POTENTIAL PROBLEMS AND DESIGN PROPOSALS FOR TURBINE/ COMPRESSOR CONTROL SYSTEMS

Here some common and probable difficulties in controlling turbocompressors in processing services are discussed, along with one logical control figuration.

J.O. Hougen
University of Texas
Austin, Tex.

Greater use of steam driven turbines in the process industry has led to renewed interest in the control of the ensembles of which they are a part. Need to control both speed and load and to avoid undesirable operation—such as surging—in the face of varying process demands, are probable reasons for a restudy of the turbine control problem.

When sufficient electrical power is not available for driving turbo-compressors, blowers, and pumps, and especially where an adequate supply of steam can be generated by the processing plant, the use of steam driven prime movers may be attractive. When steam can be used in the process after being used to drive compressors there may be added advantages in using turbines.

Even if adequate electrical power is available turbines may be preferable to electrical drivers. Gear units can usually be eliminated and high overload capacity permits a wide working range for the compressor which is frequently demanded in practice. Compressor discharge pressures and capacity can be varied widely by varying speed.

Scope of subject

It is not the purpose of this article to present a comprehensive treatment of turbine-compressor control in all its ramifications. Much of a general nature can be found in existing literature. A recent publication (1) is devoted to the problems of indirect speed control. Standard arrangements of turbine-compressor ensembles and technical data are given in manufacturers' literature (2, 3, 4). Several special studies of turbine control for electrical generating purposes have appeared in recent articles, (5, 6, 7).

Instead the objective is to consider the requirements of an arrangement typical of turbine-compressor installations found in the process industry. Common and probable difficulties will be described and plausible ways to circumvent operating problems discussed. One logical control configuration will be presented and the means of procuring the necessary data for design described. Procedures will be outlined by which a satisfactory control system can be designed.

One popular processing scheme involving a turbine-compressor system is shown in Figure 1. Tandem turbines are used to produce the power required by the compressor. The first turbine utilizes high pressure superheated steam and is operated somewhat like a back-pressure turbine. That is, the effluent steam is discharged to a distribution system at an intermediate pressure level.

The second turbine, on the same shaft as the first, is operated in a condensing fashion, taking steam from the intermediate pressure source. A variety of other users derive steam from this latter source.

High pressure steam is generated both by a direct fired boiler and by waste heat boilers, using energy from exothermic chemical reactions taking place in the process.

Operating requirements

One objective is to optimize use of the process-generated steam so that a minimum of fuel is required, deriving a maximum of energy for compression and of steam for other uses from the process. Ideally all steam would pass through the high pressure turbine and be just sufficient to drive the low pressure turbine and satisfy other users with steam at the desired intermediate pressure. At the same time the power demanded by the compressor must be supplied. This requires a given compressor to operate at a given speed in order to develop the necessary process pressure in a given installation.

To design a system to meet all these requirements simultaneouly would appear to demand a unique combination of circumstances. Quantitative design is made especially difficult because performance characteristics of few of the components are known accurately prior to construction of the plant.

Among the uncertainties are the amount of steam likely to be generated from the process, the ultimate extent of superheat, the exact requirements of all the users, and the power demanded by the compressor. The latter is difficult to estimate because the process stream composition and ratio of recycle to fresh feed may not be known precisely. It would be fortuitous, indeed, if design and practice coincided even at one given compressor load.

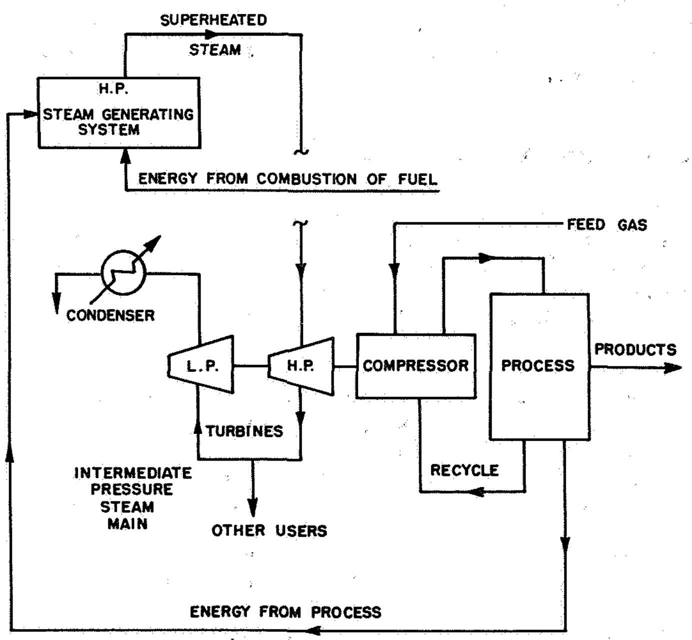

Figure 1. Schematic of turbine-compressor system associated with a process.

