MECHANICAL DRIVE TURBINE CONTROL SYSTEMS FOR THE 1980'S

by

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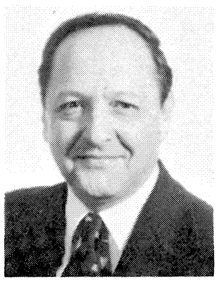

Anthony J. Rossi, a native of the State of Pennsylvania, graduated from Wayne State University, Detroit, Michigan, with a B.S.E.E. degree in Electrical Engineering in 1959.

He has been employed by the General Electric Company since 1961, and has had broad experience in the design of electronic control systems. These designs include excitation and protective systems for aircraft electrical power generating systems, speed and excitation controls for shipboard turbine generator sets, and speed/pressure control systems for mechanical drive turbines. His experience also includes a three-year assignment in turbine marketing.

Mr. Rossi is currently manager of Turbine Systems Development Engineering in the Mechanical Drive Turbine Department. In this capacity, he is responsible for development and design of turbine control, instrumentation, and protective systems.

He is a member of IEEE, and has been awarded several patents in the area of control system design.

ABSTRACT

The changes encountered in moving from mechanical to electrical control of mechanical drive steam turbines are reviewed with regard to both operation and performance. This paper examines past practices in control system design and application philosophies, and also presents new control concepts for consideration. Areas are pointed out where benefits to the user can be realized by taking advantage of the sophistication and versatility of state of the art electronic components to perform control functions formerly accomplished by mechanical means.

Both speed control and speed-extraction pressure control systems are evaluated and analyzed. Examples of the effects of changes in load and speed are given. Also, recommendations are given to ease the conversion to electrical control. These include servo actuator characteristics needed for a reliable and stable operating system.

INTRODUCTION

The performance and reliability of mechanical drive steam turbines is of paramount importance in industrial applications since these processes are, by their nature, sensitive to maintaining continuous flow. The turbine control system plays a significant role in assuring that long-term, trouble-free operation is realized.

Designs employing electronic components to perform control functions formerly provided by mechanical and pneumatic devices provide a rich opportunity for systems which are simpler, more versatile, with improved performance and reduced maintenance. Each of these are positive contributions to the reliability and availability of steam turbine drive trains.

The elimination of mechanical control devices makes possible a design virtually free of non-linearities resulting from friction, deadband and lost motion. As a result, more accurate mathematical models of the control system can be developed. The designer now can give sufficient recognition early in the design stage to the dynamic interactions between the speed and pressure governing subsystems on multi-variable turbines, such as automatic extraction machines, and between these subsystems, the turbine and the driven load. System response to routine or unexpected changes in the shaft horsepower or the process steam flow can be evaluated more accurately than formerly possible. It must be expected that sudden and violent energy unbalances can develop in the process; and the control system response, stability, and reliability are key considerations in preventing a catastrophic cascading effect on the process. The need to be concerned about abnormal energy unbalances is the primary reason why automatic control systems are a necessity.

The severe consequences resulting from unanticipated shutdown of the turbine mandate that serious consideration be given to providing means whereby the turbine is not tripped due to a malfunction in the control system. However, the attendant necessary for safe operation after an equipment malfunction cannot be neglected. A satisfactory compromise must be made. Design alternatives, including the use of redundancy, are presented for consideration.

It is the purpose of this paper to examine past practices in control system design and application philosophies and to present new concepts for consideration. The intent of this paper is to point out areas where benefits to the user can now be more readily realized by taking advantage of the sophistication and versatility of electronic components to perform control functions formerly accomplished by mechanical means. This includes the speed or pressure control subsystems as well as the overspeed protection systems. The gains which are expected to be realized lie in the area of increased turbine availability, reduced maintenance time, reduction of required operator skill level, superior performance and simplified interface with the user's application.

COMPARISON OF CONTROL SYSTEM DESIGNS

Historical Control Systems

The past five years have produced a rapid evolution in control system technology for industrial mechanical drive turbines. Applications where mechanical-hydraulic designs once predominated have rapidly swung over to electro-hydraulic designs. The motivation for this change in user preference arises from the need to solve inherent shortcomings in the mechanical hardware, mainly resulting from the complexity required to perform necessary functions by mechanical means. There has long been a need for control system designs which are not only simpler but which provide more flexibility and benefits than have previously been available.

The following descriptions of typical control systems will serve to both illustrate the hardware involved and highlight the basic differences between a mechanical-hydraulic and electro-hydraulic control system design.

Mechanical-Hydraulic Speed Governor

The basic speed control system that has been furnished with industrial turbines has consisted of the conventional mechanical-hydraulic (flyball) governor. A typical line diagram of this system is shown in Figure 1. The system uses a flyball governor driven by the turbine shaft through reduction gearing to provide a speed feedback signal. A speed setting bushing (SSB) along with the primary relay (PR) provide the necessary comparison function between desired speed and actual speed. They also act to generate the error signal required to properly position the control valves. The control valves themselves are driven by a powerful hydraulic operating cylinder which in turn is driven by the primary relay. The primary relay also provides the compensation to properly stablize the system.

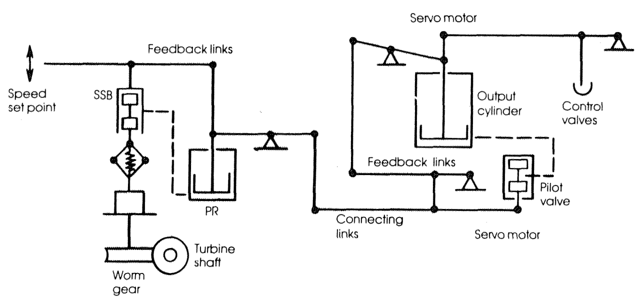

Figure 1. Mechanical-Hydraulic Governor for Speed Control.

A system of this design contains a considerable quantity of mechanical linkages, levers, pin joints and rod end bearings which are necessary for interconnection and position control of the various mechanical and hydraulic control elements. Mechanical connections are shown as solid lines, and hydraulic connections are shown by dotted lines in Figure 1. The principal disadvantages of a system of this type include: wear of mechanical gearing, bearings and pin joints; deadband; deflection of linkages; vibration; and frequent maintenance requirements.

In some of the more recent applications, the flyball governor has been replaced by an electric governor which accepts a speed feedback signal. This signal is obtained from a speed pickup used in conjunction with a toothed wheel mounted on the turbine shaft. A line diagram of this system is shown in Figure 2. The speed feedback signal is compared to a speed setpoint signal which is a preset voltage. Electronic circuitry is used for the speed comparison, error amplification, and stability compensation functions. An electric-to-hydraulic transducer (servo) accepts the valve positioning signal produced by the electronics. The output of the servo positions a pilot valve which in turn controls the flow of oil to the output cylinder. The position of the output cylinder determines the position of the control valves.

One of the main advantages of this change was the complete elimination of reduction gearing which was required for the mechanical-hydraulic control system. Some mechanical linkages have also been eliminated in the speed comparison and stability compensation functions. However, this system still retains a considerable amount of linkages, bearings, levers and pin joints with the attendant disadvantages as noted earlier.

