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REAPPROVED '79

Pressure Control Systems Used on Steam Turbine-Generator Units

PERFORMANCE TEST CODES

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS
United Engineering Center
345 East 47th Street New York, N.Y. 10017

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FOREWORD

Power Test Code Committee No. 20 on Speed-Responsive Governors was established by the ASME Power Test Codes Committee in 1921, and the Test Code for Speed-Responsive Governors as prepared by this Committee was issued in May, 1927. Subsequently, since it was found that this Code was inadequate, Power Test Code Committee No. 20 was reorganized in 1940 and its scope expanded to include speed, temperature, and pressure-responsive governors for prime movers.

In 1941, the Joint AIEE-ASME (IEEE-ASME) Committee on a Recommended Specification for Prime-Mover Speed Governing was formed. The fundamental studies for prime-mover speed governing were made. The fundamental studies for the preparation of the code for testing were identical with those for the preparation of the recommended specification. Close coordination of the work by both committees to assure the successful completion of their respective assignments was accomplished by the appointment of personnel common to both groups.

By mutual agreement, the work of the Specification Committee took precedence over that of PTC Committee No. 20, whose labors were interrupted during World War II.

In order to facilitate the use of the Code, PTC Committee No. 20 decided in 1946 to issue their assignment in several publications and in the following sequence:

(1) Test Code for Speed-Governing Systems for Steam Turbine-Generator Units
(2) Test Code for Emergency Governors for Steam Turbine-Generator Units
(3) Test Code for Pressure-Regulating Systems for Steam Turbine-Generator Units.

With the issuance of AIEE (IEEE) Publication No. 600 in May, 1949, covering "Recommended Specification for Speed-Governing of Steam Turbines Intended to Drive Electric Generators Rated 500 Kw and Up," by the Joint AIEE-ASME (IEEE-ASME) Committee, the way was cleared to proceed with the preparation of the first code in the series. This Code was issued as PTC 20.1-1958 in November, 1958. In December, 1961, the PTC Committee No. 20 was reorganized in order to write the second part of the Code under the modified title of "Test Code for Overspeed Trip Systems for Steam Turbine-Generator Units," which was approved on March 3, 1965, and issued in June, 1965.

In December, 1965, work on the third part of the Code was begun by the same Committee.

In 1967, the name of the Committee was changed to Performance Test Code Committee No. 20 and the title of the Code in preparation to Performance Test Code No. 20.3.

This Code was approved by the Performance Test Codes Committee on August 26, 1969. It was approved and adopted by the ASME Council as a standard practice of the Society by action of the Board on Codes and Standards on October 28, 1969.

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ASME PERFORMANCE TEST CODES
Test Code for
Pressure Control Systems Used on Steam Turbine-Generator Units

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SECTION 0, INTRODUCTION

0.01 This Code is applicable to steam turbine-generator units of 500 KW capacity and larger and provides:

0.01.1 Recommended format for specifications for the performance characteristics of pressure control systems subject to modifications by mutual agreement between the parties to the test and
0.01.2 Standard procedures for tests to determine the performance of pressure control systems for controlling the·

(a) Initial pressure
(b) Extraction and/or induction pressure
(c) Exhaust pressure
(d) Steam seal pressure

(See Fig. 0-1 for location of these pressures.)

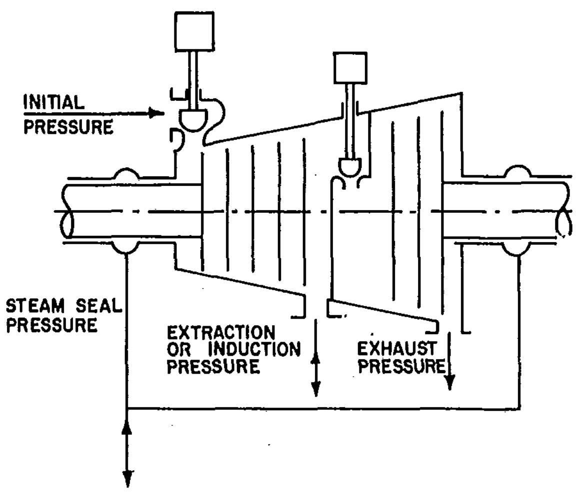

FIG. 0-1 LOCATION OF PRESSURES

0.02 Reference is made to the following codes and standards:

(a) PTC 20.1, 1958, ASME
Power (Performance) Test Code "Speed Governing Systems for Steam Turbine Generation Units."

(b) PTC 20.2, 1965, ASME
Power (Performance) Test Code "Overspeed Trip Systems for Steam Turbine Generator Units."

(c) Standard 600, 1959, AIEE (IEEE)
"Recommended Specifications for Speed-Governing of Steam Turbines Intended to Drive Electric Generators Rated 500 KW and Larger."

(d) NEMA Standard 46-112, 1946
NEMA Standard for "Speed-Governing and Pressure Control of Steam Turbine Generator Units."

(e) American National Standards Institute (ANSI) C85.1-1963
"Terminology for Automatic Control."

(f) SAMA Standard RC 20/11-1964
"Measurement and Control Terminology."

(g) PTC 6 – 1964, ASME
Power (Performance) Test Code "Test Code for Steam Turbines."

(h) PTC 19.2 – 1964, ASME
Instrument and Apparatus:
Part 2, "Pressure Measurements."

(i) PTC 19.5;4 – 1959, ASME
Instrument and Apparatus: Part 5, 4, "Flow Measurements."

SECTION 1, OBJECT AND SCOPE

The purposes of this Code are:

1.01 To establish guiding principles and provide a format for performance specifications for pressure control systems applicable to the equipment covered under this Code.

1.02 To establish test procedures for determining performance characteristics of the subject pressure control systems including:

1.02.1 Pressure set point range.
1.02.2 Rate of change of pressure set point.
1.02.3 Steady state pressure regulation.
1.02.4 Transient response characteristics.
1.02.5 Steady state frequency response.
1.02.6 Dead band and hysteretic error.
1.02.7 Stability band.
1.02.8 Interaction between the variables in a compound control system.

1.03 To establish a procedure for describing the system performance by means of test quantities obtained under Par. 1.02 and the system transfer functions.

PRESSURE CONTROL SYSTEMS USED ON STEAM TURBINE-GENERATOR UNITS

SECTION 2, DEFINITION OF TERMS

2.01 **Systems and Actions** **Table I**

Par.	Term	Description
2.01.1	Pressure control system	A system that controls the pressure at a sensing point in a designated location. Typically includes the pressure sensing element, the controller, the servomotor(s) and the pressure controlling valve(s).
2.01.2	Compound control system	A system that controls two or more variables (speed, load, pressure(s)) by means of interconnected controls with the control functions arranged to minimize interactions.
2.01.3	Controlled pressure	The steam pressure sensed and controlled by the pressure control system.
	The controlled pressures covered by this Code include: (See Fig. 0.1).	
	(a) Initial pressure	Pressure upstream of the controlling valve(s).
	(b) Extraction or induction pressure	Pressure in extraction or induction line.
	(c) Exhaust pressure	Pressure in the exhaust line.
	(d) Steam seal pressure	Pressure in the steam seal line.
2.01.4	Process steam flow	The steam flow that is used (or produced) by the plant at the controlled pressure.
2.01.5	Control action	In a control element or a control system the nature of the output change caused by the input.
2.01.6	Proportional action	The control action that is proportional to the change of the error or other input signal.
2.01.7	Integral action (reset)	The control action that is proportional to the time integral of the error or other input signal.
2.01.8	Derivative action (rate)	The control action that is proportional to the time derivative (rate of change) of the error or other input signal.
2.01.9	Proportional plus integral	The control action of a system in which the output change is proportional to the sum of the error or other input signal plus the time integral of the error or other input signal.
2.01.10	Proportional plus partial integral (reset) action	The control action of a system in which the output change is proportional to the sum of the original error or other input signal plus the time integral of a predetermined fraction of the original error or other input signal.

Note: It is assumed that the equipment is operated in a range where saturation does not occur.

2.01 **Systems and Actions (Cont'd)** **Table I**

Par.	Term	Description
2.01.11	Saturation	The condition of a control element in which any further change in input no longer results in a corresponding change of output (ANSI C85-1).
2.01.12	Unit in normal operation	Unit running at load, within design limits and with none of the operating control devices in saturation.

2.02 **Components** **Table II**

Par.	Term	Description
2.02.1	For automatic pressure control systems the following terms and description supplement the definitions in PTC 20.1.	
2.02.2	Pressure sensor	Pressure responsive element that produces a signal which is a linear function of the input pressure.
2.02.3	Pressure controller	Device that generates one (or more) actuating signal(s) in response to pressure set point and feedback signals.
2.02.4	Pressure set point changer	Device for producing the pressure reference signal to the pressure controller in response to a manual (or automatic) adjustment.
2.02.5	Valve servomotor	Amplifying device that moves the steam valves in response to the actuating signal(s) from the controller(s).
2.02.6	Steam valve(s), controlling	Device with variable flow area adjusted by the mechanical motion of the servomotor to control steam flow.
2.02.7	Plant	That portion of the system responsive to steam flow in producing the controlled pressure.

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2.03 **Definitions** **Table III**

Par.	Term	Symbol	Definition	Units
2.03.1	Angle, phase-	$\phi(\omega)$	The angular difference between the output and the input of a control loop when the input is sinusoidally oscillated at the frequency ω [rad/sec].	degrees
Note: A positive sign is used for a phase lead, a negative sign for a phase lag.				
2.03.2	Change (in general)	Δ	The difference in the values of a variable between two specified operating conditions	
2.03.3	Dead band, pressure-	ΔP_d	The range through which the controlled pressure can be varied without causing a measurable position change of the pressure controlling valve(s).	psi
2.03.4	Dead band, relative pressure-	ψ_d	The dead band referred to rated pressure.	per cent
2.03.4-1			$\psi_d = \Delta P_d \times \frac{100}{P_r}$	per cent
2.03.5	Deviation, maximum transient- (pressure)	MTD	The difference between the largest deviation of the output and the new steady state value after the transient has subsided when responding to a step input of limited magnitude. See Fig. 5-6.	psi
2.03.6	Flow, steam-	Q	Steam flow at a specified location in the system.	lb/hr
2.03.7	Flow, rated-	Q_r	Rated steam flow at a specified location in the system.	lb/hr
2.03.8	Flow change, relative-	μ	Change in steam flow at a specified location in the system between two identified operating conditions, referred to rated flow.	per cent
2.03.8-1			$\mu = \Delta Q \times \frac{100}{Q_r}$	per cent
2.03.9	Frequency, attenuation- (cut-off frequency.	ω_a	The frequency at which the magnitude ratio (G) is $1/\sqrt{2}$, (-3db).	rad/sec
2.03.10	Frequency, gain crossover-	ω_c	The frequency at which the magnitude ratio (G) is unity; (in a closed loop after the resonance peak) See Fig. 5-8	rad/sec

2.03 **Definitions (Cont'd)** **Table III**

Par.	Term	Symbol	Definition	Units
2.03.11	Function, open loop system transfer. (in Laplace transform)	$G(s)$	The ratio of the feedback signal to the error signal of a feedback control system, when both of these signals are expressed in Laplace transform.	complex dimensionless
2.03.12	Function transfer- (time domain)	$f(t)$	A mathematical statement relating the output $O(t)$ of a control or process element to its input $I(t)$ as a function of time and the initial state of such element(s). Ref: ANSI C85.1.	
2.03.13	Function, transfer- in Laplace transform	$F(s)$	The quantity obtained by performing the Laplace transform on the transfer function $f(t)$, usually with zero initial conditions.	
2.03.13.1			$F(s) = \mathcal{L}(f(t)) = \frac{\mathcal{L}(O(t))}{\mathcal{L}(I(t))}$	
2.03.14	Gain of pressure control system	G_P	The reciprocal of the relative pressure regulation expressed in per unit.	per unit
2.03.14-1			$G_P = \frac{100}{\delta_P}$	
2.03.15	Hysteretic error	ΔP_h	The maximum pressure difference between the increasing and the decreasing pressure that produces identical valve position (does not apply to integral control). See Fig. 5-4.	psi
2.03.16	Hysteretic error, relative-	ψ_h	The hysteretic error referred to rated pressure.	per cent
2.03.16-1			$\psi_h = \Delta P_h \times \frac{100}{P_r}$	per cent
2.03.17	Interaction gradient	$I(V_1, V_2)$	In a compound control system the ratio of the relative change of a first controlled variable (V_1) to the relative change of a second independent variable (V_2) when the control settings are constant and the independent variable is varied by a measured amount within 5 per cent and 95 per per cent of its rated operating range.	dimensionless

2.03 **Definitions (Cont'd)** **Table III**

Par.	Term	Symbol	Definition	Units
2.03.17-1			$I(V_1, V_2) = \dfrac{\frac{\Delta V_1}{V_{1_r}}}{\frac{\Delta V_2}{V_{2_r}}}$ For instance, on a single automatic extraction turbine supplying power to an independent system, the ratio of the relative speed change (σ) to the relative extraction flow change (μ) at constant load.	dimensionless
2.03.17-2			$I(\sigma,\mu) = \dfrac{\sigma}{\mu}$	dimensionless
2.03.18	Laplace transform	$\mathfrak{L}(f(t))$	The quantity obtained by performing the Laplace transform on the function $f(t)$.	
2.03.18-1			$\mathfrak{L}(f(t)) = \int_0^{\infty} f(t)\ e^{-st}\,dt$	
2.03.19	Laplace variable	s	The complex variable of the function $F(s)$ in Laplace transform.	1/sec
2.03.19-1	(Differential)		$s = \dfrac{d(\ )}{dt}$ conditionally, for solution of linear differential equation with constant coefficients and zero initial conditions.	
2.03.19-2	(Complex frequency)		$s = j\omega$ for frequency response analysis.	
2.03.20	Magnitude of oscillation	M	The difference between highest and lowest point of a sinusoidal oscillation of a quantity such as pressure, flow, etc.	psi, lb/hr, etc.
2.02.21	Magnitude ratio	G	The ratio of the magnitudes of oscillation of the output (M_O) to the input (M_I) of an open control loop.	per unit

2.03 **Definitions (Cont'd)** **Table III**

Par.	Term	Symbol	Definition	Units
2.03.22	Margin, phase-	γ_c	The margin with respect to -180° of the phase angle ϕ_c of an open control loop when the input oscillation has the frequency that results in a magnitude ratio of output/input of unity (the crossover c: see Appendix 7.03).	degrees
2.03.22-1			$\gamma_c = 180 - \phi_c$ where ϕ_c = phase angle at the cross-over frequency ω_c.	
2.03.23	Pressure, controlled-	P	The pressure that is held substantially constant by the pressure control system.	psi (a)
2.03.24	Pressure, rated-	P_r	The specified value of the controlled pressure for normal operation.	psi (a)
2.03.25	Regulation, steady state pressure-	R_P	The sustained pressure change that will actuate the pressure control system from maximum flow at rated pressure to minimum flow with a constant pressure set point.	psi

Note: For an extraction control system: the pressure change from rated extraction flow to zero extraction flow. For an extraction/induction pressure control system: the pressure change from rated extraction flow to rated induction flow.

Par.	Term	Symbol	Definition	Units
2.03.26	Regulation, relative steady state pressure-	δ_P	The value of the steady state pressure regulation (R_P) referred to rated pressure.	per cent
2.03.26-1			$\delta_P = R_P \times \dfrac{100}{P_r}$	per cent
2.03.27	Regulation, steady state incremental pressure-	R_i	The rate of change of the steady state pressure with respect to relative change in controlled steam flow at a given steady state operating point with constant pressure set point. (See Fig. 5-3 and Par. 3.02.2 for applicability.)	psi/unit
2.03.28	Regulation, relative steady state incremental	δ_i	The steady state incremental pressure regulation referred to rated pressure.	per cent

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2.03 **Definitions (Cont'd)** **Table III**

Par.	Term	Symbol	Definition	Units
2.03.28-1			$\delta_i = R_i \times \frac{100}{P_r}$	per cent
2.03.29	Resonance peak	G_R	The maximum magnitude ratio of the controlled variable/input oscillation of a closed loop control system occurring at the resonance frequency ω_R (See Fig. 5-8).	db
2.03.29-1			$G_R = 20 \times \log \left[G(\omega_R)\right]$	db
2.03.30	Response, frequency	$G(j\omega)$	The ratio of the output to the input of a control element or system as a function of the angular frequency of a sinusoidal oscillation impressed at the input.	complex
2.03.30-1			$G(j\omega) = \frac{O(j\omega)}{I(j\omega)}$	
2.03.31	Response, step-	–	The output change as a function of time which results from a step disturbance at a specified point in the system.	
2.03.32	Set point, pressure-	P_S	The value of the controlled pressure to which the set point device is adjusted.	psi (a)
2.03.33	Set point range, pressure-	ΔP_S	The algebraic difference between maximum and minimum set point.	psi
2.03.34	Set point range, relative pressure-	ψ_S	The pressure set point range referred to the rated pressure.	per cent
2.03.34-1			$\psi_S = \Delta P_S \times \frac{100}{P_r}$	
2.03.35	Set point, rate of change of-	$\frac{dP_S}{dt}$	The rate at which a remotely operated set point can be adjusted.	psi/sec
2.03.36	Set point, relative rate of change of-	$\frac{d\psi_S}{dt}$	The rate of change of pressure set point referred to rated pressure.	per cent/sec

2.03 **Definitions (Cont'd)** **Table III**

Par.	Term	Symbol	Definition	Units
2.03.36-1			$\frac{d\psi_S}{dt} = \frac{dP_S}{dt} \times \frac{100}{P_r}$	per cent/ sec
2.03.37	Stability, pressure control system-	–	The capability of a system to position the pressure controls so that any sustained oscillations of the system pressure or of the energy input to the turbine does not exceed a specified value under steady state pressure (or load demand) or following a change to a new steady state pressure (or load demand).	
2.03.38	Stability band, pressure-	ΔP_b	The peak-to-peak magnitude of sustained oscillation of the controlled pressure when the system is operated under steady state load conditions.	psi
2.03.39	Stability band, relative pressure-(stability index)	ψ_b	The stability band referred to rated pressure.	per cent
2.03.39-1			$\psi_b = \Delta P_b \times \frac{100}{P_r}$	
2.03.40	Steady state	–	The state of a variable where its average value exhibits only negligible change over an arbitrarily long interval of time.	–
2.03.41	Time constant	T	For a single time constant system, the time in which the output reaches 63.2 per cent of the final output change, in response to a step input of given limited magnitude. REF: ANSI C85.1. For typical transfer functions using T, see Section 7.	sec
2.03.42	Time, dead-	T_d	The interval of time between initiation of an input change or stimulus and the start of the resulting response. See Fig. 5-6. REF: ANSI C85.1; SAMA RC 20-11-1964.	sec

2.03 **Definitions (Cont'd)** **Table III**

Par.	Term	Symbol	Definition	Units
2.03.43	Time, response-	T_R	The time required for the output to reach 95 per cent of the new steady state value following a limited step input change. See Fig. 5-6.	sec
2.03.44	Time, settling-	T_S	The time required for the output to enter and remain within the pressure stability band following a step input change of limited magnitude. See Fig. 5-6.	sec

SECTION 3, GUIDING PRINCIPLES

3.01 Advance Planning and Preparation for Test

3.01.1 The plan of the test and the procedure shall be agreed upon in advance by the parties to the test and shall be confirmed in writing.