A satisfactory static design is only part of the requirement. Control of transient behavior and compensation for disturbances must also be achieved. The design of the control system demands a knowledge of the performance characteristics, both static and dynamic, of all components involved. A control system cannot be *designed* unless the control problem is quantitatively defined.

Types of speed controllers

Control of a turbine-compressor ensemble is usually achieved indirectly by controlling speed. In some manner a measure of the power required is obtained. This signal is used to alter the setting of the speed controller, and turbine speed changes until the desired power is developed.

Two basic types of speed governing are in use, mechanical-hydraulic and electro-hydraulic. The first utilizes the principle of the fly-ball governor to sense speed, the inertial forces produced by fly-weights positioning a valve which, in turn, regulates the pressure of hydraulic oil in the control system. Alternately the discharge pressure of a special pump driven by the main shaft is used as a measure of speed.

The second uses the voltage signal generated by a tachometer as a measure of speed. This is amplified and ultimately positions a pilot valve which regulates the oil pressure in the control system.

In each case the final control elements are hydraulic servo motors which develop sufficient force to position the turbine steam throttle valves.

Overspeed protection is always provided by incorporating means to close separate stop valves. In addition, in the case of tandem turbines, an over-ride feature is used by which the control valve on the high pressure turbine is throttled if the controller associated with the low pressure turbine allows the speed to exceed a preset value.

Throttle valves are of two types, single port and multi-port. A five ported multi-valve is illustrated in Figure 2. The individual ports of a multi-valve are progressively opened as the lift-bar is raised. Total lift of a multi-valve is nominally about 1 ½ in. as compared to ¼ to ½ in. for a single valve. Valve plugs, seats and throats are commonly designed consistent with thermodynamic conditions.

In addition to the control valves some turbines are provided with ports through which the throttled steam passes from the steam chest to the blades of the first stage wheel. In some turbines hand valves are used to expose groups of ports. For example, several ports may be open at all times while other groups are opened by exterior manual adjustments. The number of ports in service may vary from 6 to 20, typically. The fraction of first wheel periphery to which steam is admitted depends upon the number of ports in service.

Speed control disturbances

The action of turbine speed control systems is frequently essentially proportional. To minimize "droop" or steady state error, the open loop gain is made as high as possible. In some instances the resultant droop in speed as load is increased may be as little as 4%. The deadband in a well designed system is less than 0.1% of normal speed.

Four causes of difficulty may be encountered in turbine-compressor ensembles such as illustrated in Figure 1. These are: (1) disturbances, (2) coupling, (3) malperforming components, and (4) inferior control strategy.

Typical disturbances result from changes in:

1. Compresor demand caused by changes in available process gas.
2. Quality of steam generated by the process.
3. Quality of high pressure steam available to the high pressure turbine.
4. In the intermediate pressure steam usage.
5. Pressure of high pressure steam.

The control system must compensate for these disturbances. The coupling may be obvious or subtle. Obvious coupling occurs between the two turbines via the intermediate pressure steam line. A change in this pressure changes the distribution of power developed by each turbine, giving rise to speed changes. Speed changes, in turn, produce changes in intermediate steam pressure. The net interaction may not be severe but it must be considered.

Another obvious coupling exists between the process and the steam generated. The dynamics of this coupling is relatively slow but none-the-less it is present.

A more insidious type of coupling can occur through the hydraulic control system. Sudden volume changes in this oil system will result in corresponding changes in the pressure throughout. Over-ride mechanisms may permit a strong coupling between valve servo motors through the control oil system.