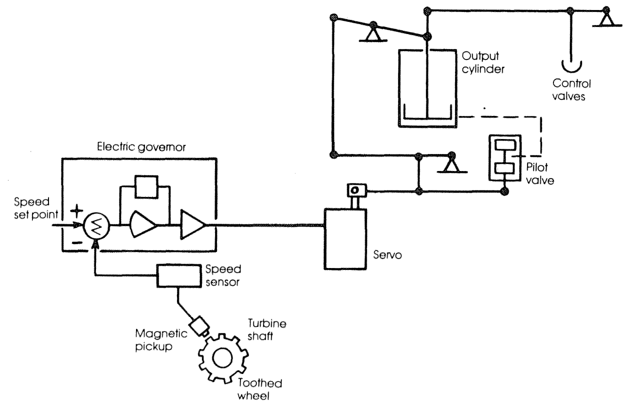

Figure 2. Electric-Mechanical-Hydraulic Governor for Speed Control.

Mechanical-Hydraulic Speed/Pressure Governor

Before proceeding into a description of the overall theory of the speed/pressure control system, it will be helpful to briefly discuss the type of turbine for which the control system is designed.

Figure 3 is a schematic diagram of a typical single automatic extraction turbine. This type of turbine utilizes two sets of control valves supplying steam to two sections arranged in tandem on the turbine shaft. Both valves are controlled simultaneously to govern turbine speed over a given load range and the extraction of steam pressure over a given extraction flow range. The turbine consists of two sections divided by an extraction valve gear which controls the extraction pressure by regulating the steam flow to the exhaust section. The control system sets the speed as directed by a speed reference and maintains the extraction pressure constant with respect to a pressure reference. Both control functions act simultaneously as follows:

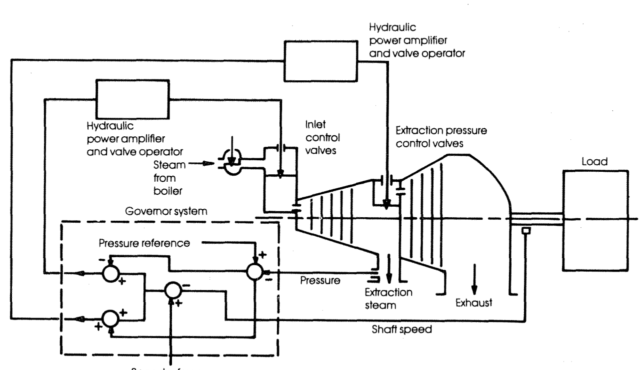

Figure 3. Automatic Extraction Turbine.

1. If the turbine speed increases with respect to the reference speed, the speed governor acts to close both the inlet and extraction valves to reduce the flow of steam through the turbine to decrease speed, but does not affect the extraction pressure.
2. If the extraction pressure is above the desired level, the pressure governor acts to close the inlet valves but open the extraction valves to decrease the extraction pressure while maintaining a constant speed.

As can be seen from this description, both control loops interact with each other.

The basic speed and extraction pressure control system that has been furnished with industrial automatic extraction turbines has consisted, again, of the conventional mechanical-hydraulic governor for speed control. However, the necessity to simultaneously control speed and extraction pressure results in a far more complicated system than described by Figure 1. A line diagram of this system is shown in Figure 4. The functions which have been added to Figure 1 to obtain the system represented by Figure 4 are enclosed in solid lines.

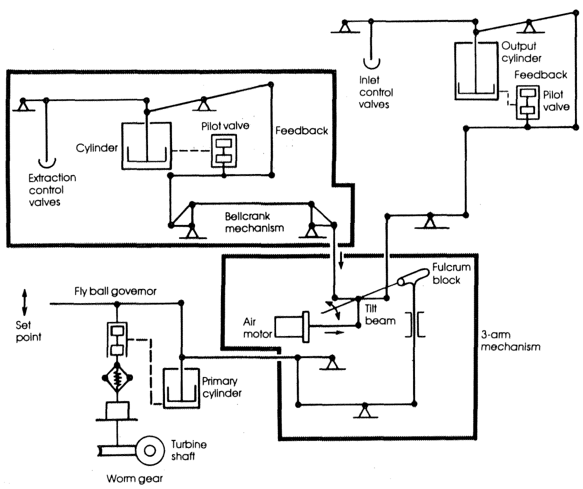

Figure 4. Speed/Extraction Control with Mechanical Governor.

One added function is the air motor, or equivalent device, required for positioning the 3-arm mechanism. The extraction pressure sensing, comparison and error generation functions generally are customer furnished and are not shown in Figure 4. Another added function is that the air motor acts to rotate a tilt beam around the axis, as shown, to allow the inlet and extraction valves to be moved in opposite directions in response to a change in extraction pressure.

A third change is that rotary motion of the tilt beam is transposed to a linear motion by means of the bell crank mechanism to position the pilot valve and output cylinder associated with the extraction control valves. The tilt beam is also mechanically connected to the pilot valve and the cylinder for the inlet valves.

Finally, a lever system has been added which causes both sets of control valves to move in the same direction for a change in speed. This is accomplished through the mechanical connection between the fulcrum block and the tilt beam.

As evidenced by Figure 4, a considerable amount of mechanical components has been introduced into the control system in going from a single variable system (speed control) to a multi-variable system (speed and extraction pressure control). The disadvantages of wear of joints and bearings, deadband, deflection of linkages and frequent maintenance requirements have been compounded compared to the system shown in Figure 1.

Electro-Hydraulic
Speed/Pressure Governor

The control systems depicted in Figures 1 and 4 can be segregated into three basic functions which are common to all control systems. These are as follows:

1. Parameter sensing and feedback functions (speed/pressure).
2. Logic/stability functions.
3. Power function (output cylinder).

The power function (output cylinder) provides the force required to position the steam control valves. The sensing, feedback and logic/stability functions, which make up the rest of the system, provide the control of the overall system. For good control system design, the overriding considerations in these latter areas are precision and rapid response in order to properly correct for disturbances in the controlled parameters (i.e., speed and pressure). High force or high work capabilities in these areas are not inherently demanded to perform the desired functions. There is a rich opportunity to simplify hardware, improve performance, reduce maintenance and increase overall system reliability and availability by replacing cumbersome mechanical devices with electronic components. The practicality of this proposal is evidenced by the electro-hydraulic design approach shown in Figure 5. Although the system of Figure 5 is drawn for a multi-variable control (speed and pressure), it will serve just as well for a single variable control such as speed control alone by deletion of the components associated with the pressure control system.

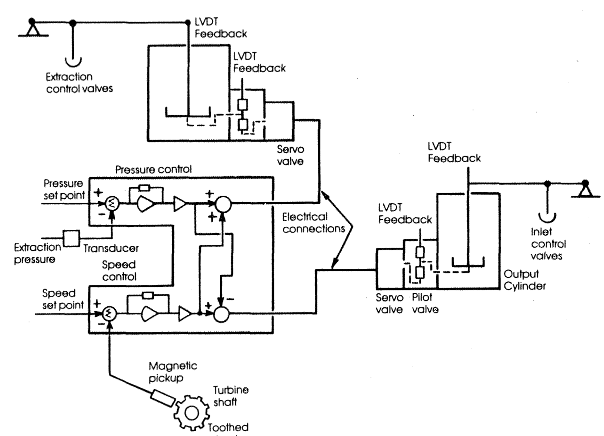

Figure 5. Electric-Hydraulic Control System for Speed/Extraction Control.