3.01.2 Test procedures shall include statements describing the manner in which the unit will be operated, and the provisions made for transfer of load.

3.01.3 Dimensions, data, drawings and diagrams of the system and components which are required for calculation purposes or inclusion in the report shall be obtained.

3.02 Items on Which Agreement Shall Be Reached

3.02.1 Definite agreement shall be reached as to the specific object of the test enumerating the system characteristics and responses that are to be tested. Specified performance of the system shall be part of such an agreement. If such specifications are not available the specification format in 3.02.4 may be used for preparing specifications.

3.02.2 If the specifications include incremental pressure regulation on an extraction turbine one must recognize the infinite number of combinations that result from the interaction of two or more controlled steam flows. For every value of electrical load on the turbine there is a unique and particular curve for incremental pressure regulation as the pressure controlled flow changes from maximum to minimum. To test for all load points would be an infinite job. For a single extraction turbine high values of incremental pressure regulation will be experienced each time both valve sets are operating just before the cracking point of their respective next valves and the lowest values will follow immediately after the simultaneous cracking of these valves. The problem will exist on any system controlling two or more flows.

3.02.3 Definite agreement should be reached to state pressures consistently either in "gage" (psig) or "absolute" (psia) values. This is particularly important when the unitized system is used for calculations.

3.02.4 Recommended Standard Specification Format

(a) Location(s) of pressure tap(s) for measuring controlled pressure(s)

(b) Rated Pressure (set point).....psi ( )
(specify gage or absolute)

(c) Pressure Set Point Range.....to.....psi ( )
or Rated Pressure +.....psi or %
−.....psi or %

(d) Pressure Set Point Limits Adjustable
yes, no

(e) Steady State Pressure Regulation
R_p.....psi or.....%
If adjustable.....psi min to.....psi max
or.....% min to.....% max
Adjustability: discrete steps or continuous.
Adjustable in operation: yes, no

(f) Stability Band.....psi max

(g) Stability Index.....% max

3.02.5 Recommended Optional Specification Format for special equipment or ciritical pressure control applications

(a) Rate of Change of Pressure Set Point
(applicable for remote operation only)
.....psi/sec or.....%/sec

(b) Steady State Incremental Regulation
.....psi min to.....psi max or..... % min to.....% max
(See 3.02.2 for applicability.)

(c) Hysteretic Error
..........max (specify input and output)

(d) Dead Band
..........psi max or..........% max
(specify input and output)

(e) Stability Band
..........psi max or..........% max
(specify input and output)

(f) System Interactions (for compound control system)
Interaction gradient limits for specified independent and dependent variables

(g) Frequency Response with specified steam volume

(i) Open loop crossover frequency
(magnitude ratio output/input = 1, zero db)
ω_c = ... rad/sec

(ii) Closed loop resonance magnitude ratio
$G_R = 20 \times \log_{20} \left[(G(\omega_R) \right]$ = ... db

(iii) Closed loop crossover frequency (magnitude ratio output/input = 1, zero db)

$\omega_{c(CL)} = \ldots\ldots$ rad/sec

(iv) Closed loop – 3db attenuation frequency

(magnitude ratio output/input $= \frac{1}{\sqrt{2}}$, – 3db)

$\omega_a = \ldots\ldots$ rad/sec

(h) Transient Performance

(i) Max transient deviation

..... psi or .....%

For ..... psi or .....% step in flow or pressure set point

(ii) Rise time (not saturated)

(iii) Saturation limit

.....psi or .....% pressure change or .... lb hr or .....% flow change

3.02.6 The following is a list of typical items upon which agreement shall be reached:

(a) Object of test and methods of operation (See Par. 3.02.1.)

(b) The intent of the specifications and guarantees. (See Par. 3.02.4 & 5).

(c) Means for maintaining test conditions and adjusting loads as required.

(d) Frequency of observations, and number of repeats to insure a representative sampling.

(e) Duration and operating range of test runs.

(f) Nature of transients and maximum rate of changes in metal temperature.

(h) Possible effect on the station equipment.

(i) Organization and number of observers, arrangements for their direction, recording of readings and calculating results.

(j) Allocation of responsibilities.

3.02.7 The method of comparing test results to the specified performance shall be agreed upon prior to the test, including allowances for instrument calibration.

3.02.8 If the purpose of the test is for verification of guaranteed performance under a contract, the test should be undertaken as soon as possible after the turbine is first put into operation.

3.02.9 The parties to the test should be accorded the right to run checks on the control system during or before initial operation of the turbine.

3.03 Tolerances

3.03.1 Tolerances for testing inaccuracies are outside the scope of this Code. The test readings shall be reported as observed, and the test results shall be reported as calculated, with such corrections as are provided in this Code.

3.04 Preliminary Test

3.04.1 It is recommended that a preliminary test be run for the purpose of:

(a) Checking the adjustment of the system and the turbine to assure that the equipment is in suitable condition for a test.

(b) Checking all instruments.

(c) Training personnel.

(d) Establishing the test points.

3.04.2 If the parties to the test agree the preliminary test may be accepted as a final test.

3.05 Operating Conditions

3.05.1 Preparatory to a test the turbine and associated equipment shall be operated for a sufficient time to attain steady operating conditions.

3.05.2 Operating conditions (pressure and temperature) shall be maintained within design values during the course of the testing, and great care shall be exercised to maintain steam and plant conditions free from any appreciable variations.

3.06 Instruments and Records

3.06.1 The accuracy and reliability of all instruments shall be as stated in Section 4. Prior to the test the initial calibration of all the instruments shall be available and the method of calibration shall be agreed upon.

3.06.2 Only such observations and measurements need be made as apply and are necessary to attain the object of the test. Each observer shall record his actual observations on the test record and each of the parties to the test shall receive a certified copy of the original test record. Corrections and corrected values shall be entered in the test in such a manner that the original entry remains legible and shall be accompanied by an explanatory note.

3.07 Evaluation of Test Results

3.07.1 If the test results show malfunctioning of the unit or any auxiliaries the defects shall be corrected before the test series is repeated.

3.07.2 If, during the conduct of a test, or during the subsequent analysis or interpretation of observed data an obvious inconsistency is found, the parties to the test should make every reasonable effort to adjust or eliminate the inconsistency by mutual agreement. Failure to reach such an agreement shall require repetition of the test.

3.07.3 If the test shows that the system does not meet the specified or guaranteed performance, resultant allowances, agreements or actions are outside the scope of this Code.

SECTION 4, INSTRUMENTS AND METHODS OF MEASUREMENTS

4.0 Selection of Instruments

4.01 General. This section describes the instruments required to determine the performance characteristics of a pressure control system. Unless otherwise specified herein, the instruments and their calibration shall be in accordance with the Performance Test Code Supplements on Instruments and Apparatus (PTC 19).

4.02 To determine the performance characteristics of a pressure control system, it is necessary to measure one or more of the following:

(a) Pressure.
(b) Time.
(c) Speed or generator frequency.
(d) Motion or travel.
(e) Power output or electrical load.
(f) Steam flow.
(g) Temperature.

4.03 Pressure Measurements

4.03.1 Accuracy. Pressure measurement is particularly critical. The accuracy and dead band must be commensurate with the performance specification of the control system.

4.03.2 Types of Instruments. The use of graphic pressure recording instruments is essential for determining pressure-stability and transient response, and for the simultaneous measurement of pressure and servomotor travel for dead band determination. The use of dead weight gage testers is recommended as the calibrated pressure source.

4.03.3 Pressure measuring instruments shall meet the specifications listed in Table V: in per cent of test pressure.

4.03.4 Pressure Source. For certain tests the application of a calibrated pressure is necessary. This can be accomplished with a dead weight gage tester connected to the pressure sensor to apply and measure pressure simultaneously.

4.04 Time Measurements

4.04.1 Time measurement is required to establish settling time for determining transient response and to determine the rate of change of pressure set point.

4.04.2 In tests where recording instruments are used the chart speed should be chosen so that the required time data can be read with an accuracy of ±2 per cent of the time value to be measured.

4.04.3 A stop watch shall be used when determining the rate of change of pressure set point.

4.05 Speed or Frequency Measurements

4.05.1 Speed or generator frequency measurements may be required on compound speed/pressure control systems particularly on small independent electrical systems.

4.05.2 The measurement of generator frequency is considered equivalent to the measurement of speed.

4.05.3 The use of graphic speed-recording instruments is desirable for determination of stability and transient response.

Table V: Specifications of Pressure Indicating and Recording Instruments

Application	Recommended Range (Per cent of Test Pressure)	Repeatability (Per cent of Test Pressure)	Chart Speed	Full Scale Response
Stability Index	95 to 105%	0.1%	Commensurate	0.1 sec
Time Response	90 to 110%	0.25%	with	0.1 sec
Dead Band	95 to 105%	0.1%	duration	–
Regulation	90 to 110%	0.25%	of test	–

4.05.4 Speed-measuring instruments shall meet the specifications listed in Table VI: in percent of rated speed.

4.06 Motion or Travel Measurements

4.06.1 Measurement of motion will generally be limited to measurement of the distance traveled by the servomotor from a point of reference. Care should be taken to minimize measurement errors caused by temperature and pressure variations.

4.06.2 For the purpose of this Code, the position of a valve set may be measured by servomotor piston stroke. The measuring device shall measure the position of the valve servomotor within ±0.5 per cent of full travel.

4.06.3 For measurement of steady state travel in excess of 2 in., a scale graduated in divisions of 0.02 in or smaller may be used.

4.06.4 For measurement of steady state travel not exceeding 2 in., use of a dial indicator with scale divisions of 0.001 in. is required.

4.06.5 For measurement of stability and transient response, recording equipment must have a repeatability of 0.5 per cent of rated travel (linear variable differential transformer or potentiometer).

4.07 Power-Output Measurements

4.07.1 It is desirable but not mandatory that power-output measurements conform with the requirements of the Test Code for Steam Turbines (PTC 6).

4.07.2 If it is not practicable or convenient to comply with the recommendations of Par. 4.07.1, properly calibrated station instruments may be used.

4.08 Steam Flow Measurements

4.08.1 Steam flow should be measured within an accuracy of ±2 per cent of maximum flow. (See PTC 19.5.)

4.08.2 For an initial pressure control, in the event a primary device is not available for steam flow measurement and the relation between a certain stage pressure and steam flow is known (from design or test), this stage pressure can be used for determining steam flow. The gage used to measure this stage pressure shall be calibrated with a dead weight gage tester.

4.09 Temperature Measurements and Special Instruments

4.09.1 Within the scope of this Code temperature measurements are necessary only to ascertain that the steam conditions are within acceptable limits as specified.

4.09.2 Normally the accuracy of indicating or recording instruments available in the station is sufficient.

4.09.3 For determining dead band and hysteretic error, the use of X-Y recorder (as described in PTC 20.1, Par. 4.43) facilitates measurement and eliminates the need for cross plotting time records of pressure and servomotor position.

4.09.4 The repeatability and dead band of the X-Y recorder shall be on the order of 10 times better than the performance specifications of the pressure control system.

4.09.5 The gains of the X-Y recorder shall be selected such that the produced hysteresis loop is at an angle of 30 to 60 degrees with respect to the two axes.

Table VI: Speed Measuring Instruments

Application	Recommended Range (Per cent of Rated Speed)	Repeatability (Per cent of Rated Speed)	Chart Speed	Full Scale Response
For Pressure-Stability Index Test	95 to 105%	0.02%	Commensurate	0.1 sec
For Time Response Test	95 to 105%	0.02%	with	0.1 sec
For Speed Regulation Test	85 to 115%	0.25%	duration of test	—

4.10 Testing (General)

4.10.1 As discussed in Section 3, the selection of tests to be performed on the unit should be based on its intended application. For most applications it should be necessary to perform only a few of the following tests.

4.10.2 Order of Tests

The tests are listed in a logical order so that it is feasible to proceed sequentially through the required tests. However, except where the results of one test influence the procedure in a subsequent test, any sequence of testing agreed upon is satisfactory.

4.10.3 Combined Testing

Certain test conditions and procedures are so related that it is more efficient, if both tests are to be performed, to conduct them virtually concurrently. These groupings are:

(a) 4.13, Steady State Pressure Regulation.
4.14, Incremental Steady State Pressure Regulation – Method I.
4.15, Hysteretic Error.
4.16, Dead Band.
(b) 4.17, Control Velocity Saturation.
4.18, Step Response – Control.
(c) 4.19, Step Response – System.
4.20, Stability Band.
4.21, System Interaction.
(d) 4.22, Frequency response control.*
4.23, Frequency response system.*

*Tests that can be performed at any time in test sequence.

4.10.4 Recording of Results

For each test the quantities to be recorded are itemized. When reference is made to operating point it is meant that the following items should be recorded:

(a) General configuration, including whether the generator is operated isolated or synchronized, steam process connections, shut-off valve positions, etc.
(b) Turbine speed.
(c) Electrical load.
(d) Steam pressures (initial or process).
(e) Process flow(s).
(f) Control settings.
(g) Special conditions or deviations.
(h) Valve position(s).
(i) Conditions of hydraulic, pneumatic and electrical power supplies.

4.10.5 Recorder Adjustments

In all cases in which a recorder is used, preliminary tests shall be made to ascertain that the channel gains are commensurate with the magnitude of the variable fluctuations recorded.

Procedures

4.11 Pressure Set Point Range

4.11.1 Equipment required:

(a) Dial indicator.
(b) Adjustable calibrated pressure source.
(c) Pressure indicator.

4.11.2 Test Conditions:

(a) Unit at standstill.
(b) Pressure control system in normal operation with pressure source replacing process pressure.

4.11.3 Procedure:

(a) Adjust pressure set point to maximum position.
(b) Adjust pressure source so that flow controlling valves are in a suitable position, and not against a limit.
(c) Run set point to tis minimum position, and adjust pressure source so that the flow controlling valves are again in the same position within ±0.5 per cent of rated travel.

4.11.4 Record for Each Test:

(a) Valve or servomotor position for cases in Par. 4.11.3 (b) and (c).
(b) Source pressure for cases in Par. 4.11.3 (b) and (c).

4.11.5 Calculations Appear in Par. 5.11.

4.12 Rate of Change of Pressure Set Point

4.12.1 It is intended that this test be run only on systems which include a remotely operated set point adjustment having a specified range. It should be ascertained that the change in an observed parameter, such as adjusting screw position, actually represents a change in set point, over its entire range. If the pressure applied to the sensing element influences the rate of change, rated pressure shall be applied to the sensing element.

4.12.2 Equipment Required:

(a) Stop watch.
(b) Means for determining limits of range.

4.12.3 Test Conditions:

(a) Unit at standstill. (Unit may be in operation provided this procedure can be tolerated by the system.)
(b) Pressure set point device in normal operation.

4.12.4 Procedure:

(a) With a stop watch measure the time it takes to run the pressure set point device from one extreme to the other in each direction.

4.12.5 Record for Each Test:

(a) Time interval.

4.12.6 Calculations are given in Par. 5.12.

4.13 Steady State Pressure Regulation

4.13.1 If Test 4.14, steady state incremental pressure regulation, is to be performed, this procedure can be omitted in its entirety and the data from that test can be used.

4.13.2 Equipment Required:

(a) Adjustable calibrated pressure source.
(b) Means to determine valve or servomotor position.

4.13.3 Test Conditions:

(a) Unit at standstill.
(b) Pressure control system in normal operation with pressure source connected in place of process pressure.

4.13.4 Procedure:

(a) Apply rated pressure by means of the pressure source.
(b) Adjust pressure set point so that the pressure controlling valves are in the minimum flow position.
(c) Vary the pressure source so that the valves move to the maximum flow position.

4.13.5 Record for Each Test:

(a) Pressure applied at maximum flow position.
(b) Pressure applied at minimum flow position.
(c) Actual valve or servomotor positions for each test.

4.13.6 Calculations are given in Par. 5.13.

4.14 Steady State Incremental Pressure Regulation – Method I

4.14.1 Refer to Par. 3.02.2 for a discussion of the types of units for which this test is applicable. Method I utilizes flow-lift curves supplied for the equipment for the flow controlling valves, and does not require testing with the unit in operation. The maximum range of valve positions over which the test shall be conducted shall correspond to 5 and 95 per cent process flow. A smaller range may be specified or agreed upon. If Test 4.15, Hysteretic Error, is to be performed, this test procedure may be omitted in its entirety.

4.14.2 Equipment Required:

(a) X-Y recorder (preferred) or strip chart recorder.
(b) Pressure transducer.
(c) Calibrated valve position transducer.
(d) Dial indicator for valve position.
(e) Adjustable calibrated pressure source.