This may become particularly troublesome if a rather rigid linkage in the over-ride arrangement is allowed to occur. For example, a change in the position of the low pressure turbine throttle valve servo motor pilot valve may induce a change in the pressure in the multi-valve servo on the large turbine. If coupling is strong enough, sustained or prolonged oscillation will occur following a disturbance. Coupling of this kind can even extend into the pneumatic components which frequently appear in the interface between the process and the turbine control systems.

Knowledge of component performance

The performance of all components, as well as the process to be controlled, must be known if the control syytem is to be designed. Unless one is accustomed to designing high performance control systems it is easy to neglect applying design principles or to evaluate properly proposed control configurations.

Because turbine-compressor systems are often the most sensitive and critical part of the process it is mandatory that they exhibit highest reliability and possess fool-proof control and fail-safe features. A single catastrophic failure of the principal turbine-compressor system may well eliminate the profit from a large processing unit for months!

Component performance requirements and the development on control strategy can be illustrated using the arrangement of Figure 1 as an example.

Let it be assumed that speed is to be regulated by the low pressure turbine, the speed reference signal being generated in response to compressor demand. The high pressure turbine is to be used as the principal source of power. Further it will be assumed that it is necessary to maintain the pressure in the intermediate pressure steam line within very close limits e.g. ± 3 lb.

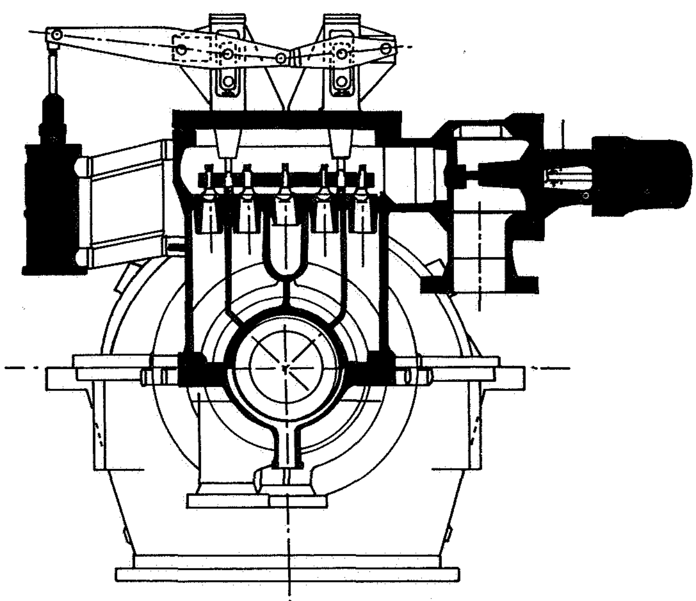

Figure 2. Five-port multivalve and actuating mechanism, with the emergency stop shown on the right. (Courtesy Siemens AG Berlin-Munich. Siemens America, Inc., N.Y. - Member of the Siemens Group.

/sq. in. at some elevated level of several hundred pounds per square inch.

Control valve sensitivity is of crucial importance in any process control design. For best results valves should have linear performance; that is, the relation between flow and positioning signal should be linear. As valves open, the effective pressure drop usually decreases (unless critical conditions exist) and usually the sensitivity begins to decrease. Single-ported valves by themselves will commonly have performance characteristics similar to that shown in Figure 3.

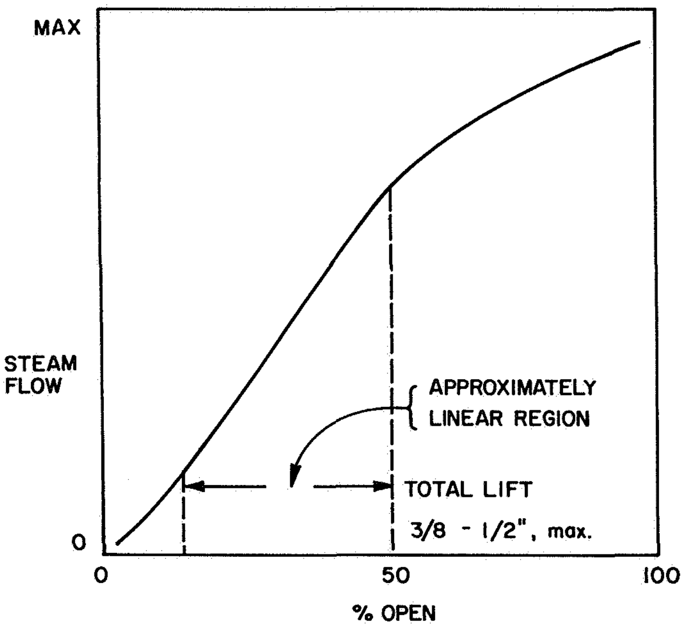

Figure 3. Typical flow-lift characteristics of a turbine single valve.