Hardware associated with the electro-hydraulic system shown in Figure 5 consists of an electronic package which houses all of the functions of the control with the exception of the turbine mounted hydraulic valve operator assemblies. The speed sensing function is provided by a simple turbine mounted magnetic pickup and toothed wheel. The pressure sensing function is provided by a common pressure to current transducer mounted on or near the turbine.

The system design shown in Figure 5 has replaced all of the mechanical logic/stability devices which appear in Figure 4 with straightforward electronic circuits. Mechanical devices which have been replaced include the primary cylinder, air motor, tilt beam, fulcrum block, bell crank, connecting linkages and the position feedback levers associated with both output cylinders. The worm gear, flyball governor and its associated speed setting bushing are replaced by a toothed wheel and a magnetic speed pickup.

Figure 6 shows a more detailed schematic design of the steam valve operator assemblies (servomotors). Each servomotor consists of an electric-to-hydraulic transducer (servo valve), a combined primary piston and pilot valve for hydraulic amplification, and the output cylinder.

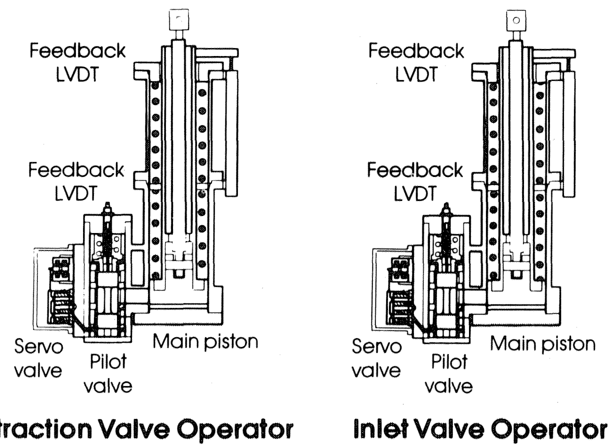

Figure 6. Typical Hydraulic Hardware for EHC Control System for Single Automatic Extraction Turbine.

The pilot valve and output cylinder shown in Figure 6 are essentially the same basic components and perform the same functions as the primary relay, pilot valve and output cylinder shown on the mechanical-hydraulic system of Figure 1. These components form part of the valve positioning loop. However, the design of Figure 6 has been considerably simplified by the complete elimination of the mechanical connecting links, rod end bearings, and mechanical feedback links required by the earlier systems. This is accomplished by designing the primary relay and the pilot valve as a single component and by replacing mechanical feedback linkages with a position sensing linear variable differential transformer (LVDT). The electrical output of the LVDT provides a position feedback signal to an electronic summing amplifier located in the electronic package to provide linear position control of the valve operators.

A simplified block diagram of the speed control loop is shown in Figure 7. The set speed can be controlled from either one of three stations. A signal converter accepts a variety of input signals and converts these to a standard signal compatible with the electronic circuits downstream. The speed reference thereby established is compared to a speed feedback signal obtained from redundant magnetic speed pickups. The resultant error signal passes through circuitry provided for system stability. This signal is combined with the electrical position feedback signals from the hydraulic power amplifier and control valve operator. The servo valve accepts this signal and completes the first stage of a hydraulically controlled subsystem which includes the hydraulic power amplifier and control valve operator. The control valve operator positions the

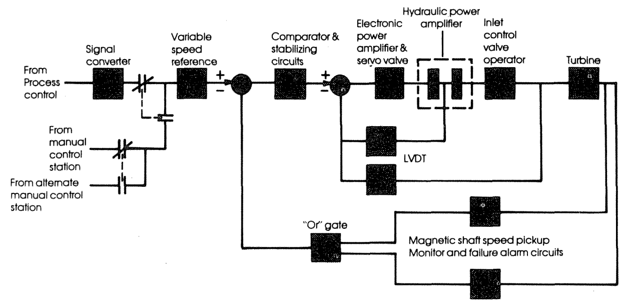

Figure 7. Speed Governing System.

inlet control valves to increase or decrease turbine speed as required.

The valve positioning portion of the control loop consists of the electronic power amplifier, the electric-to-hydraulic transducer (or servo valve), the hydraulic power amplifier, the ram which operates the steam valves, and electrical position feedback.

Figure 8 shows a simplified block diagram of the pressure control loop. The process control signal is accepted by the signal converter to provide a pressure reference as was done for analogous operations in the speed loop. A comparison is made with an electrical signal which is proportional to the extraction pressure. The resultant error signal drives the comparator, stabilizing circuitry and hydraulic elements. These position the extraction pressure control valves, the same as previously described for the speed control loop. Pressure sensing, pressure level setpoint, comparison function and error generation function are all included in this design to permit a more totally integrated design than has been available in earlier systems.

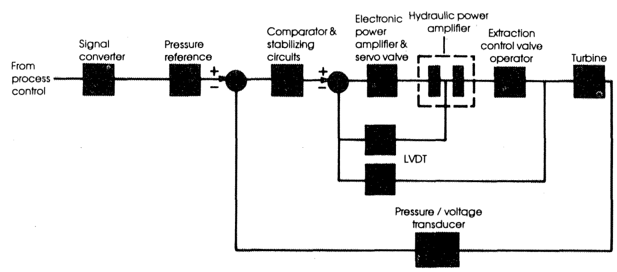

Figure 8. Pressure Governing System.

Figure 9 shows the interconnection between the speed and extraction pressure loops. The circuitry is designed to move both valves in the same direction for a change in speed with extraction pressure fixed. For a change in extraction pressure with speed fixed, the valves are moved in opposite directions.

In-depth consideration of the dynamic interaction between these two control paths is necessary to ensure that the system reacts to speed or extraction flow disturbances in a stable manner over the specified range of operating conditions.

Valve Positioning Loop — Performance Key

The design approach which uses electrical position feedback in the valve positioning loop instead of mechanical feed-

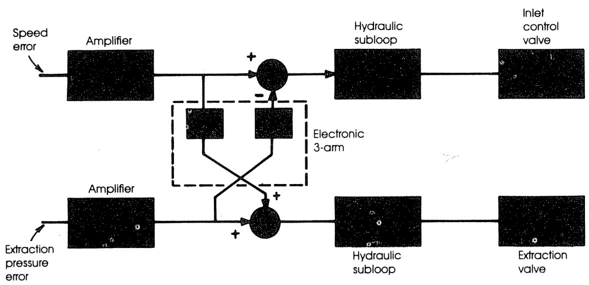

Figure 9. Speed and Extraction Pressure Subloop Interconnection.

back linkages permits a considerable simplification of hardware. More importantly, it is a major key in improving the performance and response of the entire control system. Although use of an electric governor instead of the mechanical flyball governor is a step forward in this regard, this is not as significant a factor as is total elimination of the mechanical components which normally are found in the valve positioning loop.