4.14.3 Test Conditions:

(a) Unit at standstill.
(b) Pressure control system in normal operation with pressure source connected in place of process pressure.
(c) X-Y recorder connected so as to record valve position versus source pressure.

4.14.4 Procedure:

(a) Apply rated pressure by means of the pressure source.
(b) Adjust pressure set point so that the pressure controlling valves are in the 5 per cent flow position.
(c) Vary the pressure source in small steps (2 to 5 per cent of steady state regulation) so as to obtain, from 5 to 95 per cent flow and back to 5 per cent flow, a record of valve position versus source pressure.

4.14.5 Record for Each Test:

(a) Control settings.
(b) X-Y or strip chart record of valve position versus source pressure.

4.14.6 Calculations are given in Par. 5.14.

4.15 Hysteretic Error

4.15.1 This test procedure is quite similar to the preceding test except that exceptional care must be taken to prevent path reversals which would reduce the amount of recorded hysteretic error.

4.15.2 Equipment Required. See Par. 4.14.2.
4.15.3 Test Conditions. See Par. 4.14.3.

4.15.4 Procedure:

(a) From flow-lift curves determine the flow controlling valve positions corresponding to 5 and 95 per cent flow.
(b) Vary the source pressure in small steps (2 to 5 per cent of steady state regulation) so that the flow controlling valve operates from the 5 per cent flow position to the 95 per cent flow position and back again. Do not reverse the direction of pressure movement except at the ends of test range, and do not cause the servomotor to overshoot by applying too large or too sudden pressure change.

4.15.5 Record:

(a) Valve positions calculated for 5 and 95 per cent flow.
(b) Control settings.
(c) X-Y or strip chart records of valve position versus source pressure.

4.15.6 Calculations are given in Par. 5.15.

4.16 Dead Band

4.16.1 Process flow values at which tests are to be conducted shall be specified. If not otherwise specified, use minimum, maximum, and a mid-flow value.

4.16.2 Equipment Required:

(a) X-Y recorder (preferred) or strip chart recorder.
(b) Pressure transducer.
(c) Calibrated valve position transducer or dial indicator for valve position.
(d) Adjustable calibrated pressure source.

4.16.3 Test Conditions:

(a) Unit at standstill.
(b) Pressure control system in normal operation with pressure source connected in place of process pressure.
(c) Recorder connected so as to measure valve position versus source pressure. (For this test, recorder gains must be considerably higher than required for Tests 4.14 and 4.15.)

4.16.4 Procedure:

(a) Using the set point adjustment and/or the pressure source, set the pressure controlling valve to a desires operating point.
(b) Make very small changes in the pressure source until a change in valve position is noted. Then make small changes in the opposite direction until the valve position changes to the opposite direction.

4.16.5 Record for Each Test Point:

(a) Valve or servomotor position.
(b) Control settings.
(c) X-Y or strip chart records of valve position versus source pressure.

4.16.6 Calculations are given in Par. 5.16.

4.17 Control Velocity Saturation

4.17.1 This test determines the magnitude of transients at which saturation is observed in the control system. Usually the servomotor velocity will reach a limiting value. The step may be applied either to the set point or by means of a pressure source.

4.17.2 Equipment Required:

(a) Strip chart recorder.
(b) A pressure source capable of producing step changes, or a means for producing quasi-step set point changes.
(c) Means for recording position and/or velocity.
(d) Means for recording input step.

4.17.3 Test Conditions:

(a) Unit at standstill.
(b) Control system in normal operation, with pressure source connected in place of process pressure.
(c) Strip chart recorder set to obtain traces of valve or servomotor position and input step of suitable magnitude.

Note: Since the system's transient characteristics, are affected by supply characteristics, consideration should be given to obtaining a hydraulic oil supply flow capability substantially equal to that when the unit is at rated speed.

4.17.4 Procedure:

(a) Set pressure control system for approximately mid-range operation.
(b) Utilizing either the set point device or the pressure source apply a step change of approximately 5 per cent magnitude.
(c) Apply a step double the magnitude used in (b).
(d) Using the calculation method described in Par. 5.17, compare the maximum output velocity of steps (b) and (c).
(e) As described in Par. 5.17, depending on the results of the calculations either increase or decrease step size and repeat this procedure until control saturation has been established.

Note: Control velocity saturation may be detected by observing the shape of the strip chart record. (See Fig. 5-5).

4.17.5 Record for Each Step:

(a) The initial and final operating point.
(b) Control settings.
(c) Maximum control velocities for each observed response.
(d) Strip chart records of response.

4.17.6 Calculations are given in Par. 5.17

Note: If steam forces on the valves are suspected to substantially influence the hysteretic error, dead band or velocity saturation, ways must be found to test with the unit in operation. The test procedure must then be specified by the parties to the test.

4.18 Step Response – Control

4.18.1 This test is identical to the preceding test, 4.17, except that, unless otherwise specified, step size approximately midway between the dead band and velocity saturation is used. Data obtained during the previous test may be used.

4.18.2 Equipment Required. See Par. 4.17.2
4.18.3 Test Conditions. See Par. 4.17.3

4.18.4 Procedure:

(a) Determine an input step size approximately midway between the magnitude corresponding to the control dead band and the magnitude corresponding to velocity saturation.
(b) Adjust the pressure control system for approximately mid-range operation.
(c) Start recorder sufficiently in advance of test so as to obtain an adequate record of steady state conditions.
(d) Apply an input step of the calculated magnitude as rapidly as possible. The time taken to produce the step should be 1/10 or less of the system response time.
(e) After steady state conditions are re-attained, operate the recorder long enough to establish the new steady state condition.
(f) Obtain similar records for a step in the opposite direction.

4.18.5 Record for Each Transient:

(a) Initial steady state operating point.
(b) Final steady state operating point.
(c) Strip char record of input and output (usually servomotor position).
(d) Control setting.

4.18.6 Calculations are given in Par. 5.18.

4.19 Step Response – System

4.19.1 This test is run to determine if the flow capability and control settings of the system are satisfactory. The step applied may either be a set point change or flow demand change.

4.19.2 Equipment Required:

(a) Strip chart recorder.
(b) Process pressure transducer.
(c) Flow transducer or set point recording means.

4.19.3 Test Conditions:

(a) Unit in normal operation.
(b) The particular flow and steam volume conditions as specified.
(c) If adjustments are available to the operator in normal operation, these may be adjusted to optimize the response for the test conditions; but they must remain fixed for the duration of the test.

4.19.4 Procedure:

(a) Start recorder sufficiently in advance of test so as to obtain an adequate record of steady state conditions.
(b) Rapidly apply step as determined in

Par. 4.18.4. If the time to produce the step is more than 1/10 of the resulting system response time, the test must be repeated with a correspondingly shorter step time.

(c) After steady state conditions are obtained, continue to operate recorder to establish the new steady state conditions.

(d) If feasible, apply step of the same size in the opposite direction.

4.19.5 Record for Each Applied Step Change:

(a) Initial operating point.
(b) Final operating point.
(c) Strip chart records of process pressure, process flow or set point.

4.19.6 Calculations are given in Par. 5.19.

4.20 Stability Band

4.20.1 This test is run to determine the magnitude of steady state fluctuations during normal operation. The electrical load and the process conditions at which the tests are to be run should be specified. If not specified, tests shall be conducted at approximately 5, 50 and 95 per cent rated steam flow. Every effort should be made to minimize fluctuations due to external disturbances. This test may be run concurrently with Test 4.19, step response, if recorder gains are properly adjusted to obtain the required data. The length of the strip chart record should be sufficient to establish the maximum and minimum pressure.

4.20.2 Equipment Required:

(a) Strip chart recorder(s).
(b) Process pressure transducer.
(c) Other system variable transducers as may be desired by test engineer.

4.20.3 Test Conditions:

(a) Unit in normal operation.
(b) Operating point of unit as specified or agreed upon.
(c) Recorder connected so as to measure process pressure and other significant variables.

4.20.4 Procedure:

(a) At specified operating point obtain strip chart record of process pressure and other system variables.
(b) If possible deactivate a portion of the control loop for the variable under test maintaining approximate operating conditions in order to obtain a measure of the fluctuations that are not attributable to the control system under test.

4.20.5 Record:

(a) System operating point uner Par. 4.19.4 (a) and (b)
(b) Recorder gain calibrations.

4.20.6 Calculations are given in Par. 5.20.

4.21 System Interactions

4.21.1 This test is to determine how system demand changes affect other controlled variables. Either static or dynamic tests can be made.

Typical interaction measurements are: (a) when supplying power to an isolated system, the frequency change due to a change in process flow demand; (b) when synchronized with a larger power system, the load change due to a change in process flow demand; (c) the process pressure change due to a change in electrical load; (d) for multi-extraction machines, the process pressure change due to process flow demand changes at other locations; (e) for cases in which process pressure is fixed by other controls, the flow change due to change in electrical load or other extraction pressure.

4.21.2 Equipment Required:

Note: If static tests only are to be made, station indicating instrument data are usually adequate.

(a) Strip chart recorder(s).
(b) Pressure transducers.
(c) Steam flow transducers.
(d) Electrical load transducer.
(e) Other transducers as desired by test engineer. Valve position and controls signals may be of particular interest.

4.21.3 Test Conditions:

(a) Unit in normal operation.
(b) If manual adjustments are readily available to the operator, they may be made to suit the operating point, but the settings must remain fixed for the duration of the tests except as required by the test procedure.

4.21.4 Procedure:

(a) Obtain steady state operating conditions.
(b) Start recorder sufficiently in advance of test so as to obtain a reasonable record of steady state conditions.
(c) Apply specified disturbance. After conditions have stabilized, continue to operate recorder until a record of the new steady state conditions has been obtained.
(d) If feasible repeat steps (a) through (c) using a step in the opposite direction.
(e) Repeat steps (a) through (d) for other agreed upon steps and operating points.
(f) During these tests care should be taken to avoid control system saturation unless knowledge of the effect of saturation is specifically desired.

4.21.5 Record:

(a) All initial operating points.
(b) Value of disturbance applied.
(c) All final operating points.
(d) Control settings used for each test.
(e) Strip chart records of all transients.
(f) Recorder gain calibrations.

4.21.6 Calculations are given in Par. 5.21.

4.22 Frequency Response – Controls

4.22.1 This test will determine if the control system dynamic performance meets specifications. If the specification is concerned with system pressure performance, Test 4.23, Frequency Response – System, shall be used.

4.22.2 Equipment Required:

(a) Strip chart recorder with appropriate gains such that at very low frequencies the input and output oscillations have the same magnitude (applicable only to systems with no integral action), or servo-analyzer which measures the phase angle and the return amplitude directly.
(b) Valve or servomotor position transducer.
(c) Pressure transducer for input pressure.
(d) Device for generating a sinusoidal pressure signal.

4.22.3 Test Conditions:

(a) Unit at standstill.
(b) Control system in normal operation with sinusoidal pressure signal in place of process pressure.
(c) Sufficient power fluid supply flow capability. (See Par. 4.17.3).

4.22.4 Procedure:

(a) Adjust the control system for approximately mid-range operation.
(b) Apply the sinusoidal pressure signal. The input amplitude should be as large as possible without saturating any part of the system. This can be checked by observing the servomotor position trace which should be approximately sinusoidal. If practical, adjust sinusoidal pressure signal amplitude at each frequency so that the input to the portion of the system under test has approximately a constant amplitude over the range of test frequencies.
(c) Vary the test frequencies so as to cover a frequency range of at least 3 decades centered about the region of interest. This will include the significant break frequencies. The attenuation is measured from the asymptote of the response curve to the left of the critical area. If the controller has reset, a low frequency gain contributed by this source should be discounted in determining the –3 decibel point.
(d) If a strip chart recorder is used, observe the amplitude of the recorder tracings and adjust the recorder gains at each frequency so that a trace of sufficient amplitude is obtained; keep a record of the gains for each frequency.
(e) If a servo-analyzer is used, obtain amplitude and phase lag readings at the pressure input location and at the system output location. Phase and amplitude readings at other points may be taken also, if desired.

4.22.5 Record at Each Frequency:

(a) Average operating point.
(b) Oscillation amplitudes of:
1. Oscillator output.
2. Pressure signal input.
3. System output (servomotor position).
(c) Other variables may be recorded as desired.
(d) If servo-analyzer is used, record system input phase lag and system output phase lag.

(e) Strip chart records of several cycles of operation at each frequency.

4.22.6 Calculations are given in Par. 5.22.

4.23 Frequency Response – System

4.23.1 In addition to running a frequency response test on the control system, it is also possible to determine the frequency response of the entire system during normal operations. This requires a means for sinusoidally oscillating the set point of the pressure control system, or of applying a sinusoidal process flow demand on the system.

4.23.2 Equipment Required:

(a) Strip chart recorder and (if available) servo-analyzer.
(b) Valve position transducer.
(c) Pressure transducer.
(d) Device for applying a sinusoidal pressure set point, or
(e) Device for generating a sinusoidal flow demand.

4.23.3 Test Conditions:

(a) Unit operating near a specified operating point.

4.23.4 Procedure: Follow Par. 4.22.4 except vary *pressure set point.*

4.23.5 Record at Each Frequency:

(a) System operating point.
(b) Oscillation amplitudes of
  1. Oscillator output.
  2. Signal input.
  3. System output pressure.
(c) Other variables may be recorded as desired.
(d) If servo-analyzer is used, also record system input phase lag and system output phase lag.

(e) Strip chart records of several cycles of operation at each frequency.

4.23.6 Calculations are given in Par. 5.23.

4.24 Incremental Pressure Regulation – Method II. (See Par. 3.02.2 for applicability.)

4.24.1 When flow-lift curves are not available and it is desired to determine incremental regulation, this test method can be used. Provisions shall be made to adjust the process flow demand in small increments over the test range.

4.24.2 Equipment Required:

(a) Steam flow transducer.
(b) Process pressure transducer.
(c) X-Y recorder (preferred) or strip chart recorder.

4.24.3 Test Conditions:

(a) Unit in normal operation.

4.24.4 Procedure:

(a) Operate unit at approximately 5 per cent flow conditions with set point adjusted to obtain rated process pressure.
(b) Increase process flow demand in small steps so as to obtain a record of process pressure between 5 and 95 per cent flow.
(c) Reverse direction of steps so as to obtain a similar record in the opposite direction.

4.24.5 Record:

(a) Data at minimum test flow.
(b) Data at maximum test flow.
(c) Recorder gain calibrations.
(d) X-Y or strip chart records.

4.24.6 Calculations are given in Par. 5.24.

SECTION 5, EVALUATION AND COMPUTATION OF RESULTS

5.01 The tests performed as described in Section 4 are evaluated as described in the following paragraphs.

5.02 Paragraph numbers for the computations of a particular test correspond with the numbers of the description of that test in Section 4.

5.11 Pressure Set Point Range

5.11.1 (a) The source pressure at the maximum pressure set point is

$$P_{S_{max}}$$

(b) The source pressure at the minimum pressure set point is

$$P_{S_{min}}$$

5.11.2 The range of pressure set point adjustment (ΔP_S) is

5.11.2–1 $\Delta P_S = P_{S_{max}} - P_{S_{min}}$ [psi]

5.11.3 The average range of the pressure set point range is:

5.11.3–1 $$P_{S\,(average)} = \frac{\Delta P_{S_1} + \Delta P_{S_2} + \ldots \Delta P_{S_n}}{n}$$

for n tests

5.12 Rate of Change of Pressure Set Point

5.12.1 The time measured to run the remotely operated set point device from the minimum to the maximum set point is ΔT_S in the increasing and decreasing direction differ by more than 10 per cent, both values shall be recorded.

5.12.2 The rate of change of pressure set point, $\frac{dP_S}{dt}$ is

5.12.2–1 $$\frac{dP_S}{dt} = \frac{\Delta P_S}{\Delta T_S} \text{ [psi/sec]}$$

5.12.3 The relative rate of change of pressure set point $\frac{d\psi_S}{dt}$ in per cent is

$$\frac{d\psi_S}{dt} = \frac{\Delta P_S}{\Delta T_S} \times \frac{100}{P_r} \text{ [per cent/sec]} \qquad (2.03.36–1)$$

5.13 Steady State Pressure Regulation

5.13.1 (a) The source pressure at the minimum flow position of the valves is P_1

(b) The source pressure at the maximum flow position of the valves is P_2

5.13.2 The steady state pressure regulation R_P is

5.13.2–1 $R_P = |P_2 - P_1|$ [psi]

5.13.3 The relative steady state pressure regulation δ_P in per cent is

$$\delta_P = R_P \times \frac{100}{P_r} \text{ [per cent]} \qquad (2.03.26–1)$$

Note: The absolute value of R_p or δ_p is stated as the steady state or relative steady state pressure regulation, respectively. If it is not obvious which way the pressure is changing with flow, this shall be stated with the regulation.

5.13.4 Average Steady State Gain of Pressure Controller

The average steady state gain (G_P) of a pressure controller is the inverse value of the relative steady state pressure regulation expressed in units

5.13.4–1 $$G_P = \frac{100}{\delta_P} \left[\frac{\text{per unit of flow change}}{\text{per unit of pressure change}} \right]$$

5.14 Steady State Incremental Pressure Regulation – Method I

5.14.1 The record of servomotor stroke versus source pressure obtained in Test 4.14 is usually a fairly linear characteristic. A typical example for an *initial pressure control* system is shown in Fig. 5-1.

5.14.2 A typical flow versus servomotor (valve) position curve used to evaluate the incremental pressure regulation for a multivalve unit is shown in Fig. 5-2.

5.14.3 Select a number of points of servomotor positions and read the corresponding pressure from Fig. 5-1 and the steam flow from Fig. 5-2.