Controlling valve sensitivity

In order that the control system exhibit high performance at all loads the sensitivity (slope of above curve) of the valve must be constant or continuously compensated for by changes elsewhere in the control system. Since adaptive features are not widely accepted in the process industry it is easier to strive for constant valve sensitivity.

In the high lift region the sensitivity of the valve tends to decrease. Thus a sound strategy in the above arrangement is to assure that the valve is always positioned in a nearly linear region. An acceptable operating region might be as illustrated in Figure 3. How this is done will be described subsequently. Multiported throttle valves, such as used on large turbines and illustrated in Figure 2, tend to exhibit linear behavior over an appreciable portion of the 1-1/2 to 2 in. lift.

Some manufacturers provide facilities to vary the number of ports through which steam from the steam chest flows to the first turbine wheel. The number of ports in service is changed by an adjustment made manually from outside the turbine case. In such instances the linear range of the throttle valve is determined by the number of ports in service. Figure 4 shows typical behavior.

It is readily seen that if, for example, only six steam chest ports are open, both the effective range and the linear region is restricted. As more steam chest ports are opened the linear range is extended. It is likely that when steam pressure is high (say 500 lb./sq. in. or more) and all ports open, the throttle valve may operate critically; that is, flow is a function of upstream pressure and valve displacement only. The decrease in valve sensitivity may be the result of the increased pressure between valve and steam chest ports which causes departure from critical flow.

If it is necessary to operate the turbine over a range where control valve performance is non-linear, it will not be possible to design a simple control system which will give high performance at all levels . Either performance will deteriorate as the valve moves out of the linear region and the system will have decreasing response, or, if designed to operate well in the non-linear region, will become increasingly oscillatory as the valve moves back into

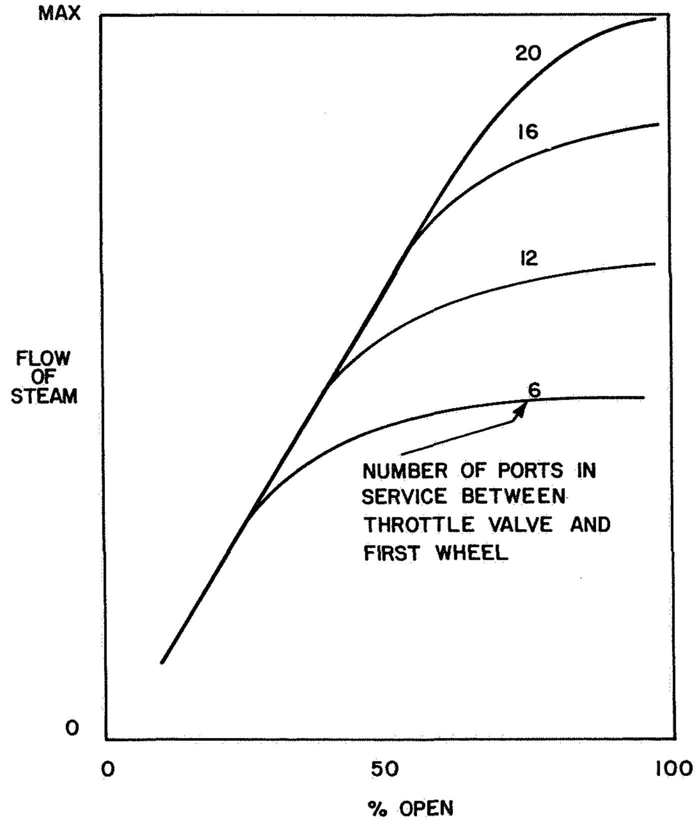

Figure 4. Flow-lift characteristics of a turbine throttle valve operating in series with steam chest ports.

the linear zone.

Therefore, it is advisable to have as many steam chest ports open as is required to assure reasonably linear performance of the throttle valve over the anticipated range of operation.