The reason that the valve positioning loop is critical to performance is that this portion of the total control loop inherently has the poorest frequency response and, as a result, represents a major constraint on overall system response and precision of control. This constraint is especially critical in designs which rely on mechanical means for position control of the valve operator. The associated linkages and levers contribute to control system inaccuracy and to reduced stability margin because of the effects of friction, deadband, and lever flexibility which are inherent with mechanical control devices. Elimination of these shortcomings not only improves performance, but also permits the designer to better predict the response of the system. A more optimum control system can now be realized, and the overall turbine response to load disturbances can be more accurately assessed and accounted for in the design stage.

EFFECT OF MECHANICAL CONTROL COMPONENTS ON SYSTEM STABILITY MARGIN AND PERFORMANCE

As has been pointed out earlier, mechanical control components are inherent sources of friction, deadband and flexibility, all of which act to degrade performance. Figure 10 shows a typical valve positioning loop for a design which retains such devices.

An accepted measure of the performance of a control system is to evaluate the response of the output to sinusoidal changes in the input. Parameters which are measured are the gain and phase relationships over a given frequency range. Phase shift, or the time lag between a change in the output in response to a change in the input signal, is of particular significance since it is critical to stable operation. Excessive phase shift (reduced phase margin) results in increased overshoots in the controlled parameters, and if excessive, will force the turbine into an oscillatory mode of operation.

Negative Effects of Friction, Deadband and Lever Flexibility

The curves labelled V/S in Figure 11 show phase shift comparisons between calculated data and test data taken on a mechanical feedback system similar to the design shown in Figure 10. The input is taken at point S and the output is measured at point V, as shown in Figure 10. The difference

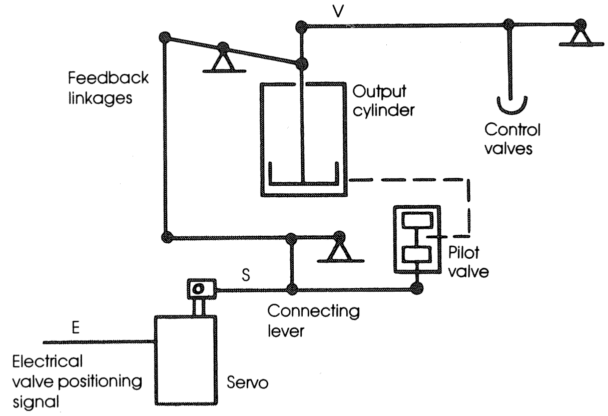

Figure 10. Typical Valve Positioning Loop with Mechanical Feedback Linkages and Levers.

between the calculated and test results is attributed to the effects of friction, deadband and lever flexibility. The net effect of the added phase shift is that the time constant for the output cylinder and pilot valve combination is 17 radians per second (phase shift = 45°), instead of the calculated value of 34 radians per second. The curve labeled V/E shows the total accumulated phase shift resulting from the addition of the electrical to mechanical servo in the valve positioning loop, as shown in Figure 10. As a result, the system response to upsets will be slower than calculated. Similar comparisons made on the electro-hydraulic design which eliminates all mechanical control components show virtually no difference between the calculated and test data.

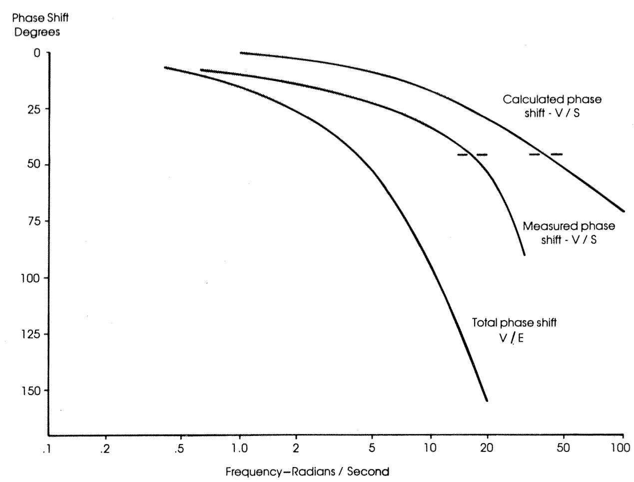

Figure 11. Phase Shift Error Inherent in Mechanical Feedback Systems.

Figure 12 shows a comparison of phase shifts for the valve positioning loop designs employing electronic position feedback and mechanical position feedback. This data shows the considerable improvement in frequency response of the valve positioning loop made possible by complete elimination of mechanical links and levers. The following discussion shows how this improvement reflects itself on overall control system stability and performance.

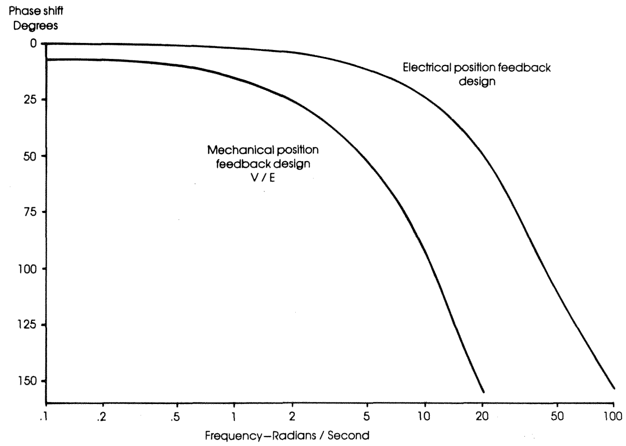

Figure 12. Valve Position Loop Phase Shift Comparison Showing Reduced Phase Shift (Time Lag) with Electrical Position Feedback.

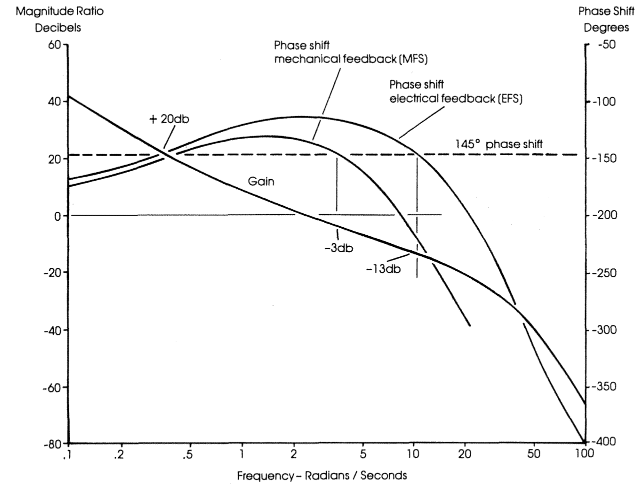

Figure 13. Stability Margin Comparison of Mechanical Feedback (MFS) and Electrical Feedback (EFS) Systems — Open Loop Mode Plots.