5.14.4 The number of points should be sufficient to plot the steam flow versus pressure as typically shown in Fig. 5-3.

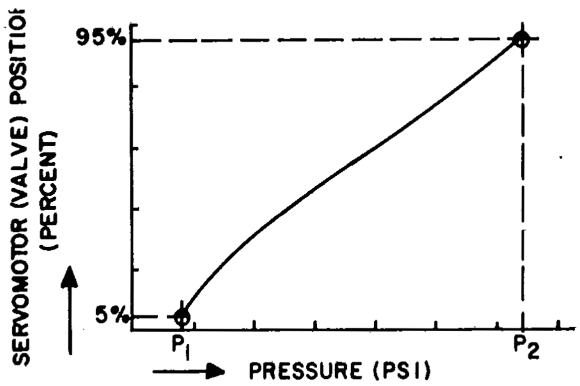

FIG. 5-1 TYPICAL SERVOMOTOR (VALVE) POSITION VERSUS STEAM PRESSURE RECORD

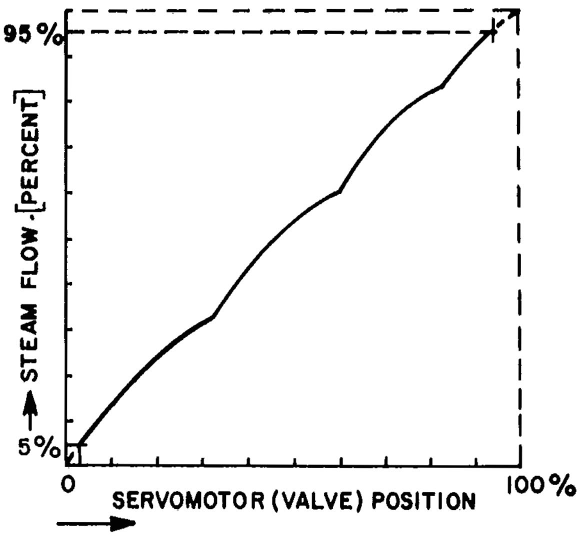

FIG. 5-2 TYPICAL STEAM FLOW VERSUS SERVOMOTOR (VALVE) POSITION DATA

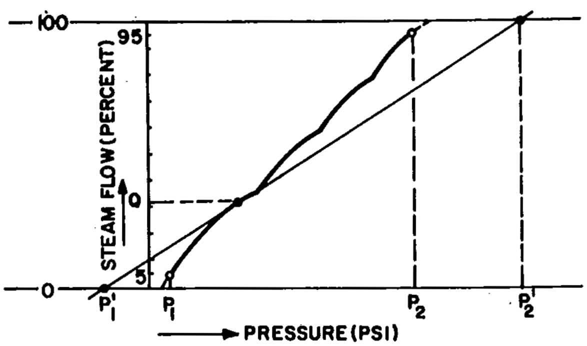

FIG. 5-3 TYPICAL PLOT OF STEAM FLOW VERSUS PRESSURE FOUND FROM FIG. 5-1 AND 5-2

5.14.5 At any point where the incremental pressure regulation is to be evaluated, draw the tangent on the curve in Fig. 5-3 extending it so that it crosses the zero and 100 per cent flow line, establishing the pressures P_1' and P_2', respectively, as shown in Fig. 5-3.

5.14.6 The incremental pressure regulation at the flow Q is $R_i\ (Q)$ and is:

5.14.6-1 $R_i\ (Q) = P_2' - P_1'$ [psi]

5.14.7 The relative incremental pressure regulation at the flow Q is $\delta_i\ (Q)$:

$$\delta_i\ (Q) = R_i\ (Q) \times \frac{100}{P_r}\ \text{[per cent]} \qquad (2.03.28\text{-}1)$$

Note: The absolute value of R_i δ_i is stated for the incremental or relative incremental regulation, respectively. If it is not obvious which way the pressure is changing with flow, this shall be stated with the regulation.

5.15 Hysteretic Error

5.15.1 The test record taken under Par. 4.15 results in a "hysteresis loop," as typically shown in Fig. 5-4.

5.15.2 The hysteretic error ΔP_h is the largest distance (in psi) between the "up" and the "down" trace measured parallel to the pressure axis, establishing the pressures P_{h_1} and P_{h_2}:

5.15.2-1 $\Delta P_h = P_{h_2} - P_{h_1}$ [psi]

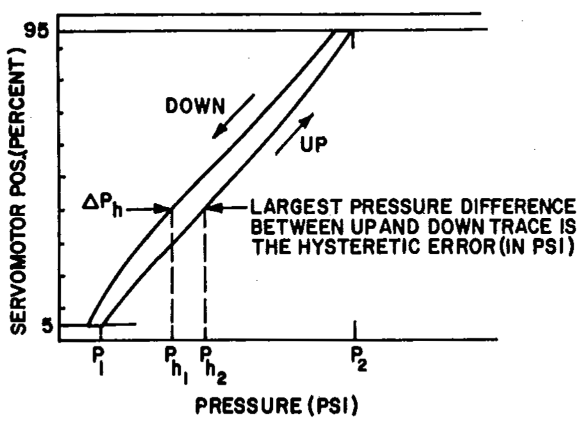

FIG. 5-4 TYPICAL DIAGRAM FOR DETERMINING THE HYSTERETIC ERROR

5.15.3 The relative hysteretic error ψ_h in per cent is

$$\psi_h = \Delta P_h \times \frac{100}{P_r} \quad \text{[per cent]} \qquad (2.03.16\text{-}1)$$

5.16 Dead Band

5.16.1 The pressures measured in the test in Par. 4.16, at which the servomotor started to move in each direction, shall be called P_{d_1} and P_{d_2}, respectively.

5.16.2 The dead band ΔP_d is

5.16.2-1 $\Delta P_d = P_{d_2} - P_{d_1}$ [psi]

5.16.3 The relative dead band ψ_d in per cent is:

$$\psi_d = \Delta P_d \times \frac{100}{P_r} \quad \text{[per cent]} \qquad (2.03.4\text{-}1)$$

5.17 Control Velocity Saturation

5.17.1 Servomotor position traces, as obtained by by the test in Par. 4.17, are typically shown in Fig. 5-5 (for a critically damped system).

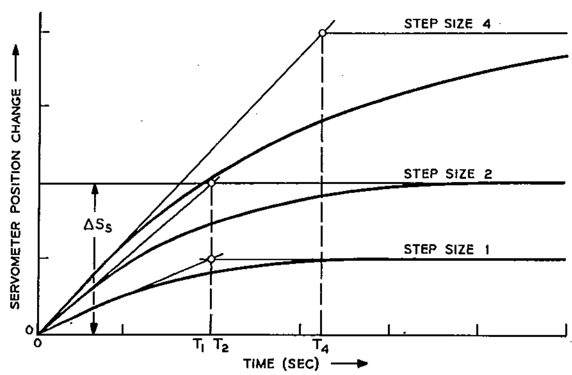

FIG. 5.5 TYPICAL SERVOMOTOR POSITION VERSUS TIME TRACES FOR SATURATION VELOCITY

5.17.2 Draw the tangent on the servomotor position versus time trace at the time $t = 0$.

As long as this tangent intersects the horizontal line of the new steady state position at the same time (T_1 and T_2) the servomotor has not attained saturation velocity.

At the step size this time starts to increase such as shown for T_4, the saturation velocity is reached.

5.17.3 If a given step size definitely shows saturation, decrease the step size toward the next lower step that was not above saturation to determine the step size at which saturation starts.

5.17.4 The saturation velocity (V_{sat}) can now be calculated by measuring the stroke ΔS_{sat} (see Fig. 5-5) and the time T_2 (assuming that step size 2 was just saturating).

5.17.4-1 $$V_{sat} = \frac{\Delta S_{sat}}{T_2} \quad \text{[in./sec]}$$

5.18 Step Response – Control

5.18.1 A typical output (for example servomotor stroke) versus time trace, as obtained from the test in Par. 4.18, is shown in Fig. 5-6 (less than critically damped system.)

5.18.2 Determine performance data as defined in ASA C85.1, see Fig. 5-6.

5.19 Step Response-System (Closed Loop)

5.19.1 Follow same procedure as described under 5.18.

5.20 Stability Band

5.20.1 Determine the maximum pressure P_{max} and the minimum pressure P_{min} [psi] that occur within a test record of sufficient length.

5.20.2 Calculate the stability band ΔP_b as

5.20.2-1 $\Delta P_b = P_{max} - P_{min}$ [psi]

5.20.3 Calculate the relative stability band or stability index ψ_b in per cent as

$$\psi_b = \Delta P_b \times \frac{100}{P_r} \quad \text{[per cent]} \qquad (2.03.39\text{-}1)$$

5.21 System Interaction (for Automatic Extraction Units)

5.21.1 Determine for each operating point tested the interaction gradient I as the ratio between the dependent and the independent variables.

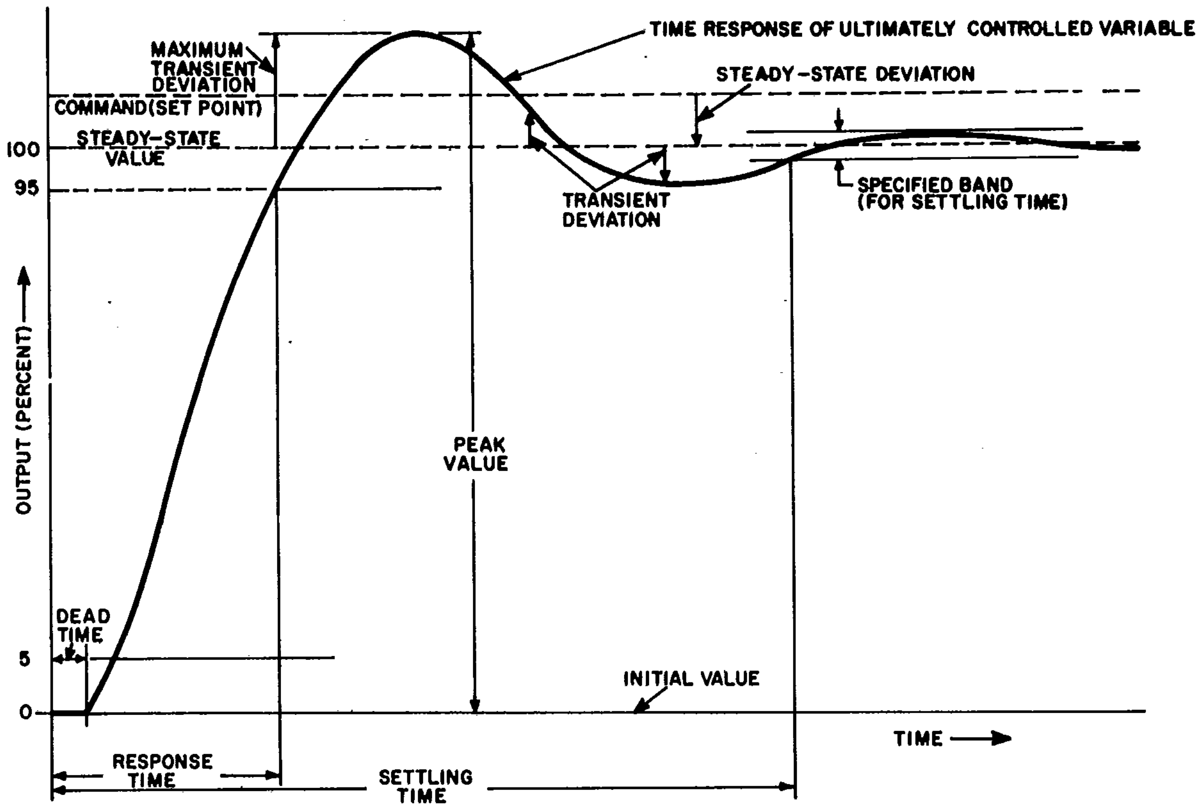

FIG. 5-6 STEP RESPONSE RECORD (TYPICAL)

5.21.2 If the imposed flow change is called ΔQ and the resulting power output change is called ΔO, the interaction gradient (I) is

5.21.2-1 $$I = \frac{\Delta O}{\Delta Q} \quad \left[\frac{KW}{lb/hr}\right]$$

Note: The interaction gradient can have either a plus or a minus sign.

5.22 Frequency Response – Control (Open Loop)

5.22.1 If in the test described in Par. 4.22 a strip chart recorder was used, the following data shall be obtained for a number of frequencies (ω) over a range sufficient to draw a Bode plot:

5.22.2 For each sinusoidal pressure signal of the frequency ω [rad/sec] and the magnitude $\Delta P(\omega)$ [psi] the relative magnitude $M_p(\omega)$ per unit is:

5.22.2-1 $$M_P(\omega) = \frac{\Delta P(\omega)}{P_r} \quad \text{[per unit]}$$

5.22.3 For these same imposed frequencies (ω) compute the relative magnitude of the servomotor oscillation $M_Y(\omega)$

5.22.3-1 $$M_Y(\omega) = \frac{Y(\omega)}{Y_F} \quad \text{[per unit]}$$

where: $Y(\omega)$ is the magnitude of the servomotor oscillation [inches] at the frequency ω and Y_F is the servomotor stroke between zero and full-flow position [inches]

5.22.4 Determine the magnitude ratio $G(\omega)$ of the open loop at the frequency ω in decibels (db):

5.22.4-1 $$G(\omega) = 20 \times \log_{10} \frac{M_Y(\omega)}{M_P(\omega)} \quad \text{[db]}$$

5.22.5 Plot the magnitude ratio $G(\omega)$ over a logarithmic scale for ω in order to obtain a Bode plot of the control system, as shown in Fig. 5-7 for a control with a 10 per cent steady state regulation.

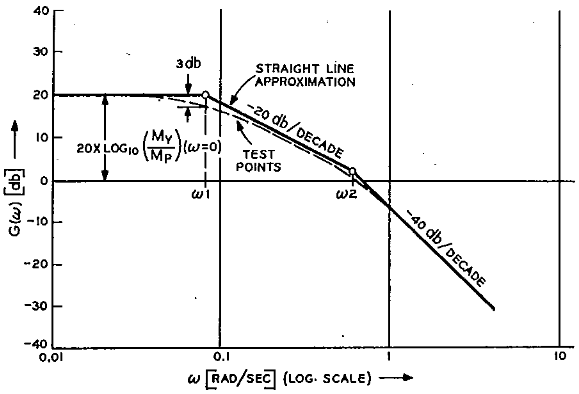

FIG. 5-7 BODE PLOT OF OPEN LOOP CONTROL

5.22.6 If the transfer function $H(j\omega)$ of the plant is available, the open loop transfer function $F(j\omega)$ of the entire system can be plotted.

$$5.22.6\text{-}1 \quad M(F(j\omega)) = 20 \times \left(\log_{10}\left(\frac{M_Y(\omega)}{M_P(\omega)}\right) + \log_{10}\left(H(\omega)\right)\right)$$

5.22.7 For stability discussion see Appendix section 7.03.4.

5.23 Frequency Response–System (Closed Loop)

5.23.1 If in the test in Par. 4.23 a strip chart recorder was used the following values shall be calculated for a sufficient number of frequencies (ω) to draw a closed loop Bode diagram:

5.23.2 For the frequency $\omega = 0$, determine the magnitude of the pressure set point change (M_{set}) by recording the steady state values of the controlled pressure ($P_{max}(0)$, $P_{min}(0)$ in psi), corresponding to the maximum and minimum of the applied set point oscillation.

$$5.23.2\text{-}1 \quad M_{set} = P_{max}(0) - P_{min}(0) \quad [\text{psi}]$$

5.23.3 For each given frequency ω determine the magnitude $M_P(\omega)$ of the change of the controlled pressure by reading the maximum pressure $P_{max}(\omega)$ and the minimum pressure $P_{min}(\omega)$ of the recorded pressure trace:

$$5.23.3\text{-}1 \quad M_P(\omega) = P_{max}(\omega) - P_{min}(\omega) \quad [\text{psi}]$$

5.23.4 Calculate the magnitude ratio $G(\omega)$ of the *closed loop* in decibels:

$$5.23.4\text{-}1 \quad G(\omega) = 20 \times \log_{10} \frac{M_P(\omega)}{M_{set}} \quad [\text{db}]$$

5.23.5 Plot the values of $G(\omega)$ in decibels over a logarithmic scale of the frequency (ω), as shown in Fig. 5-8.

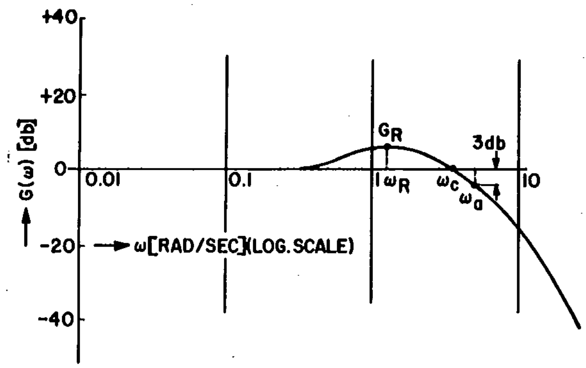

FIG. 5-8 TYPICAL CLOSED LOOP BODE PLOT OF PRESSURE CONTROL SYSTEM

5.23.6 Values of interest that can be specified for this control system are G_R, ω_c and ω_a (3.02.5).

5.23.7 If flow oscillations were used for the test in 4.23, the following values shall be calculated in order to draw the closed loop Bode diagram:

5.23.8 Determine the magnitude of the flow oscillation (M_F) at the frequency $\omega = 0$ referred to rated flow as:

$$5.23.8\text{-}1 \quad M_F = \frac{F_{max}(0) - F_{min}(0)}{F_r} \quad [\text{per unit}]$$

where: $F_{max}(0)$ = maximum test flow (lb/hr) at $\omega = 0$

$F_{min}(0)$ = minimum test flow (lb/hr) at $\omega = 0$

F_r = rated flow (lb/hr)

5.23.9 For each imposed frequency ω determine the magnitude of the controlled pressure oscillation $M_P(\omega)$ referred to the steady state pressure regulation as

5.23.9-1 $$M_P(\omega) = \frac{P_{max}(\omega) - P_{min}(\omega)}{R_P} \text{ [per unit]}$$

5.23.10 Calculate the magnitude ration $G(\omega)$ of the *closed loop* in decibels:

5.23.10-1 $$G(\omega) = 20 \times \log_{10}\left(\frac{M_P(\omega)}{M_F}\right) \text{ [db]}$$

5.23-11 Plot the values of $G(\omega)$ in decibels over a logarithmic scale of the frequency (ω) similar to Fig. 5-8.

