL1, PP0, DC1, DT0, C0, E1.

Recommended Accessory: HP 7090A Measurement Plotting System

Aging rate: ±5 × 10⁻⁸/week (72-h warm up); ±1 × 10⁻⁷/month (after 15 days continuous operation).
Ambient stability: ±5 × 10⁻⁸ (0° to +55°C).
Warm-up time: reference will be within ±1 × 10⁻⁷ of final value 15 minutes after turn-on for an off time of less than 24 hours.

Frequency range: 1 µHz to 1 MHz

Range: 4.00 mVpp to 40.00 Vpp (500 Ω, <500 pF load).
Accuracy and Flatness at Full Output
Sine, square, and triangle waves: ±2% at 2 kHz
Ramps: ±2% at 500 Hz
Flatness: ±10% relative to programmed amplitude
Sinewave distortion: harmonically related signals will be the same as the standard instrument to 1 MHz
Maximum output current: 80 mApp.
Output impedance: <2 Ω at dc, <10 Ω at 1 MHz

DC offset range: 4 times the specified range of the standard instrument.

Operating environment
Temperature: 0°C to 55°C.
Relative humidity: 95%, 0°C to 40°C.
Altitude: ≤15,000 ft.
Storage temperature: -40°C to +75°C.
Storage altitude: <50,000 ft
Power: 100/120/220/240 V, +5%, -10%, 48 to 66 Hz; 90 VA, 120 VA with all options; 10 VA standby
Weight: 9 kg (20 lb) net; 14.5 kg (32 lb) shipping.
Size: 132.6 H × 425.5 W × 497.8 mm D (5.25" × 16.75" × 19.63")

HP 3325A Frequency Synthesizer
Opt. 001 High Stability Frequency Reference
Opt. 002 High Voltage Output
Opt. 907 Front Handle Kit (standalone orders P/N HP 5061-0089)
Opt. 908 Rack Flange Kit (standalone orders P/N HP 5061-0077)
Opt. 909 Rack Flange and Handle Combination Kit (standalone orders P/N HP 5061-0083)
*HP-IB cable not supplied. See page 675