Comparison of System Stability Margin

Once the frequency response of the valve positioning loop has been determined for both designs, it is possible to assess the effect these characteristics will have on the total system composed of the control system and the turbine. The data of Figure 13 shows the calculated open loop frequency response of the overall turbine and control system to small signal changes for both types of valve positioning loops. The curve labelled "Phase Shift-Electrical Feedback" shows the total system phase shift for a design which is free of mechanical control components in the valve positioning loop. The curve labelled "Phase Shift-Mechanical Feedback" shows the total system phase shift for a design which employs a valve positioning loop as depicted in Figure 10. Both turbine control systems are identical in every other respect. The curve labelled "gain" is common to both designs. The information in Figure 13 can be summarized as follows:

1. The EHC (electro-hydraulic control) design has a gain band width (phase shift is less than 145°) of 33dB (45 to 1 gain change tolerance). The system which retains mechanical feedback has a reduced gain band width of 23dB (14 to 1 gain change tolerance). The EHC design provides much more latitude for system gain changes which naturally occur from differing operating modes, steam conditions and equipment condition over a period of time.
2. The system crossover frequency (frequency at which the gain curve crosses the zero magnitude ratio point) is a measure of the response of the turbine to changes in operating conditions. This point is shown at approximately 2.5 radians per second. The system responds more quickly as the crossover is increased provided that the phase shift is not excessive. The crossover frequency can be increased by increasing the system gain. A standard criteria recommended for selection of the crossover frequency is that a gain change tolerance of 6dB (gain change by a factor of 2) should not result in a total phase shift of more than 145°. The crossover frequency of the EHC design can therefore be increased to approximately 5 radians per second, while that of the system using mechanical feedback should be reduced to approximately 1.5 radians per second. The resultant system gain will be 3.33 times greater in the EHC design. This is a significant difference which will reflect itself as lower offspeeds and shorter recovery times for correction of system disturbances. The following comparisons of the transient response for a turbine/compressor train to upsets in shaft horsepower and extraction steam flow illustrate the performance to be expected of both design approaches.

Control System Response To Load Changes

The preceding discussions show how complete elimination of mechanical control devices in the valve positioning loops permit higher gain systems as well as increased stability margin. This results in improved system recovery from energy unbalances which can occur in the process.

The contrast between the performance of both types of valve positioning loops is illustrated by the following calculated transient responses to system load changes. The turbine design, which is used as an example, is an automatic extraction machine rated at 31,000 horsepower at 4670 RPM. The rated extraction flow is 300,000 lb/hr at a pressure of 474 psia. The combined inertia of the turbine and driven load is 15,300 lb-ft^2. The combined rotor time constant at rated speed is 6.67 seconds. Both control systems are identical except for the retention of mechanical devices in the valve positioning loop for the response curves labeled "Mechanical Feedback System."

Figure 14 shows the calculated speed overshoot and speed recovery time when the shaft line is suddenly and completely reduced from rated load to zero load as specified by API 612. The maximum allowed speed overshoot for this condition is 1.07 times the rated speed (4997 RPM). The improvement in peak overspeed (1.03 vs. 1.087 × rated) is evident. Recovery back to the rated speed (4670 RPM) occurs in 23 seconds for the system with electrical feedback, whereas the time for the mechanical feedback system extrapolates to 63 seconds. The considerable improvements shown by Figure 14 result directly from the higher system gain which is permitted by the total elimination of mechanical control devices.

Figure 15 shows comparative speed and pressure transients for a case where the shaft load is initially reduced from full load to 50% load. After the turbine speed has recovered,

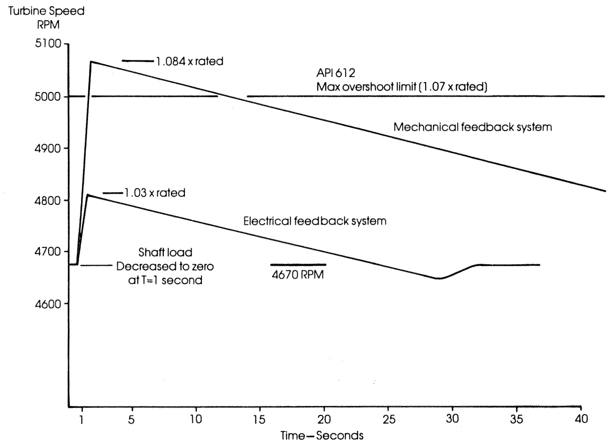

Figure 14. Comparative Response for Sudden Loss of Rated Load.

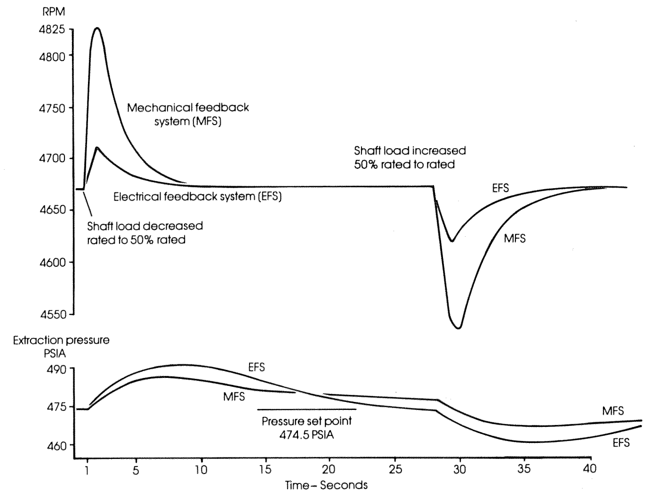

Figure 15. Comparative Response For Removal and Re-Application Of 50% Rated Shaft HP.

the turbine shaft load is stepped from 50% of rated load back to full rated load. The extraction flow requirement is maintained at 300,000 lbs/hr during the shaft horsepower changes. The results of Figure 15, as summarized in Table 1, show a dramatic difference in turbine offspeed and speed recovery time in favor of the electrical feedback system design. Extraction pressure transients show very little difference except in recovery time. The system with electrical feedback recovers to the pressure setpoint of 474 psia in 22 seconds after the decrease in shaft horsepower. The system with mechanical feedback devices has not yet recovered at the time the shaft load is reapplied (T = 28 seconds). This can be attributed to the lower gain of this design which results in long recovery times.

Calculations for a reduction of the extraction flow from 300,000 lbs/hr to 200,000 lbs/hr at the rated shaft horsepower are illustrated in Figure 16. These show negligible differences in the extraction pressure transient overshoot and recovery to the setpoint pressure. Recovery to the setpoint of 474 psia occurs in approximately 50 seconds. Transient speed overshoots and recovery time to rated RPM are slightly larger for the system with mechanical feedback (MFS). The slow response of the pressure control loop to changes in extraction steam flow requirements results from the large volume of external piping (1300 cubic feet) which was assumed.