5.23.12 If a servo-analyzer was used for the test the values of $G(\omega)$ can be read directly from the instrument and plotted similarly to Fig. 5-8.

5.24 Incremental Pressure Regulation – Method II (for Initial or Back Pressure Controls Only)

5.24.1 Use the evaluation procedure of Par. 5.14, plotting the steam flow versus pressure data (Fig. 5-3) as obtained from test.

SECTION 6, REPORT OF TESTS

6.01 The "Report of Tests" shall include test and calculated information, sufficient to demonstrate whether or not the performance of the pressure control system is within specified limits as agreed between the parties to the test.

6.02 General Information:

(a) Owner.
(b) Owner's designation of unit.
(c) Name and location of plant.
(d) Manufacturer of unit.
(e) Serial number of unit.
(f) Name plate data: Rated speed, pressure, steam conditions, flow(s) electrical load, type of unit.
(g) Date of first commercial operation.
(h) Report number, date of tests and date of report.
(i) Personnel engaged in or observing the test, their functions and their affiliation.
(j) Name of person in charge of test.

6.03 Summary: Brief outline of test results and calculated values, including sifnificant history of pressure control system operation.

6.04 Record of Agreements: All agreements affecting the test results must be included in the test report.

6.05 Description and results of tests.

6.05.1 Detailed description of test.
6.05.2 Data taken according to Section 4, including controlled pressure and steam flow.
6.05.3 Calculated results.

6.06 Appendix of Test Report: This part of the report is to include copies of the original data sheets and records of calibration of instruments, detailed calculations not in body of report and curves showing source data and test results.

SECTION 7, APPENDIX
ANALYSIS OF PRESSURE CONTROL SYSTEMS
(FOR NOMENCLATURE SEE PAR. 7.11)

Note: All values given in per cent in the main body of this Code are being used in per unit in this Appendix.

7.01 Basic Task of Pressure-Control Systems. The applications of pressure controls as listed in Par. 0.01.2 are always one of the two following basic control systems or a combination of both:

7.01.1 Initial Pressure Control controlling the pressure upstream of the control valves within a specified range between a certain minimum flow and maximum flow, Fig. 7-1.

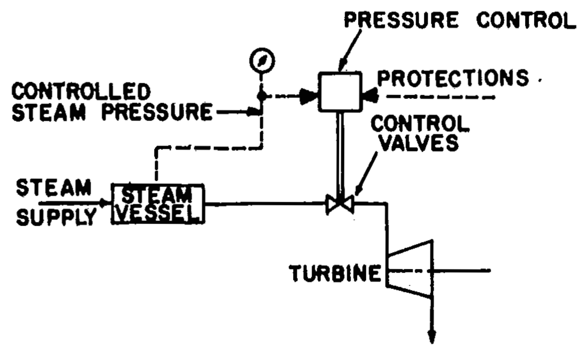

FIG. 7-1 INITIAL PRESSURE CONTROL

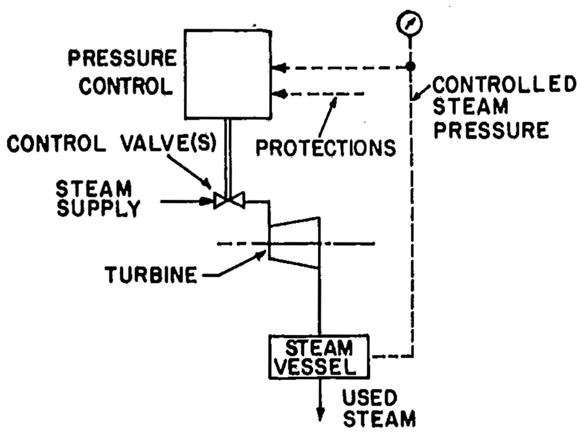

FIG. 7-2 BACK PRESSURE CONTROL

7.01.2 Back Pressure Control controlling the downstream pressure of the turbine or the controlling valve within a specified range, Fig. 7-2.

7.01.3 Extraction Pressure Control controlling one or more extraction pressure(s) within specified limits. Depending on the turbine design, this control can be a combined initial and back pressure control, Fig. 7-3, or a simple back pressure control.

7.01.4 In order to analyze a pressure control system the behavior of the components must be expressed mathematically within a system transfer function to which the performance and stability criteria apply.

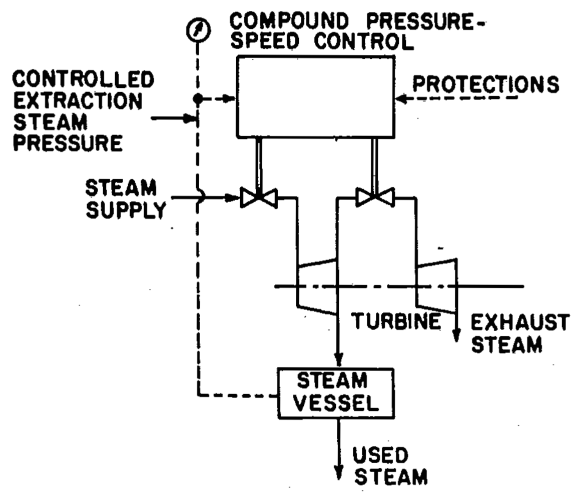

FIG. 7-3 TYPICAL EXTRACTION PRESSURE CONTROL (SINGLE AUTOMATIC)

7.01.5 The following assumes that the Laplace transform representation of differential equations is known. Further information can be found in the literature listed at the end of this section.

7.02 Transfer Functions of Basic Elements (Linear Analysis). The basic elements of a control system are "computing elements" and their operation can be expressed mathematically in a transfer function, describing the relationship of output to input.

A functional classification includes these elements:

(a) Transducers
(b) Summers
(c) Differentiators
(d) Integrators
(e) Amplifiers

Any one device may perform several of these functions simultaneously. The transfer functions of a typical selection of elements are explained in the following paragraphs.

7.02.1 The Transducer. A transducer measures a certain variable and produces an output signal that has a given relation to that variable, including some limits. A mechanical pressure transducer can be a spring-loaded bellows, Fig. 7-4.

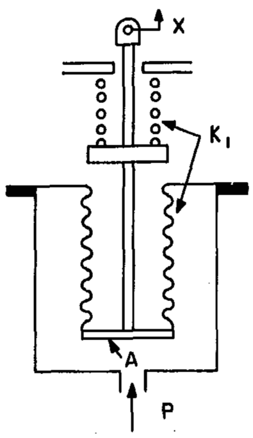

FIG. 7-4 MECHANICAL PRESSURE TRANSDUCER

The transfer function of this pressure transducer is expressed by the following equation

$$7.02.1\text{-}1 \quad \Delta X = \Delta P \frac{A}{K_1}$$

where:

P = input pressure

A = effective bellows area

K_1 = system spring gradient

X = output stroke

An electrical pressure transducer (Fig. 7-5) in the form of a Bourdon tube operating a linear variable differential transformer (LVDT), transforms

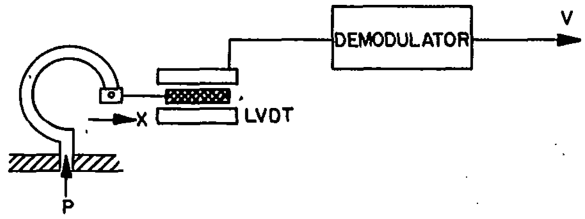

FIG. 7-5 ELECTRICAL PRESSURE TRANSDUCER

pressure into mechanical motion (X) which is converted to an electrical signal by the LVDT and the demodulator.

The transfer function of this pressure transducer can be expressed by the equation

$$7.02.1\text{-}2 \quad \Delta V = K_2 \Delta P$$

where:

P = input pressure

V = output voltage

$K_2 = \frac{\Delta V}{\Delta P}$ = transducer gain

The LVDT is actually a position-measuring device that can be used in such applications as the measurement of valve positions.

7.02.2 The Summer. A summer performs the algebraic summation of two or more quantities. The added quantities can be either variables or constant values. In most cases the summer multiplies each variable with some constant value before adding it.

The simplest mechanical displacement summer is a "floating lever," Fig. 7-6.

The following equation applies to Fig. 7-6.

$$7.02.2\text{-}1 \quad Z = X \left(\frac{b}{a+b}\right) + Y \left(\frac{a}{a+b}\right)$$

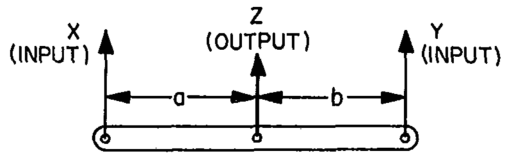

FIG. 7-6 MECHANICAL SUMMER (FLOATING LEVER)

In the particular case where $a=b$, the sum becomes

7.02.2-2 $$Z = \frac{X+Y}{2}$$

Electrically the summation of d-c voltages can be performed by means of a high gain d-c amplifier (A) called "Operational Amplifier," shown in Fig. 7-7.

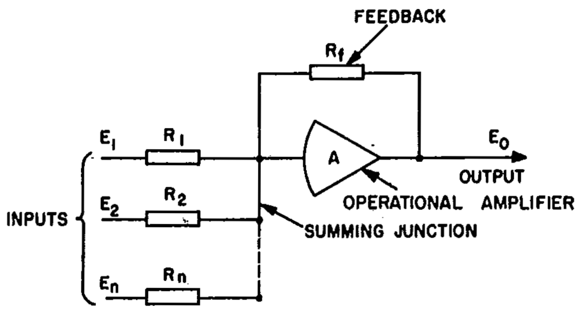

FIG. 7-7 ELECTRICAL SUMMER (WITH RESISTORS)

Here the equation for the output voltage is

7.02.2-3
$$E_o = -\left(E_1 \frac{R_f}{R_1} + E_2 \frac{R_f}{R_2} + \ldots . E_n \frac{R_f}{R_n}\right)$$

NOTE: The output of this summer is always of reversed polarity with respect to the sum of the inputs.

The simple resistors R_f, R_1 .... R_n can be replaced by any other kind of impedances (Z) in order to produce almost any desired transfer function of the circuit.

7.02.3 The Differentiator. The rate of change of a certain variable can be measured by differentiating its value with respect to time: A dashpot (Fig. 7-8) is a mechanical device that acts very much like a differentiator at low input frequencies.

The transfer function of this device is

7.02.3-1 $$\frac{Y(s)}{X(s)} = \frac{T\ s}{1 + T\ s}$$

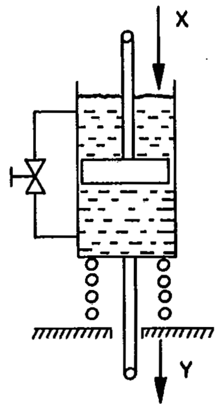

FIG. 7-8 MECHANICAL POSITION DIFFERENTIATOR (FOR LOW FREQUENCY)

As long as

7.02.3-2 $$T\ s << 1$$

($s = j\omega \rightarrow$ very small)

the value of $\frac{Y}{X}$ is close to

7.02.3-3 $$\left(\frac{Y}{X}\right)_{(Ts\ <<\ 1)} = T\ s$$

7.02.3-4 $$Y(s) = T\ X(s)\ s$$

The operator "s" in the numerator signifies a differentiation.

7.02.3-5 $$Y(t) = \frac{dX}{dt}$$

An electrical differentiator can be built with an operational amplifier using a capacitor as the input impedance, Fig. 7-9.

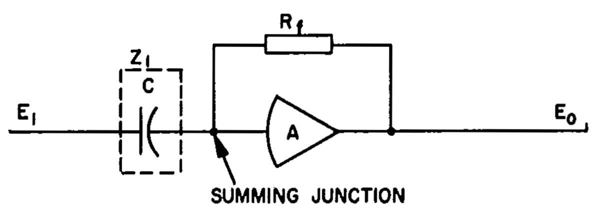

FIG. 7-9 ELECTRICAL DIFFERENTIATOR

The impedance of the capacitor (C) is

7.02.3-6 $$Z_1 = \frac{1}{C\ s}$$

The transfer function of the circuit is, assuming a perfect amplifier

7.02.3-7 $$-\frac{Eo}{E_1} = R_f\ C\ s$$

7.02.3-8 or $$Eo(s) = -R_f\ C\ s\ E_1$$

7.02.3-9 or $$Eo(t) = -R_f\ C\ \frac{dE_1}{dt}$$

7.02.4 The Integrator. An integrator is a device that integrates the value of a variable with respect to time. A good example of a mechanical displacement integrator is the combination of a pilot valve and a piston, Fig. 7-10.

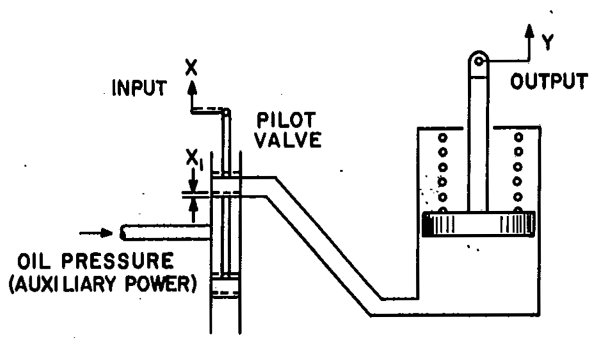

FIG. 7-10 MECHANICAL INTEGRATOR

If the pilot valve is lifted by the amount X_1 above the neutral position, the piston will start moving immediately and keep on traveling at a given speed until the pilot valve is returned to the neutral position, or until the piston reaches a stop (which is also called saturation).

The particular meaning of the term "integration" is shown in Fig. 7-11.

Expressed mathematically, the integration performed in Fig. 7-11 is

7.02.4-1 $$Y(t) = G \int_0^t X(t)\ dt$$

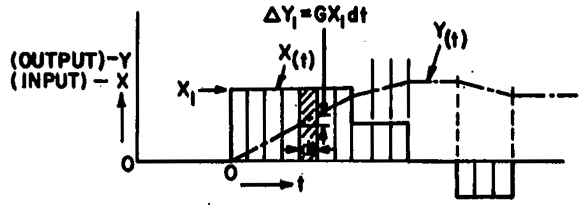

FIG. 7-11 INTEGRATION

Using the Laplace variable s, 7.02.4-1 can be written as

7.02.4-2 $$\frac{Y(s)}{X(s)} = \frac{G}{s}$$

The "s" in the denominator signifies and integration.

G is called the gain of this integrator. It is a constant containing the physical parameters such as oil pressure, port width, piston area and flow coefficients.

Other examples of integrators are the turbine shaft of which the rotational energy is proportional to the time integral of the sum of all torques applied to it, or the pressure in a steam vessel that is proportional to the time integral of the algebraic sum of steam flows into the vessel (flow out of the vessel has a negative sign).

The electrical integrator can be built with an operational amplifier, Fig. 7-12.

The impedance Z_f of the feedback capacitor is

7.02.4-3 $$Z_f = \frac{1}{C_f\ s}$$

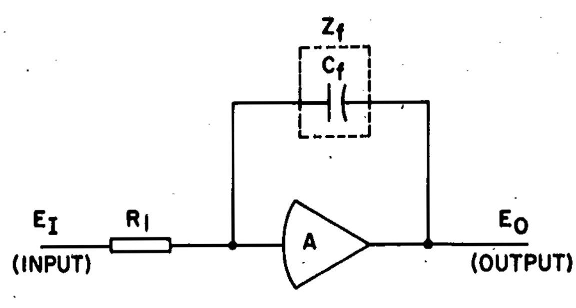

FIG. 7-12 ELECTRICAL INTEGRATOR

and the transfer function of the amplifier circuit for zero initial conditions can be expressed by the equation

$$7.02.4\text{-}4 \qquad E_o(s) = -\frac{E_I(s)}{R_1\ C_f\ s}$$

or

$$7.02.4\text{-}5 \qquad E_o(t) = -\frac{1}{R_1\ C_f}\int_o^t E_I(t)\,dt$$

Similar to the mechanical integrator, $\frac{1}{R_1\ C_f}$ is called the gain of this integrator and its dimension is $\frac{1}{\text{sec}}$. It is the rate at which E_o changes with one volt at the input E_I, (until saturation occurs).

7.02.5 The Amplifier. Amplifiers cover a wide range of devices basically intended to increase the level of a signal to a higher level in magnitude or in force, or in voltage or current to transform a signal in a predetermined way to a higher level

Mechanical Displacement Amplifier. A simple example is a lever, Fig. 7-13.

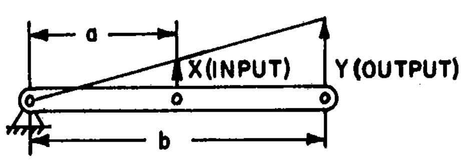

FIG. 7-13 MECHANICAL STROKE AMPLIFIER

Its transfer function is

$$7.02.5\text{-}1 \qquad Y = \frac{b}{a}\ X$$

This amplifier has no time lag; its gain $\frac{b}{a}$ is It amplifies only the displacement, while the energy level of the output is substantially the same as the input.

Mechanical Hydraulic Amplifier. The most common one is the servomotor. It uses hydraulic fluid under pressure for auxiliary power, Fig. 7-14.

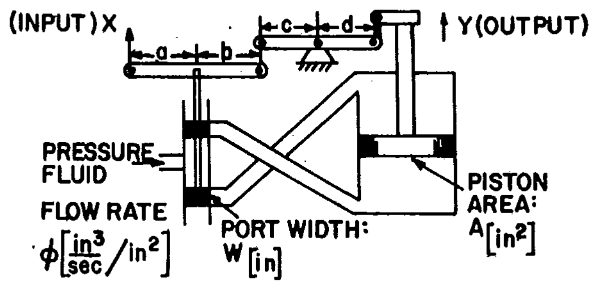

FIG. 7-14 MECHANICAL HYDRAULIC POWER AMPLIFIER (SERVOMOTOR)

The amplifier, shown in Fig. 7-14, can amplify the stroke and the energy level and can be used to drive substantial loads. The output (Y) follows a change in input (X) position with a time lag.

The transfer function of this servomotor is

$$7.02.5\text{-}2 \qquad \frac{Y(s)}{X(s)} = \frac{\frac{bd}{ac}}{1 + T\,s}$$

$\frac{bd}{ac}$ is the steady-state gain, and T is the time constant of the servomotor in seconds.

$$7.02.5\text{-}3 \qquad T = \frac{A}{\frac{ac}{(a+b)\,d}\ W\phi} \quad [\text{sec}]$$

For nomenclature see Fig. 7-14.

A step change of the input X is followed by a movement of Y, as shown in Fig. 7-15. This response is described completely by the transfer function 7.02.5-2

Other elements that have a transfer function similar to the one of the servomotor are:

(a) A steam vessel with a fixed outlet opening, where the steam pressure (ψ) in this vessel is a function of the steam flow (μ) into the vessel:

$$7.02.5\text{-}4 \qquad \frac{\psi}{\mu} = \frac{1}{1 + T\ \ s}$$

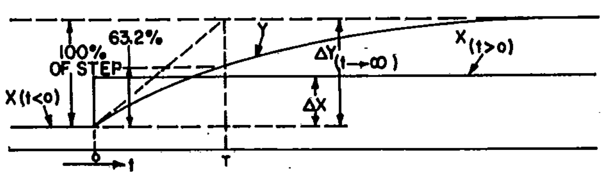

FIG. 7-15 RESPONSE OF SERVOMOTOR

where T is the time it would take to increase the pressure from zero to rated pressure with a constant, rated steam flow into the vessel and no flow out of it.

(b) A turbine rotor on which the rotation losses increase with speed, where the speed (σ) of the rotor is a function of the steam flow μ_{NL} (referred to no load flow at rated speed).

7.02.5-5 $$\frac{\sigma}{\mu_{NL}} = \frac{1}{1 + Ts}$$

where T is the time it would take the rotor to accelerate from zero ($\sigma = 0$) to rated speed ($\sigma = 1$) with rated speed/no load steam flow ($\mu_{NL} = 1$) and no losses.

The three above examples are simple feedback loops represented in block diagram form (Fig. 7-16) generally with R (reference) as independent variable, C as controlled variable, G as the forward loop transfer function and H as the feedback transfer function.

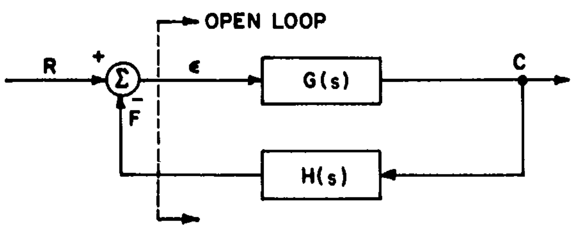

FIG. 7-16 SIMPLE FEEDBACK LOOP

The *open loop* transfer function of this loop is

7.02.5-6 $$\frac{F}{\epsilon} = GH$$

From the open loop transfer function the stability of this loop can be determined.

The *closed loop* transfer function of this loop is

7.02.5-7 $$\frac{C}{R} = \frac{G}{1 + GH}$$

The closed loop transfer function is used for the system transfer function when a feedback sub loop is a part of a system.

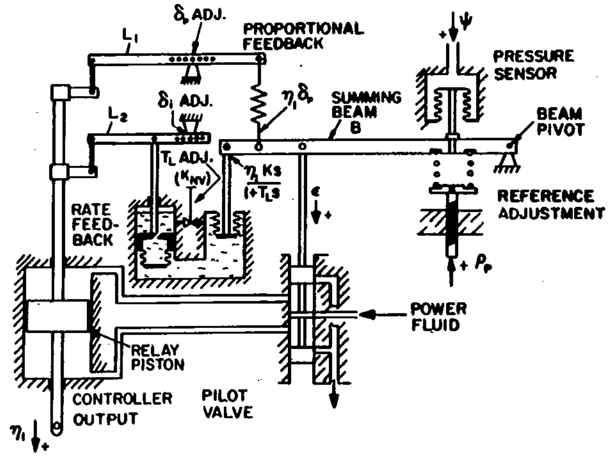

FIG. 7-17 TYPICAL PRESSURE CONTROLLER

7.02.6 Example of Pressure Controller. An example of a versatile pressure controller incorporating the features to achieve any usually needed transfer functions is shown in Fig. 7-17.

The controller consists of a summing beam B on which the forces of the pressure-sensing bellows, ψ, the reference, ρ_P, the steady-state feedback δ_p, and the rate feedback δ_i, are summed with the proper algebraic sign.

The beam operates the pilot valve. The amount the pilot valve is off its neutral (on port) position is the input to the integrator, the relay piston.

The controller output, η_1, operates the subsequent force and displacement amplifier (servomotor) and the feedback lever, L_1, producing the steady-state regulation, δ_p, determined by its lever ratio and feedback spring gradient and also the feedback lever, L_2, producing the instantaneous regulation, δ_i, by means of the lever ratio, L_2, and the subsequent hydraulic differentiator.

A controller of this basic design, with all pivots and rod connections on the summing beam built as flexure pivots and a rotating pilot-valve bushing, has been shown to have a deadband of less than 0.01 per cent.

The transfer functions of this controller, for the different possible modes of operation are as follows:

(a) Proportional Control – (Rate feedback disconnected or needle valve open, $K_{NV} = \infty$) – block diagram, Fig. 7-18.

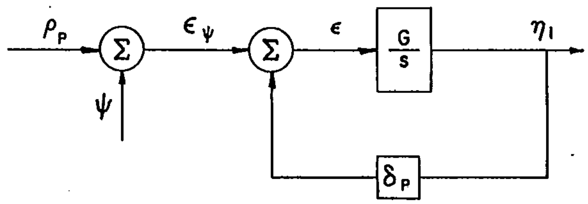

FIG. 7-18 BLOCK DIAGRAM OF CONTROLLER FOR PROPORTIONAL CONTROL

For a constant reference set ($\rho_P = 0$)

$$7.02.6\text{-}1 \qquad \epsilon_\psi = \psi$$

The transfer function for η_1/ψ can be written as

$$7.02.6\text{-}2 \qquad \frac{\eta_1}{\psi} = \frac{\frac{1}{\delta_P}}{1 + T_R\, s}$$

$$7.02.6\text{-}3 \quad \text{where} \quad T_R = \frac{1}{G\,\delta_P}$$

$1/G$ is the time it would take for the output η_1 to go through full stroke (unit stroke) if a pressure error of rated pressure was applied (extrapolated, from small error without saturation and no feedback was applied). This time is given by the layout of the pressure-sensing element, including the gradients of all connected springs and the hydraulic characteristic of the pilot valve and piston.

(b) Constant-Pressure Control (porportional plus integral) – block diagram, Fig. 7-19.

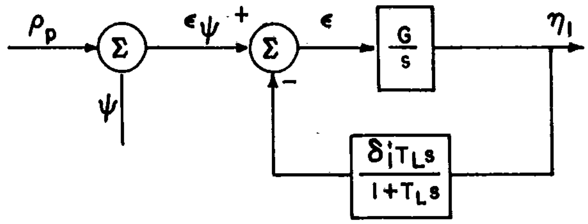

FIG. 7-19 BLOCK DIAGRAM OF CONTROLLER FOR PROPORTIONAL PLUS INTEGRAL CONTROL THE TRANSFER FUNCTION OF THIS CONTROLLER IS (AGAIN $\rho_P = 0$)

$$7.02.6\text{-}4 \qquad \frac{\eta_1}{\psi} = \frac{\frac{G}{1 + G T_L\, \delta_i}(1 + T_L\, s)}{s\left(1 + \frac{T_L}{1 + G T_L\, \delta_i}\, s\right)}$$

where G has the same significance as in the preceding paragraph, and $T_L = C/K_{NV}$ where C is a mechanical constant and K_{NV} is the flow factor [cu in./sec psi] in the damping needle valve.

The steam-seal regulator can be a controller of this type.

(c) Proportional and Partial Integral Control – block diagram, Fig. 7-20.

The transfer function of the controller shown in Fig. 7-20 is

$$7.02.6\text{-}5 \qquad \frac{\eta_1}{\psi} = \frac{\frac{1}{\delta_P}\,(1 + T_L s)}{(1 + T_{R_1} s)(1 + T_{R_2}\, s)}$$

$$7.02.6\text{-}6 \quad T_{R_1}(-),\ T_{R_2}(+) = \frac{1}{\frac{1}{2}\left[\frac{1}{T_L} + G(\delta_i + \delta_P)\right] \pm \frac{1}{2}\left\{\left[\frac{1}{T_L} + G(\delta_i + \delta_P)\right]^2 - \frac{4G\,\delta_P}{T_L}\right\}^{1/2}}$$

and T_L again is

$$7.02.6\text{-}7 \qquad T_L = \frac{C}{K_{NV}}$$

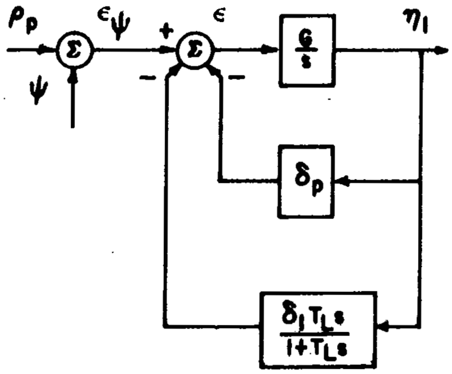

FIG. 7-20 BLOCK DIAGRAM FOR CONTROLLER WITH PROPORTIONAL-PLUS-PARTIAL-INTEGRAL CONTROL

With aproper choice of G, T_L, δ_i and δ_P it is possible to adapt this controller to most pressure-control problems likely to be encountered in connection with steam turbines.

It is understood, however, that on many applications a less versatile controller will do the job, provided it is designed for the particular condition.

It is also possible to control the pressure in a small volume by an integrating controller, provided the gain of the integrator is made small enough.

7.03 Frequency Response. The frequency response of a control element or system is used to determine performance and stability of a loop or a system.

In order to use the transfer function for the frequency analysis, the Laplace variable "s" is replaced by the term $j\omega$, where $j = \sqrt{-1}$ and ω is the angular frequency (rad/sec) of a sinusoidal oscillation imposed to the input of a loop.

7.03.1 Simple Integrator with Feedback

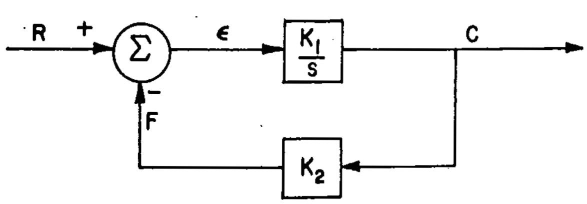

FIG. 7-21 SIMPLE INTEGRATOR FEEDBACK LOOP

The *open loop* transfer function is

7.03.1-1 $$\frac{F}{\epsilon} = \frac{K_1 K_2}{s} = \frac{K_1 K_2}{j\omega}$$

The magnitude of $\frac{F}{\epsilon}$ decreases linearily if the frequency ω increases.

The phase angle of F with respect to ϵ is always -90° (inherent characteristic of integrator).

The *closed loop* transfer function is

7.03.1-2 $$\frac{C}{R} = \frac{K_1}{s\left(1 + \frac{K_1 K_2}{s}\right)} = \frac{\frac{1}{K_2}}{1 + T \cdot s}$$

$$= \frac{\frac{1}{K_2}}{1 + T\ j\omega}$$

7.03.1-3 where

$$T = \frac{1}{K_1 K_2}$$

7.03.2 The magnitude of $\frac{C}{R}$ is approximately $\frac{1}{K_2}$ as long as $T\omega \ll 1$ or $\omega \ll 1/T$. The magnitude is $\frac{1}{K_2 T\omega}$ when $\omega \gg 1/T$. This is called a lag break at

$\omega = \frac{1}{T}$ (corner frequency). A plot of magnitude versus frequency (in log/log scales) is shown in Fig. 7-22 (Bode Diagram).

For vertical scale (magnitude) usually *decibels* (db) are used:

7.03.2-1 $$M(\text{db}) = 20 \times \log_{10}(M)$$

A transfer function of

7.03.2-1 $$\frac{C}{R} = 1 + T_L s$$

would result in an upward break (lead break at the $\frac{1}{T_L}$ (see dotted line in Fig. 7-22).

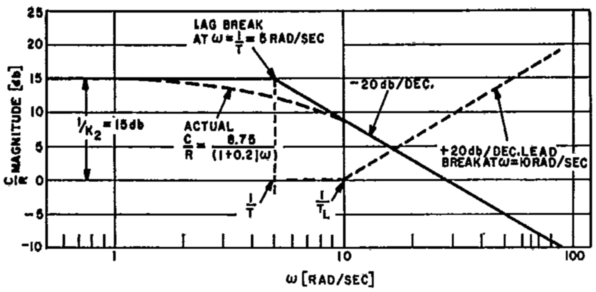

FIG. 7-22 BODE DIAGRAM OF SIMPLE LAG BREAK (DOTTED – LEAD BREAK)

7.03.3 The phase lag (or lead) contribution (ϕ) of a break in a Bode diagram at the frequency (ω) with respect to a given frequency (ω_c) can be found with the equation

$$7.03.3\text{-}1 \qquad \phi = \tan^{-1}\left(\frac{\omega}{\omega_c}\right) \quad \text{[degrees]}$$

A *servomechanism scale* can be designed for any given semi-log paper that can be placed on the Bode plot with the 45° mark on the frequency ω and the phase lag (or lead) contribution of each break can be read on the scale.

The equation for the design of the scale is

$$7.03.3\text{-}2 \qquad \phi = \cot^{-1}\left(\frac{\omega}{\omega_c}\right) \quad \text{[degrees]}$$

Values for such a scale are given in Table 7-23.

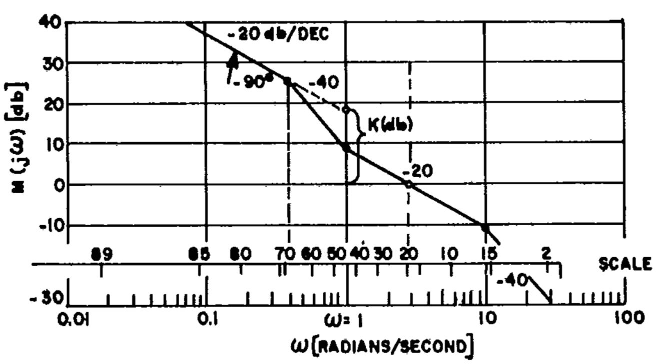

FIG. 7-24 EXAMPLE OF USE OF SERVOMECHANISM SCALE

Table 7-23

Frequency (ω)	Phase shift, [deg]
$0.018\,\omega_c$	89
$0.089\,\omega_c$	85
$0.176\,\omega_c$	80
$0.364\,\omega_c$	70
$0.577\,\omega_c$	60
$0.840\,\omega_c$	50
$1.000\,\omega_c$	45
$1.19\ \omega_c$	40
$1.73\ \omega_c$	30
$2.75\ \omega_c$	20
$5.68\ \omega_c$	10
$11.20\ \omega_c$	5
$57.00\ \omega_c$	1

A general example is shown in Fig 7-24 for the following typical transfer function.

$$7.03.3\text{-}3 \quad M(s) = \frac{K\,(1+s)}{(1+2.5\,s)\,(1+0.1\,s)}$$

The phase angle ϕ of the output with respect to the input at the frequency $\omega = 1$ is (notice 45° of servo scale placed at $\omega = 1$):

The –20 db/dec initial slope contributes –90° (the lag break is far to the left!).

The $\omega = 0.4$ lag break contributes –68°

The $\omega = 1$ lead break contributes +45°

The $\omega = 10$ lag break contributes –6°

The phase angle ϕ of the output with respect to the input at $\omega = 1$ is:

$$7.03.3\text{-}4 \quad \phi = -90 - 68 + 45 - 6 = -119 \text{ [degrees]}$$

7.03.4 Stability. For stability it is necessary for the phase angle to be less than –180° at the frequency where the magnitude $M(s) = 1$

$M(s)$ [db] = 0: *Zero Decibels*

The phase margin (γ_c) *must be greater than zero.* See evaluation of phase margin (7.10).

The *margin* that is present in a system, compared to the maximum permissible phase shift of $-180°$ is called *phase margin.*

The phase margin (γ_c) is

7.03.4-1 $\gamma_c = 180 - \Sigma\, \phi_{lag} + \Sigma\, \phi_{lead}$ [deg]

where $\Sigma\, \phi_{lag}$ is the sum of the phase angle contribution of all lag breaks with respect to the crossover frequency ω_c

and $\Sigma\, \phi_{lead}$ is the sum of the phase angle contribution of all lead breaks with respect to ω_c. See Fig. 7-25.

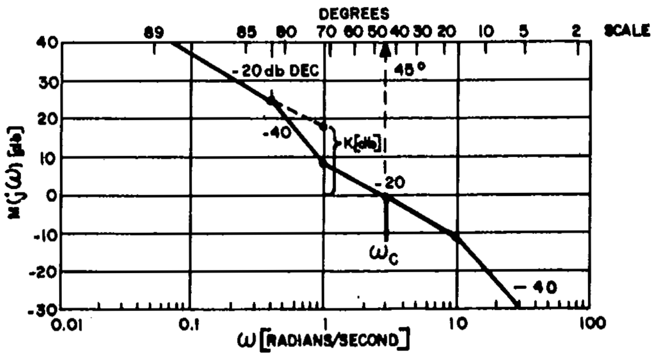

FIG. 7-25 EXAMPLE OF USE OF SERVOMECHANISM SCALES FOR CROSSOVER FREQUENCY

The phase margin of the system in Fig. 7.25 is

7.03.4-2 $\gamma_c = 180 - (90 + 82 + 17) + 71 = +62$ [deg]

7.04 Nonlinear Analysis. Pressure-control systems often exhibit the phenomenon of steady-state oscillations or "limit cycles." These oscillations may be caused by nonlinearities in the system and it is important that they be considered when analyzing performance.