Summary of Performance Improvements

The preceding discussions have dealt with the effect of

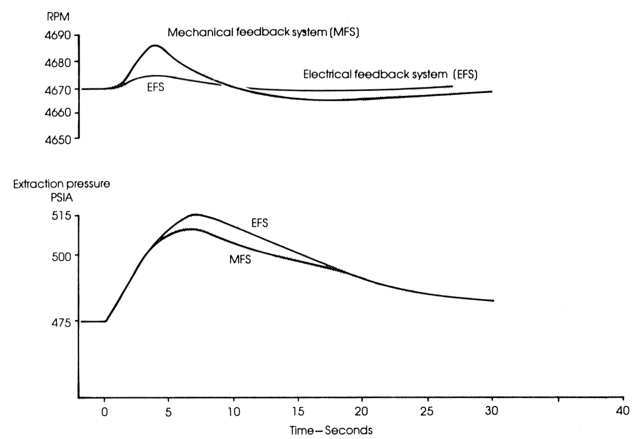

Figure 16. Comparative Response for Step Change in Extraction Steam Flow from 300,000 Lb/Hr to 200,000 Lb/Hr.

mechanical control components on system stability margin and performance. Figure 11 shows the large phase shift (time lag) existing in designs which rely on mechanical components in the valve positioning loop. This added time lag is due to the un-

TABLE 1. SUMMARY OF PERFORMANCE SHOWN ON FIGURE 15.

	Mechanical Feedback Design (MFS)	Electrical Feedback Design (EFS)	Ratio
%Speed Overshoot	3.32	0.86	3.86
% Speed Undershoot	2.91	1.07	2.72
Speed Recovery Time (Seconds) to ±0.1% Band			
Load Removal	9	5	1.8
Load Application	10	6	1.67
Peak Extraction Pressure (psia)	486	492	0.986
Minimum Extraction Pressure (psia)	467	461	1.013

predictable effects of friction, lost motion, and lever flexibility, all of which are inherent in mechanical control devices. Figure 12 shows the reduction in phase shift which results from use of electrical feedback instead of mechanical feedback designs. Figure 13 demonstrates how the improved phase shift permits the control system to be designed with higher gains at no sacrifice in the system stability margin. Figures 14 through 16 show the extent of the resultant system performance improvement for various disturbances in shaft horsepower or extraction steam flow.

An added benefit is that elimination of nonlinearities in the control system permits a more accurate mathematical simulation and allows the dynamic performance and stability characteristics to be evaluated early in the design stage instead of in the field during initial startup.

OTHER BENEFITS

The comparisons made thus far have pointed out hardware simplification, maintenance reductions and performance advantages offered by the electro-hydraulic design concept. The following improvements in turbine start-up and operation can also be realized.

The control system can be designed to permit remote start-up and operation of the turbine from a central process control area. The remote control panel should include all necessary controls for speed and pressure setpoint adjustments, extraction or admission steam loading adjustment, overspeed and overpressure test, and turbine trip/reset. A simplified control panel to allow speed control or turbine trip should be available for use at the turbine if so desired.

The speed control system can be designed with the capability to start the turbine from standstill. This eliminates the need to have an operator manually throttle steam at the turbine by means of a trip throttle valve until the minimum control range of the speed governor is reached.

The capability for start-up from zero speed under control of the speed governor and the turbine steam control valves permits the use of stop valves in place of manually controlled trip throttle valves in the steam lines. Advantages to the user include:

1. *Improved Hardware*
   a. Handwheel, lead screw, and mechanical latches are replaced by a hydraulic servomotor — hydraulic power replaces manual labor for easier valve cracking.
   b. Positive valve stem backseating eliminates stem steam leakage.
2. *Improved Reliability*
   a. Positive spring loaded closing improves valve closing times by a factor of at least 2 to 1 as compared to trip throttle valves. As a result, speed overshoots on trip-out will be less.
   b. Positive valve backseating protects against steam carryover deposits in the close clearances between the valve stem and its bushing. Fouling in this area is one of the main reasons why throttle valves fail to close when required.

Electronic speed control systems which employ a simple toothed wheel on the turbine shaft eliminate reduction gearing, shaft extensions and mechanical governors, all of which result in larger rotor overhang and increased rotor sensitivity to unbalances.

RELIABILITY CONSIDERATIONS

The overall reliability and availability of the steam turbine is significantly influenced by the overall reliability and availability of the control system design. Historic data on field forced outages for mechanical drive turbines shows that upwards of 25% of field incidents are attributed to the control system. The criticality of maintaining continuous flow in industrial processes cannot be denied, and it is mandatory that measures be taken to insure that the control system be designed for dependable service.

High standards in this regard should not be limited solely to the electronic components in the system, but need to apply equally as well to hydraulic driven components, system wiring and installation. The following discussions present proposed means for improving control system reliability in this regard.

Redundancy

Consideration of redundancy as a measure to avoid system shut down in the event of a failure is not new. However, the extent to which redundancy can be successfully applied is often times misunderstood. Widespread use of redundant electronic components, or even of entire circuit boards, is more easily spoken of than accomplished in a practical manner.

For example, individual electronic components can fail in either a short circuit or an open circuit mode. Therefore, parallel as well as series duplications would have to be considered if redundancy is to be effective. While this can be implemented in the case of passive devices such as resistors or diodes, redundant amplifiers or other similar active solid state devices present an almost impractical problem since dynamic characteristics would have to be exactly matched for proper operation.

Attempts to make entire circuit boards redundant will result in similar problems. In addition, consideration would also have to be given to failure modes which result in partial degradation rather than complete short circuits or open circuits. It is highly unlikely that redundancy alone can be relied upon to satisfactorily handle partial failures.

The impracticality of widespread component duplication does not necessarily result in an unreliable design. There are viable alternatives. Along with conservative design practices, a production program which incorporates component reliability testing can be of equal or more benefit. Such a test program should include the individual testing of each component for initial screening of "out of spec" devices, temperature cycling of electronic components and assembled printed circuit boards to bring out mechanical weaknesses, and powered operation of all circuitry for extended periods of time prior to shipment to weed out components susceptible to early failure.

Redundancy is practical in a number of areas. These include speed and pressure sensing devices, internal power supplies and electronic overspeed sensing channels. In addition, uninterruptable power supplies based on a DC battery backup are becoming as standard as a backup source for the external 120 volt, 60 Hz power source normally provided by the customer for the control system and other electrical equipment on the turbine. Due to the relatively low power requirements of an electronic control, the uninterruptable power supply can be a compact package that readily lends itself to location close to the electronic package to further reduce the possibility of loss of power from faults on incoming power wiring.

Redundant components or systems are of little value if failures cannot be clearly identified and if on-line replacement cannot be accomplished without the need to first trip the turbine. Therefore, in those areas where redundancy is applied, on-line test, automatic monitoring, and failure alarm systems should be included in order that the user can be given sufficient notice to plan corrective actions.

System Wiring

Interconnecting wiring between the turbine, the electronic package, and the remote control station is as critical to reliable performance as any other component in the system. Sensitive electronic circuitry can malfunction as the result of electrical noise emitted by other systems or equipment being capacitively or magnetically coupled into the control signal wiring. In order to protect critical circuitry, the control designer should clearly identify grounding system connections, wire shielding and wire separation requirements. Shielding specifications should include allowable shield impedances and shield termination means. All wiring should have the wire size and maximum allowable impedance specified in order to limit voltage drops to acceptable levels.

Shielding and wire separation will not protect the control system circuitry against transient voltage swings which commonly exist on primary power input lines such as the 120 VAC or DC power furnished by the user. Such transients are present for a variety of reasons which include energizing or deenergizing of magnetic loads (transformers), switching of inductive loads (motors, relay coils, solenoids), lightning strikes, power line faults, etc. The resultant voltage surges may have peak values ranging from several hundred to several thousand volts and oftentimes are oscillatory in nature at a very high frequency. These are best protected against by adding surge suppressors to critical wiring in the electronic package itself. The intent is to short-circuit the transient voltages before they are conducted into critical electronic circuitry.