The most important nonlinearity in pressure control systems is the deadband in the pressure controller, relay, and servomotor.

The method to analyze nonlinearities used here is the *describing function.*

7.04.1 The describing function of a nonlinear element is obtained by comparing the fundamental of the output-wave form to an input wave to determine phase shift and gain as a function of amplitude. The complex ratio of the approximated output to the input then becomes the describing function of the element. Because of the approximation of a nonsinusoidal wave form by its fundamental, it is difficult to predict amplitude and frequency of any limit cycle closer than about 20 per cent. Also, this method cannot be used when more than one nonlinearity occurs in the system.

For quantitative analysis where two or more nonlinearities are present a dynamic analysis with analog or digital computers is recommended.

Since deadband is the predominant nonlinearity in pressure-control loops, it will be considered in detail. An approach is presented which is based on describing functions, but is extended to allow the use of Bode plots.

7.04.2 Analysis of Nonlinear Element. The block diagram of a nonlinear element in general is shown in Fig. 7-26.

In Fig. 7-26, *NL* is the amplitude sensitive nonlinearity and $G(j\omega)$ is the linear, frequency sensitive element.

For the purpose of the analysis a sine wave is applied to the input of the nonlinear element: This input is represented by the equation

7.04.2-1 $\epsilon_1 = E_1\, e^{j\, \omega t}$

The fundamental of the output of the nonlinear element is

7.04.2-2 $\epsilon_2 = E_2\, e^{j\,(\omega t + \phi)}$

where E_2 and ϕ are function of the magnitude E_1 of the input. The ratio of ϵ_2 to ϵ_1 for this input at E_1 is given by

7.04.2-3 $DF(E_1) = \dfrac{\epsilon_2}{\epsilon_1} = \dfrac{E_2\, e^{j\,(\omega t + \phi)}}{E_1\, e^{j\, \omega t}}$

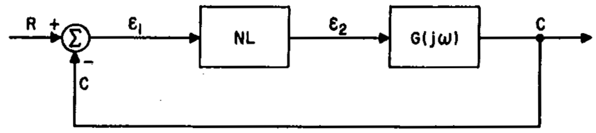

FIG. 7-26 CONTROL LOOP WITH NONLINEARITY (GENERAL)

where $DF(E_1)$ is defined as the describing function of NL as a function of E_1. Reduction of of this equation results in

7.04.2-4 $\quad DF(E_1) = K(E_1)\, e^{j\phi(E_1)}$

Thus, the describing function is a complex number which varies in magnitude and angle as the amplitude of the input signal changes. This is analogous to the variation of frequency-dependent elements in magnitude and angle as the frequency changes.

A condition for oscillations to occur in a system requires that the gain around the loop be unity at the frequency where the phase margin is zero deg. The equation to be satisfied is

7.04.2-5 $\quad G(j\omega)\,DF(E_1) = e^{-j\pi} = -1$

or

7.04.2-6 $\quad G(j\omega) = \dfrac{-1}{DF(E_1)}$

The magnitude and frequency of oscillations can be determined by plotting $G(j\omega)$ and $-1/DF(E_1)$ in the complex plane. If the two curves intersect, oscillations will occur with approximately the frequency and magnitude at the intersection, since equation (25) is satisfied at that point.

Linear stability theory and the describing-function method are both based on an examination of the phase shift around a loop at all frequencies. If the phase shift exceeds −180 deg at unity gain for any frequency, the system will oscillate at that frequency. It is possible on a Bode graph to predict approximately the frequency at which oscillations will occur for deadband nonlinearities. The amplitude of the oscillations can also be estimated by a method described below:

7.04.3 Deadband and Integrator Loop. A nonlinear element with deadband typical for a pressure control loop is shown in block diagram form in Fig. 7-27.

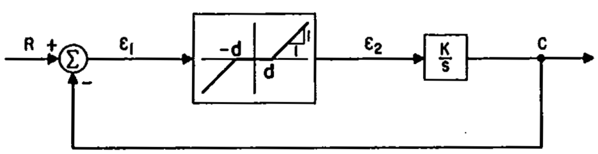

FIG. 7-27 CONTROL LOOP WITH DEADBAND IN INTEGRATOR INPUT

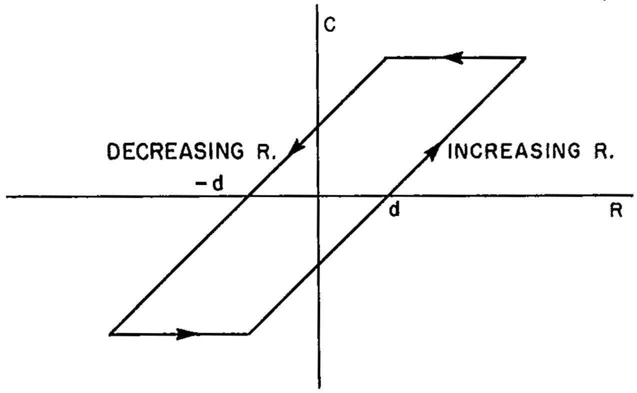

FIG. 7-28 HYSTERESIS RESPONSE LOOP RESULTING FROM DEADBAND

The linear transfer function of this element is

7.04.3-1 $\quad \dfrac{C}{R} = \dfrac{1}{1 + \frac{1}{K}s}$

The steady state response of the nonlinear loop with deadband is a *hysteresis loop*, shown in Fig. 7-28.

7.04.4 System with Deadband and Integrator Loop. In the Bode diagram the linear loop produces a lag break at the frequency K [rad/sec]. The nonlinear loop will produce a lag break close to the frequency K as long as the magnitude of the input oscillation (M) is large compared to the deadband "d". As the magnitude (M) decreases, the break frequency will become lower. In particular, when the magnitude (M) becomes equal to or smaller than "d", the break frequency (ω_b) will go to zero.

An example of a control system containing a nonlinear (deadband and integrator) load is shown in Fig. 7-29.

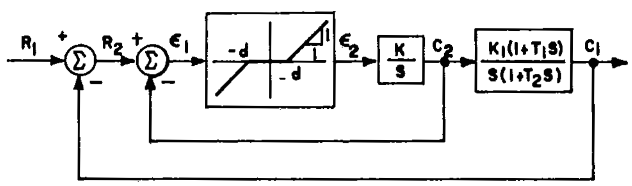

FIG. 7-29 SYSTEM WITH ONE DEADBAND AND INTEGRATOR LOOP

The open loop transfer function of the linear system is:

$$7.04.4\text{-}1 \quad \frac{C_1}{R_2} = G(s) = \frac{K_1\,(1 + T_1\,s)}{s\,(1 + T_2\,s)}$$

Typical values used in this transfer function are:

$K_1 = 14\ (\approx 23\ \text{db})$

$T_1 = 0.31\ \text{sec}\ (1/T_1 = 3.2\ \text{rad/sec})$

$T_2 = 0.66\ \text{sec}\ (1/T_2 = 1.5\ \text{rad/sec})$

$\omega_b = 20\ \text{rad/sec}$

The Bode diagram of this system with the listed values for K_1, T_1, T_2 and ω_b is shown in Fig. 7-30.

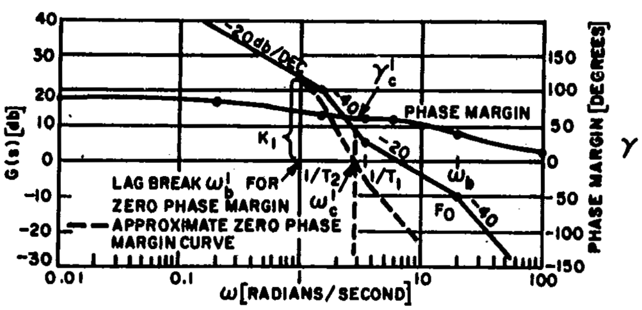

FIG. 7-30 BODE PLOT FOR SYSTEM WITH SINGLE DEADBAND

The determination of the effect of the nonlinear subloop is performed by letting the break frequency ω_b of the linear loop decrease in successive steps, while observing the phase margin of the nonlinear system (see 7-30). The location of the attenuation characteristic corresponding to ω_b' where the crossover phase margin is just zero is indicated in dotted lines.

7.04.5 Frequency of Sustained Oscillation. The nonlinear crossover frequency ω_c' (2.9 rad/sec in example) is approximately the frequency at which the system will oscillate.

The phase lag produced by the nonlinear subloop (ϕ_{NL}) is responsible for cancelling the phase margin (γ_c') of the linear system at the frequency

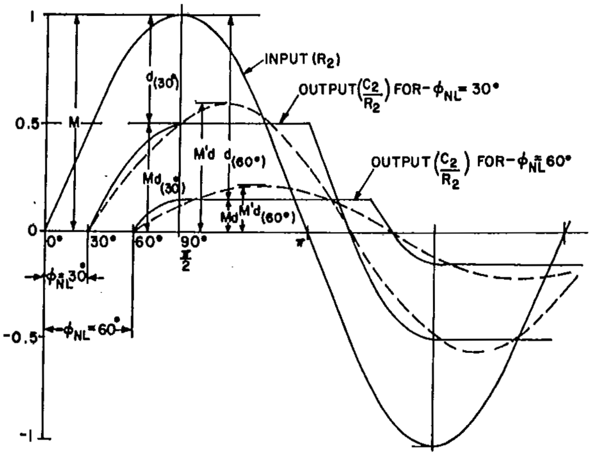

FIG. 7-31 DESCRIBING FUNCTION FOR ESTIMATING MAGNITUDE OF SUSTAINED OSCILLATION AT ω_c'

(ω_c') ($\gamma_c' \approx 60°$ in example). The conditions for oscillation are, therefore, met at ω_c' when

$$7.04.5\text{-}1 \quad -\phi_{NL} = \gamma_c'.$$

7.04.6 Magnitude of Sustained Oscillation. The magnitude of sustained oscillations can be estimated from a plot of the describing function with a given phase lag ϕ_{NL} (see Fig. 7-31): M is the input magnitude (normalized to unity), M_d is the actual output magnitude that produces the phase lag ϕ_{NL}, M_d' is the fundamental sine wave approximating M_d. The magnitude of the "half deadband" is again "d" which is:

$$7.04.6\text{-}1 \quad d = 1 - M_d\ [\text{per unit of } M]$$

The procedure used in Fig. 7-31 is directly applicable if

$$7.04.6\text{-}2 \quad \omega_c' < \frac{\omega_b}{10}\ [\text{rad/sec}]$$

If $\omega_c' > \frac{\omega_b}{10}$ the phase lag $\phi_L\,(\omega_c')$ of the linear subloop should be added in order to estimate the magnitude of sustained oscillations. The approximate ratio of the magnitude of sustained oscillations to the deadband M/d is shown in Fig. 7-32 as a function of the phase margin of the linear system γ_c' at the crossover frequency ω_c' of

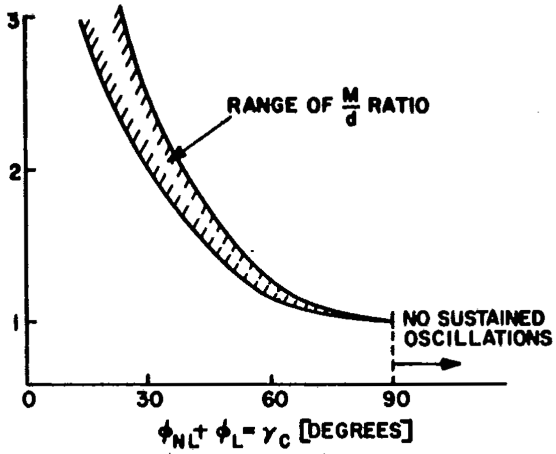

FIG. 7-32 $\frac{M}{d}$ ESTIMATE AS FUNCTION OF γ_c'

the nonlinear system. A range is given rather than a single line because of the nonsinusoidal wave shape of the actual output.

Note: The deadband *"d"* and the magnitude *M* in Fig. 7-31 are given as "half deadband" and zero to peak magnitude, respectively. If a deadband as defined in 2.03.3 is used, Fig. 7-32 is still applicable but it will yield *peak to peak magnitude of M.*

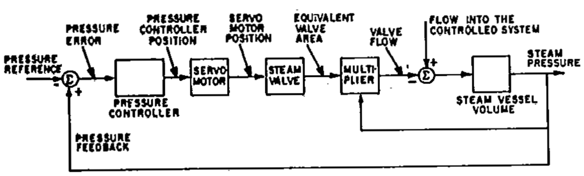

FIG. 7-33 BLOCK DIAGRAM OF INITIAL-PRESSURE CONTROL SYSTEM

7.05 Analysis of Pressure Control Systems

7.05.1 To describe the pressure control system quantitatively, the transfer functions of each member are introduced.

Generally the pressure-control system can be represented in block-diagram form, as shown in an example of an initial pressure control in Fig. 7-33.

7.05.2 *Dimensionless Representation.* To make the analysis as universally applicable as possible, a dimensionless unit system is chosen that permits writing the transfer functions of the system with a minimum of detailed knowledge of its components.

7.05.3 *Reference Terms for Dimensionless (Unitized) Representation.* To make the variables in the control system independent of the size and to some extent independent of the design of the system, they are made dimensionless according to a well-defined unitizing system. The variables are in general:

7.05.3-1 Pressure $\psi = \frac{\Delta P}{P_r}$

Pressure change [psi] referred to rated pressure [psi (a)].

7.05.3-2 Stroke (of hydraulic relay or servomotor) $\eta = \frac{\Delta Y}{Y_r}$

Stroke change [in.] referred to stroke change [in.] necessary to go from zero to rated steam flow.

7.05.3-3 Equivalent Valve Area $\alpha = \frac{\Delta A}{A_r}$

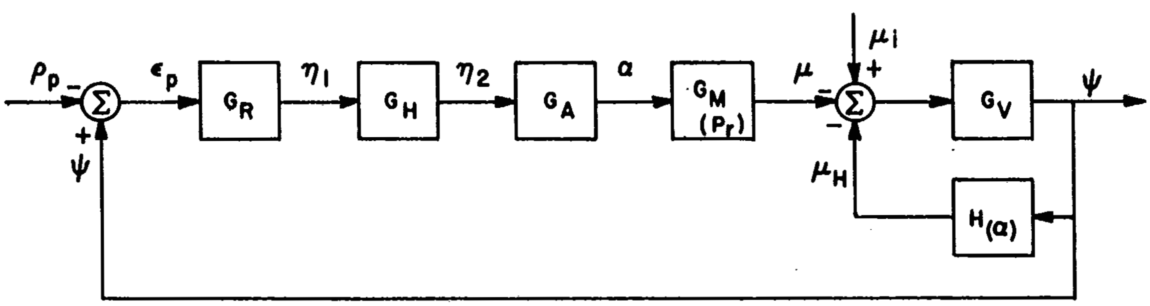

FIG. 7-34 BLOCK DIAGRAM OF INITIAL-PRESSURE CONTROL SYSTEM USING SYMBOLS

Equivalent flow area change [sq in.] referred to equivalent area [sq in.] corresponding to rated flow.

7.05.3-4 Where the simplified flow equations is $m = AP$

7.05.3-5 Flow $\mu = \frac{\Delta m}{m_r}$

Flow change [lb/sec] referred to rated flow [lb/sec].

7.05.3-6 Reference $\rho_P = \frac{\Delta R}{R_r}$

Reference change referred to reference value for rated pressure.

7.05.3-7 Regulation (droop) $\delta_P = \frac{\Delta P^*}{P_r}$

Steady-state pressure change [psi] that will put the regular output through unit stroke referred to rated pressure [psi (a)].

7.05.3-8 Pressure Controller Gain

$$G_P = \frac{1}{\delta_P}$$

Steady-state gain of pressure controller [per unit].

With these variables incorporated and the *multiplier* eliminated as explained in 7.06 the system is represented by the block diagram, Fig. 7-34.

7.06 Transfer Functions of Elements of Pressure Control System

7.06.1 Pressure Controller. (See 7.02.6.) Depend-on the required steady-state accuracy and stability, three pressure-controller transfer functions are common:

$$G_R = \frac{\frac{1}{\delta_P}}{1 + T_R s} \quad (7.06.1\text{-}1)$$

for strictly proportional control, where δ_P is the regulation in unitized form, for instance:

5-per cent regulation: $\delta_P = 0.05$

$$G_R = \frac{K_R (1 + T_L s)}{s (1 + T_R s)} \quad (7.06.1\text{-}2)$$

for constant pressure (called proportional plus integral) control where

$$K_R = \frac{G}{1 + GT_L \delta_i} \quad (7.06.1\text{-}2a)$$

See 7.02.6 for significance of G, T_L, and δ_i

$$G_R = \frac{\frac{1}{\delta_P}(1 + T_L s)}{(1 + T_{R_1} s)(1 + T_{R_2} s)} \quad (7.06.1\text{-}3)$$

for a controller with a broad instantaneous regulation δ_i and a narrow steady state regulation δ_P (also called proportional and partial integral).

A pressure controller that can be used for any one of these three transfer functions is described in 7.02.6.

7.06.2 Hydraulic Servomotor (G_h). (See 7.02.5.) The linear transfer function of this simple hydraulic force and stroke amplifier is

$$G_h = \frac{1}{1 + T_2 s} \quad (7.06.2\text{-}1)$$

7.06.3 Steam Valve Area (G_A)

$$G_A = 1 \quad (7.06.3\text{-}1)$$

7.06.4 Steam Flow (G_m)

$$G_m (P_r) = 1 \quad (7.06.4\text{-}1)$$

7.06.5 Steam Vessel (Volume), (G_V). The steam vessel acts as an integrator. Any steam flow into the vessel that is not cancelled by an equal flow out of the vessel (flow error) will increase the pressure at a rate given by the gain K_V of this integrator. This gain is

$$K_V = \frac{1}{T_V} = \frac{m_r}{W_r} \quad (7.06.5\text{-}1)$$

The characteristic time T_V of the steam volume is

$$T_V = \frac{W_r}{m_r} \quad (7.06.5\text{-}2)$$

The transfer function of the steam vessel is

$$G_V = \frac{K_V}{s} = \frac{1}{T_V s} \quad (7.06.5\text{-}3)$$

7.06.6 Pressure – Flow Feedback. This feedback must be determined by linearizing the multiplier (see Fig. 7-33).

Assuming that the flow through the valve is proportional to the equivalent area times pressure, whereby the equivalent area has been linearized in the valve drive (cams) or in the valve itself (profiled valve), multiplication of two variables is present which is not usable in a linear analysis.

To linearize this multiplier, the absolute differential is formed which describes the change as a sum:

7.06.6-1 $$\Delta\mu = \left(\frac{\partial\mu}{\partial\alpha}\right)_{P = \text{const}} \times d\alpha + \left(\frac{\partial\mu}{\partial\psi}\right)_{A = \text{const}} \times d\psi$$

Since the pressure signal varies only slightly from rated pressure for all operating points, the first partial differential can be evaluated as

7.06.6-2 $$\left(\frac{\partial\mu}{\partial\alpha}\right)_{P = \text{const}} \cong \left(\frac{\partial\mu}{\partial\alpha}\right)_{P_r}$$

which is the relative flow change μ per unit of change in α which is by definition of the units

7.06.6-3 $$\left(\frac{\partial\mu}{\partial\alpha}\right)_{P_r} = 1$$

Since a unit change in η_2 produces an equivalent area change α of unity,

$$G_A = 1 \quad (7.06.3\text{-}1)$$

Therefore

7.06.6-4 $$G_A \times G_m(P_r) = 1$$

except for some noncompensated nonlinearity in G_A. The value describing such deviation from unit gain, is called K_3. The second part of the absolute differential $\left(\frac{\partial\mu}{\partial\psi}\right)_{A = \text{const}} \times d\psi$ can now be introduced in the flow summing point.

In the initial pressure-control application, Fig. 7-34, where μ and μ_H have the same direction (same sign), the steam-volume portion of the loop is represented by Fig. 7-35.

The feedback $H(\alpha)$ is approximately equal to μ; therefore, the time constant of the closed loop depends on the operating condition.

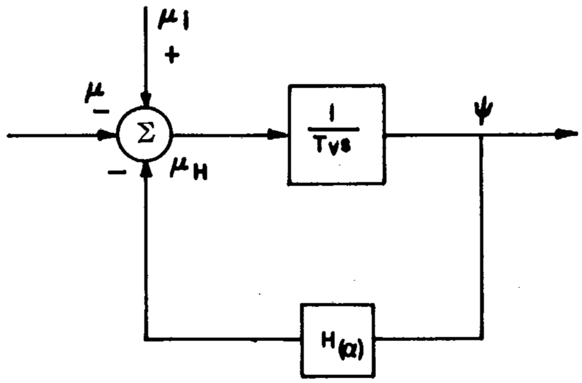

FIG. 7-35 BLOCK DIAGRAM FOR FLOW-VOLUME-PRESSURE RELATION FOR INITIAL-PRESSURE CONTROL SYSTEM

For $\mu = 1$: $H = 1$

For $\mu = 0$: $H \approx 0$ (except some leakage)

The time constant of the closed loop is

7.06.6-5 $$T = \frac{T_v}{H}$$

T increases inversely proportional to H. At the same time, the gain of the closed loop increases. The transfer function for $0 < H < 1$ is

7.06.6-6 $$\frac{\psi}{\mu_i - \mu} = \frac{\frac{1}{H(\alpha)}}{1 + \frac{T_v}{H(\alpha)} s}$$

This can be visualized with the following example: The final pressure to pass the steam flow $\mu = 1$ at an opening of 0.5 would be twice the rated pressure, if no other variables were changed.

The lower the valve opening becomes, the lower H will become, and at $\alpha = 0(+)$ or $\mu = 0(+)$ the transfer function will be

7.06.6-7 $$\left(\frac{\psi}{\mu_i - \mu}\right)_{H \to 0} = \frac{1}{T_v s}$$

Usually the time constant T_v is large compared to other time constants in the system so that for $s \gg 0$

7.06.6-8 $$\frac{1}{1 + T_v s} \approx \frac{1}{T_v s}$$

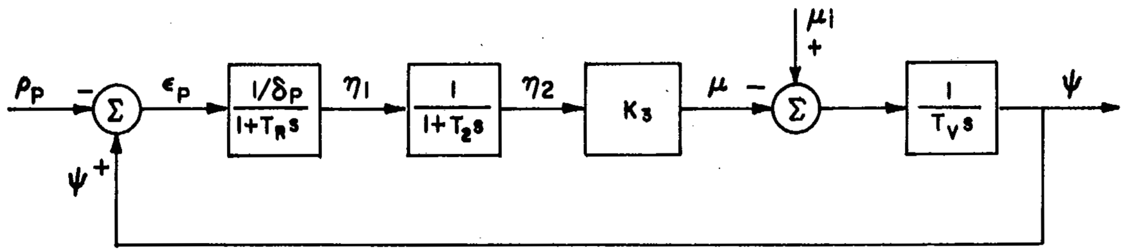

FIG. 7-36 BLOCK DIAGRAM FOR PROPORTIONAL INITIAL-PRESSURE CONTROL WITH LARGE STEAM VESSEL

In other words, the phase lag of $1/(1 + T_v s)$ with respect to the crossover frequency is substantially the same as the one produced by $1/T_v s$, or 90 deg. It is therefore acceptable in most cases to consider the steam volume as an integrator.

7.07 Proportional Initial Pressure Control System (Linear Analysis)

7.07.1 With the transfer functions of the elements, the block diagram is as shown in Fig. 7-36.

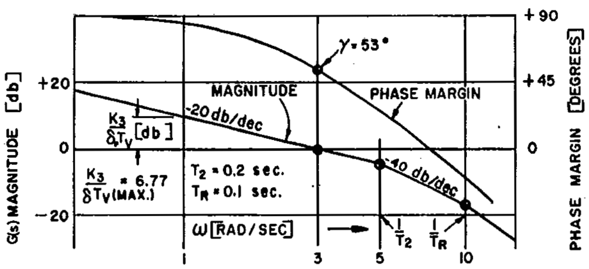

FIG. 7-37 BODE DIAGRAM OF PROPORTIONAL-PRESSURE CONTROL SYSTEM

7.07.2 The stability of this system can be evaluated from the over-all open loop transfer function which is:

7.07.2-1

$$G(s) = \frac{\psi}{\epsilon_P} = \frac{\dfrac{K_3}{\delta_P T_v}}{s(1+T_R s)(1+T_2 s)}$$

7.07.3 The Bode diagram of this loop for the values of $\dfrac{K_3}{\delta_P T_v}$, T_R and T_2 marked in the figure is shown in Fig. 7-37.

7.07.4 The quantity which most likely can be adapted to obtain stable operation is δ_P. It can be seen that a larger regulation δ_P produces a more stable system at the expense of a larger steady-state pressure error. The stability must be evaluated at the value $(K_3/\delta_P T_v)$ max which will yield the smallest phase margin.

7.07.5 The required phase margin to assure acceptably small sustained oscillation depends on the deadband of the different components of the system. See evaluation of phase margin (7.10).

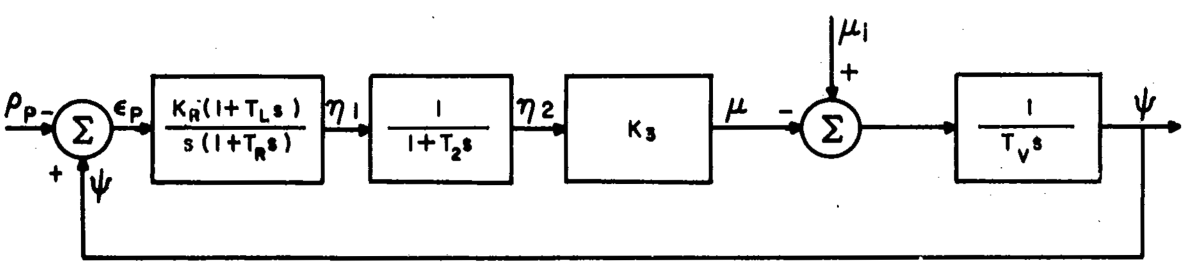

FIG. 7-38 BLOCK DIAGRAM OF INITIAL PRESSURE CONTROL WITH PROPORTIONAL AND INTEGRAL CONTROLLER (CONSTANT PRESSURE)

7.08 Proportional Plus Integral Initial Pressure Control System (Linear Analysis)

7.08.1 The block diagram of an initial pressure control with proportional plus integral controller (see 7.02.6-4) is shown in Fig. 7-38.

7.08.2 The open loop system transfer function is:

7.08.2-1

$$G(s) = \frac{\frac{K_R K_3}{T_v}(1+T_L s)}{s^2(1+T_R s)(1+T_2 s)}$$

where

7.08.2-2

$$K_R = \frac{G}{1 + G\delta_i T_L}$$

7.08.3 The Bode diagram of a typical system is shown in Fig. 7-39.

7.08.4 This system is conditionally stable. If the gain of it is raised or lowered too far, it will become unstable. Systems of this kind will operate satisfactorily only with control devices with small deadband.

Also, a pressure control of this kind cannot be paralleled with another pressure controller.

A typical application for this type of control is the steam-seal pressure controller.

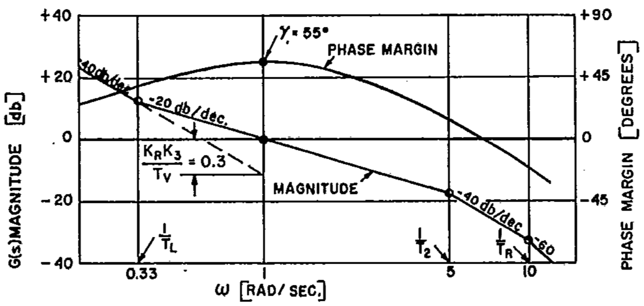

FIG. 7-39 BODE DIAGRAM OF INITIAL PRESSURE CONTROL SYSTEM WITH PROPORTIONAL PLUS INTEGRAL CONTROLLER

7.09 Proportional Plus Partial Integral Initial Pressure Control System (Linear Analysis)

7.09.1 The block diagram of an initial pressure control system with proportional plus partial integral controller is shown in Fig. 7-40.

7.09.2 The open loop system transfer function is:

7.09.2-1

$$G(s) = \frac{\frac{K_3}{\delta_P T_v}(1+T_L s)}{s(1+T_{R_1} s)(1+T_{R_2} s)(1+T_2 s)}$$

7.09.3 The Bode diagram of this system is shown in Fig. 7-41.

7.09.4 This system has an almost constant degree of stability over a certain gain range $2 \leq K_3/\delta_P T_v \leq 10$ and usually δ_P, T_r and T_L are made adjustable within practical limits to be able to adjust for best performance.

7.09.5 This is the most common-type controller preferred because of its capability of relatively small steady-state pressure error combined with good stability and the capability of operating in parallel with other controllers (flow sharing).

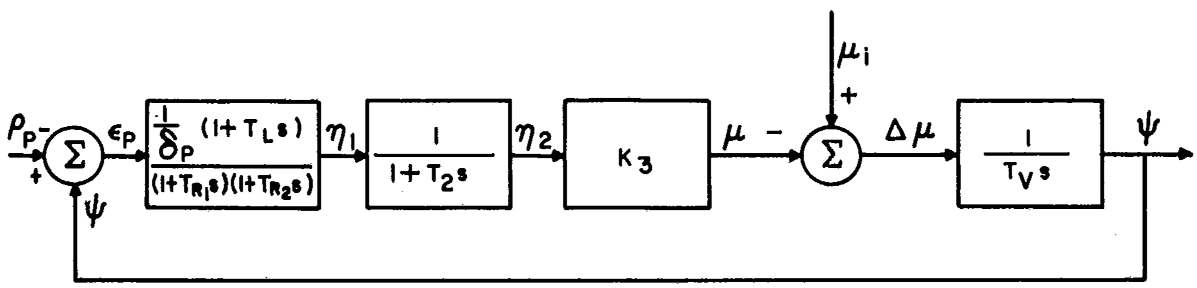

FIG. 7-40 BLOCK DIAGRAM OF PROPORTIONAL PLUS PARTIAL RESET-PRESSURE CONTROL SYSTEM

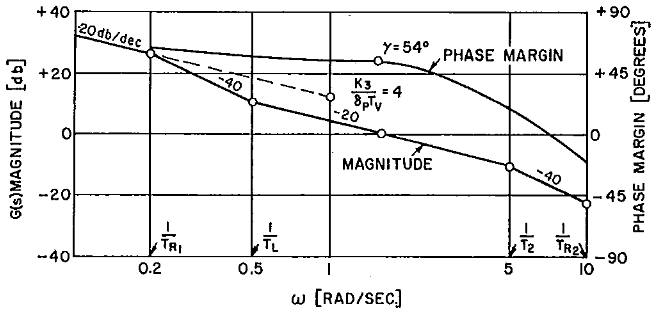

FIG. 7-41 BODE DIAGRAM OF INITIAL PRESSURE CONTROL WITH PROPORTIONAL PLUS PARTIAL INTEGRAL CONTROLLER

7.10 Interpretation Of Phase Margin

7.10.1 In order for a system to be stable the phase margin (γ_c) *must be larger than zero.*

7.10.2 With a given deadband the sustained oscillations of a control system get bigger the smaller the phase margin of the linear system is.

7.10.3 In a practical system a phase margin of $+40^\circ$ or more will usually hold sustained oscillations caused by deadband to an acceptably small magnitude.

7.10.4 On an integral control system (7.08) the sustained oscillations caused by deadband tend to be larger than on a proportional system (7.07) if the phase margin (γ_c) of both linear systems was the same.

7.11 Nomenclature For Section 7, Appendix

A = equivalent valve-flow area such that the flow $m = AP$, lb/sec

A_r = equivalent area for rated flow at rated pressure

ΔA = change in equivalent valve-flow area

C = controlled variable (general)

e = base of natural logarithm

E_1 = magnitude of oscillation of input (general)

E_2 = magnitude of output after nonlinearity (general)

$F_o(s)$ = open-loop transfer function in Laplace form (general)

G = gain of hydraulic integrator (units per sec per unit error)

$G(s)$ = open-loop system transfer function in Laplace form (general)

G_A = transfer function between servomotor stroke and equivalent valve area (units per unit)

G_h = transfer function between pressure regulator and servomotor stroke (units per unit)

G_m = transfer function between equivalent area and steam flow (units per unit)

G_R = transfer function of pressure regulator (units per unit)

G_V = transfer function of steam vessel in which the pressure is being controlled (units per unit)

$G(j\omega)$ = general frequency-dependent transfer function

H = feedback transfer function (general) (units per unit)

$H(s)$ = feedback transfer function in Laplace form (general)

$H(\alpha)$ = feedback transfer function being a function of magnitude of relative equivalent area (units per unit)

$j = \sqrt{-1}$

K = constant (general)

K_3 = incremental slope of equivalent area versus servomotor stroke referred to average slope

m = steam flow, lb per sec

m_r = rated steam flow, lb per sec

Δm = steam flow change, lb per sec

P = steam pressure, psi (a)

P_r = rated steam pressure, psi (a)

ΔP = steam-pressure change, psi

ΔP^* = pressure change causing a change of unity at the controller output, psi

R = reference (general)

R_r = rated pressure reference, psi (a)

ΔR = pressure-reference change, psi

s = Laplace variable, 1/sec

T = time constant (general), sec

T_L = lead time constant, sec

T_R = time constant of proportional regulator, sec

Y_{R_1} = first lag-break time constant of proportional plus partial integral pressure regulator, sec

T_{R_2} = second lag-break time constant of proportional plus partial intregal pressure regulator, sec

W_r = weight of steam in steam vessel at rated pressure, lb

Y = stroke of hydraulic relay or servomotor (general), in.

Y_r = rated stroke of hydraulic relay or servomotor (general), in.

ΔY = stroke change of hydraulic relay or servomotor (general), in.

$\alpha = \dfrac{\Delta A}{A_r}$ = relative equivalent area change dimensionless)

$\delta_P = \dfrac{\Delta P^*}{P_r}$ = steady-state pressure regulation of pressure controller (dimensionless)

$\delta_i = \dfrac{\Delta P_i^*}{P_r}$ = instantaneous regulation of pressure controller (dimensionless)

ϵ = relative error signal (dimensionless)

$\epsilon_P = \rho_P - \psi$ = relative pressure error (dimensionless)

ϵ_1 = signal applied to input of nonlinearity (general)

ϵ_2 = output signal of nonlinearity (general)

ϕ = phase angle of describing function or general

$\eta_1 = \dfrac{\Delta Y_1}{Y_{1r}}$ = relative position change of pressure-controller output (dimensionless)

$\eta_2 = \dfrac{\Delta Y_2}{Y_{2r}}$ = relative position change of servomotor output (dimensionless)

$\mu = \dfrac{\Delta m}{m_r}$ = relative flow change (dimensionless)

$\mu_i = \dfrac{\Delta m_i}{m_i}$ = relative flow change of steam supply to system (dimensionless)

$\mu_o = \dfrac{\Delta m_o}{m_r}$ = relative flow change out of system (dimensionless)

$\mu_H = \dfrac{\Delta m_H}{m_r}$ = relative feedback flow change (dimensionless)

$\psi = \dfrac{\Delta P}{P_r}$ = relative pressure change (dimensionless)

$\rho_P = \dfrac{\Delta R_P}{R_{P_r}}$ = relative pressure-reference change (dimensionless)

π = 3.14 radians

ω = frequency (general), radians per sec

7.12 Recommended Literature

"Servomechanisms and Regulating Systems Design," H. Chestnut and R. W. Mayor, John Wiley & Sons, Inc., New York 1963, Vol. 1 & 2 (second edition).

ACKNOWLEDGMENT

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