Although the exact characteristics of voltage surges cannot be predicted, it is suggested that American National Standards Publication, ANSI C37.90a-1974, "Guide for Surge Withstand Capability (SWC) Tests," be used as a specification of voltage surge suppression requirements. This standard defines the frequency of the oscillatory transient (1.0 MHz to 1.5 MHz), peak value (2500 to 3000 volts), decay time envelope, repetition rate, and a source impedance in order that energy dissipation requirements can be calculated.

Failure to do the proper engineering in the vital area of system wiring will invariably result in electrical noise problems which will require a significant amount of time, patience and expense to correct in the field.

Environmental Considerations

Although electronic circuits can be designed for continuous operation at elevated ambient temperatures such as 140°F or higher if military specification devices are used, reliability will be significantly improved if they are installed in areas with less severe temperatures. Installation at a site remote from the turbine is readily accomplished with electro-hydraulic control systems. This permits the electronic control package to be installed in a more hospitable and oftentimes a more convenient area. Design for hazardous atmospheres is not necessary and the packaging thereby becomes less complex and more accessible in the event that a printed circuit board changeout or some other maintenance action becomes necessary. In order to permit the user the maximum flexibility in location of the electronic package, the design should accommodate a distance of at least 1500 feet.

Electrical devices mounted on the turbine itself should be designed for the prevalent atmosphere as required. It is also recommended that wiring for turbine mounted components be routed through rigid conduit for protection and terminated in a junction box so that user connections can be conveniently made.

HYDRAULIC SERVOMOTOR DESIGN FOR RELIABILITY

The hydraulic powered servomotor is critical to both the performance and safety of the turbine. In addition to providing the work capability required for steam valve cracking forces and steam valve positioning, it must act as the first line of defense in cutting off the steam flow to protect against turbine overspeed. Work capability of up to 15,000 ft-lbs is commonly required.

A servomotor basically consists of a main piston sized to develop the necessary force capability with a given hydraulic pressure, a pilot valve (hydraulic amplifier) which controls the flow of oil underneath the main piston, and a preliminary stage which acts to position the pilot valve.

On EHC systems employing all electric feedback in the valve positioning loop, the preliminary stage consists of an electric-to-hydraulic transducer commonly referred to as a servo valve. A typical servomotor of this type is shown in Figure 6.

On systems which require mechanical feedback linkages in the valve positioning loop, the preliminary stage is generally an electric-to-position transducer (servo) which includes its own hydraulic amplifier consisting of a piston and a pilot valve. The output of this preliminary stage is mechanically linked to a second pilot valve which controls oil flow to the main piston. A typical design is shown in Figure 10.

Considerations which have primary impact on the design of the hydraulic servomotor are as follows:

1. The required work capability of the main piston is determined basically by steam cracking forces, valve gear friction, spring loads, and minimum supply oil pressure (usually 0.75 to 0.80 of nominal).
2. Oil from beneath the main piston is dumped through the pilot valve and the oil from beneath the pilot valve piston is dumped through the servo valve. Therefore, the primary influence on the sizes of both of these components is determined by the required oil dump capacity needed for the control system.
3. The oil dump capacity determines the maximum closing velocity of the main piston and, thus, the turbine speed overshoot when shaft horsepower is suddenly decreased. The oil dump capacity necessary to safely limit the turbine speed overshoot is a direct function of the main piston size and stroke and is an inverse function of the rotor time constant. An increase in piston area or stroke or a decrease in rotor time constant increases the required oil dump capacity of the system.

An additional important design constraint is the minimization of added time lags or phase shifts in the system. Both act to detract from performance and reliability. Additional phase shift introduced by mechanical control components and the resultant negative effect on system performance has been demonstrated by the performance comparisons shown in Figures 14 through 16. Excessive friction resulting from the binding of linkages or from tight clearances in low force devices such as pilot valves results in significantly increased phase shifts and accelerated negative impact on performance and reliability. Although close clearances in hydraulic devices act to minimize oil leakage, a penalty must be paid in the form of increased risk of malfunction. Generous clearances will result in increased leakage flow; however, because the higher leakage flow tends to provide a continuous flushing action, tolerance to oil contaminants such as particle matter, water and air is significantly improved. Larger clearances also eliminate the need for constant rotation of the pilot valve by an oil or electric motor in order to reduce high friction coefficients which otherwise would result in malfunction of the servomotor.

Single Acting Versus Double Acting Servomotors

For mechanical drive turbine applications, single acting pistons which require hydraulic pressure to open and are spring loaded to close are preferable to double acting designs which rely on hydraulic pressure for both opening and closing. The following are advantages of the single acting design with respect to safety and reduced hydraulic power supply requirements.

In the event that the hydraulic supply to the servomotor is lost, the control system is no longer effective as a speed or pressure control means and the control valves should close to avoid possible machinery damage. This is readily accomplished by the spring in the single acting design. In the case of the double acting design, the control valves may close, or they may open, depending upon the differential pressure across the servomotor cylinder, steam forces acting to close the valves and steam blowout forces which act to open the valves. While it is true that loss of hydraulic pressure will also act to close the trip throttle valve in any event, failure to close the control valves results in loss of the primary defense against overspeeding of the turbine.

The negative effect of double acting servomotor designs on hydraulic power requirements is readily assessed by evaluating the oil flow requirements in the opening and closing direction. Main piston velocity in the opening direction is determined by the frequency response required of the system. Typical opening velocities may range from 3-6 inches per second. As noted earlier, the required closing velocity is determined by the needed work capability of the servomotor and the rotor time constant. Table 2 shows a comparison of the required oil flow capability on the hydraulic power supply for both designs for a 10-inch diameter piston to stroke 8 inches. The servomotor has been designed for a 200 lb oil system and has a maximum closing velocity intended to meet the API 612 limits of overspeed of 1.07 times the maximum speed for the sudden loss of full load for a 6-second rotor time constant.

TABLE 2

	Required Hydraulic Capacity	
	Double Acting Servomotor	Single Acting Servomotor
Opening Velocity — 5 inches/second	102 GPM	102 GPM
Closing Velocity — 35 inches/second	714 GPM	0 GPM

The oil flow necessary to achieve the required closing velocity for the double acting servomotor represents an unacceptable penalty on the design of the oil system. Accumulators could be used to ease the demand on the oil pumps; however, the added costs for hardware and maintenance as well as the possible reduced system reliability need to be assessed.

EMERGENCY CONTROL PROVISIONS

Although the service history of electronic controls are indicating improved reliability and availability over their mechanical counterparts, the penalties associated with unanticipated shutdowns warrant consideration of fail-safe and emergency control provisions as a contingency in the event of failures. This is a case where the concerns of rotating equipment protection and the necessity of maintaining an uninterrupted process flow can be in conflict. The following discussion points out a number of areas where fail-safe or backup control provisions can serve to avoid turbine shutdown.

Fail-Safe and Backup Provisions for External System Failures

In most applications, the speed setpoint for the turbine is automatically generated as a function of the state of process variables. As shown in Figure 7, the signal path for the automatic mode of operation is from the process control to the signal converter to the variable speed reference. Fail-safe and backup control provisions for failure in the automatic process control function should include the following:

1. Maximum and minimum limits in the variable speed reference to limit speed between rated or some fraction of rated speed, depending on the direction of the process control failure. Such limits will also protect against trip or shutdown of the turbine for similar failures in the signal converter circuitry.
2. Memory circuitry which can retain the last valid speed setpoint generated by the process control and act to freeze turbine speed when it is determined that the process control signal is faulty.
3. Manual control stations which can be activated to place the turbine under the control of an operator in the event of the failure of the process control system or of the signal converter circuitry.

Fail-Safe and Backup Provisions for Control System Failures

Complete protection against turbine trip due to failure of any single component in the control system requires that the system be totally self-monitoring, have the capability to differentiate between a component failure and normal fluctuations in component voltages or currents, and be capable of automatic switch over to an independent standby control in the event of a true failure. Systems designed to this high level of sophistication are not presently available. Fail-safe features and backup provisions against control system failures, which can be implemented with present-day technology, include the following:

1. Redundancy provides protection for loss of one of the speed sensing pickups. If both pickups fail, the steam control valves would normally go to full lift and cause the turbine to overspeed. A fail-safe design should recognize a critical failure of this type and close the control valves without forcing the turbine to trip on overspeed.
2. A manual valve positioning system for position control of the hydraulic servomotor should be available which bypasses all circuitry except the minimum number of circuit boards required for valve positioning or safety. The control should be readily available to the operator in the control room and should be capable of rapid and precise positioning. Transfer from normal operation of

the manual valve positioning system and vice versa should be possible without prior turbine shutdown or excessive shock to the system. When operating in the manual valve positioning mode, all circuit boards not associated with the positioning system or with safety should be capable of being replaced without de-energizing the control.

Servomotor Stops

The fail-safe and backup provisions discussed thus far can be implemented with minimum risk to turbine safety. The manual servomotor positioning system does require a minimum number of circuit boards to be operative as described, and the turbine may experience a trip before the positioning system can be activated. In order to avoid any possibility of a turbine shutdown due to the closing of the steam control valves, the main piston of the valve positioning servomotor would have to be mechanically or hydraulically restrained at the desired valve lift. This is not recommended for the following reasons:

The net result of introducing a restraint which prevents the steam control valves from closing is that the primary means of protection against an overspeed has been disabled. A subsequent energy unbalance on the system, such as a reduction in shaft horsepower or extraction steam flow will result in an accelerating torque. This may drive the turbine to an overspeed condition with reliance on the overspeed sensing system and trip throttle valve as the sole line of protection. Figure 17 shows what can be expected of turbine speed acceleration for loss of 50% of rated horsepower with both servomotors restrained to full load lift. The calculation has been made for a turbine/compressor train with a combined inertia of 15,000 lb-ft^2. The combined rotor time constant is 6.5 seconds.

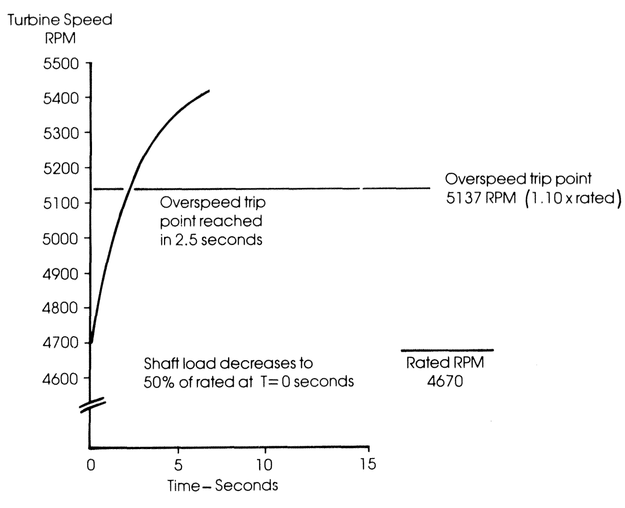

Figure 17. Speed Rise for Loss of 50% Rated Load with Servomotors Physically Stopped at Rated Load Lift.

Because the inlet and extraction valves have been prevented from closing when load loss occurred, the turbine rapidly accelerates to trip speed (5137 RPM) in 2.5 seconds and will trip out. It is questionable whether an operator can react quickly enough to prevent this end result. As shown by Figure 15, the peak speed for this type of load change is limited to 4710 RPM if the servomotors are not blocked open.

Field Maintenance Provisions for Improved Availability

Electronic components do not wear out or deteriorate over a period of time in the same sense that mechanical devices do; however, changes can occur in component characteristics as the result of temperature changes, supply voltage variations and aging. The effect of these changes can be readily assessed and accounted for in the design stage so that periodic readjustment or replacement of circuit boards in the field need not be a concern. Protective epoxy coating of circuit boards should be standard to avoid circuit malfunctions or changeout needs as the result of accumulation of moisture, dust or other airborne contaminants.

There is a need for the user to have procedures established for system checkout and adjustment during initial installation or for troubleshooting a circuit malfunction. Procedures should be detailed and easy to follow. Simplified schematic drawings and logic diagrams should be included complete with voltages and gains to be measured for calibration or troubleshooting. All necessary test and adjustment points should be located on the printed circuit card edge for ready access by plant instrumentation personnel. Troubleshooting procedures should be directed at locating problems at the plug-in printed circuit card level. Spare cards will permit the system to be restored to service as quickly as possible.

In those cases where redundant components have been provided, changeouts should be possible while the turbine is on-line. Replacement of internal power supplies should not require de-energizing of the entire control system. If speed pickups are to be changed out on-line, turbine hardware should be designed for ready access. Proper gapping of the replacement pickup to the speed pickup wheel should be possible before installation in the turbine for safety reasons and performance considerations.

CONCLUSION

The control system design concepts described by this paper represent a significant advancement over control systems previously furnished for industrial mechanical drive turbines. The successful development of electronic circuitry to replace the control system logic functions formerly provided by mechanical devices results in several improvements:

The control system can be designed with minimum deadband, reduced time (phase) lag and considerably less maintenance requirements. Performance of the system is greatly improved and is more predictable. This permits the stability of the turbine recovery from unexpected energy unbalances to be evaluated early during the design stage rather than in the field. The major key to these benefits is elimination of all mechanical control devices in the valve positioning loops.

Complete remote start-up and control capability eliminates manpower requirements at the turbine formerly needed for turbine start-up, normal control adjustments or control system monitoring. Design of the speed governor for start-up of the turbine from zero speed permits the use of stop valves in place of trip throttle valves for improved hardware and reduced maintenance in this critical area.

The overall reliability and availability of the steam turbine can be enhanced by use of redundancy in the system, reliability testing of electronic components, and the use of test and monitoring systems which provide the operator with ready and convenient information regarding the status of the control system.

Guidelines have been laid down for providing means whereby the control system can be made more immune to electrical noise emanating from external systems for a variety of reasons.

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