Suggested control system

One possible control configuration is shown in Figure 5. A measure of load demand is developed in component L, utilizing pertinent measurements from the process. Sometimes a measure of compressor suction pressure is sufficient.

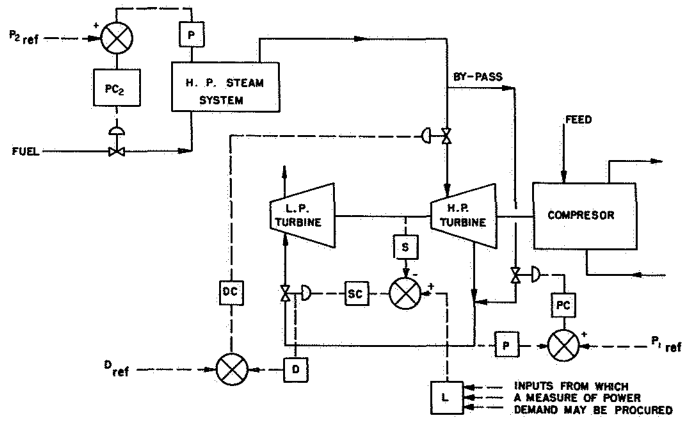

Figure 5. A proposed turbine-compressor control scheme.

The output from component L sets the reference speed. The difference between reference and actual speed, as measured by speed sensor, S, is used as the input to speed controller SC. The output from SC ultimately adjusts the low pressures turbine steam valve so that agreement between desired and actual speed is obtained.

A measure of the displacement of the low pressure turbine valve stem is obtained via component D. This is compared with a reference (desired) value of stem displacement and the difference signal supplied to displacement controller, DC. The output from DC ultimately adjusts the position of the throttle valve on the high pressure turbine. When steady state has been restored, the low pressure throttle valve will be in its original position and the high pressure throttle valve in the new position required to satisfy the load.

A logical scheme by which to control the pressure in the intermediate pressure main is to regulate a small flow of high pressure steam into this line. The response of intermediate pressure to changes in steam inventory will usually be very rapid owing to the relatively small capacitance of this main. Pressure controller PC, must provide compensation in a relatively high frequency range and the associated (let-down) valve must be free from stiction and be very responsive.

A well-designed, high performance, control system for intermediate pressure control will virtually eliminate intermediate pressure variations, thus removing one source of disturbance and coupling. By adjusting the ratio of power delivered by each turbine, by adjusting D ref, the steady state quantity of let-down steam required can be made as small as desired for a given operating load.

The sizing and performance of the let-down valve are very important. *Flow-lift relations must be known quite precisely if quantitative design is to be achieved.* Simplication results if the flow through the let-down valve is in the critical regime.

Means for achieving control

A block diagram of the complete system is shown in Figure 6. In the upper left hand portion the steam generating system is shown. Control of the pressure of high pressure steam is achieved by regulating the flow of fuel. This system reacts with the remaining system principally via the pressure on the high pressure turbine control valve and the by-pass (let-down) valve. Disturbances from this source will generally be slow.

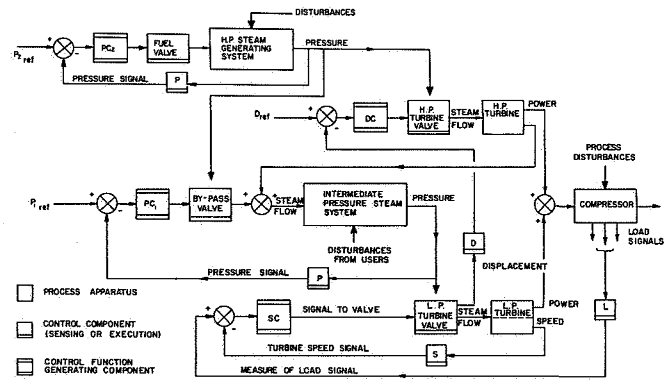

Figure 6. Block diagram of a proposed control system for a turbine-compressor ensemble. (Large arrows indicate where disturbances enter or where coupling may occur between components).

Intermediate steam pressure is controlled by the by-pass valve, as described previously.

The sum of the power generated by the turbines (product of torque and speed) is shown entering the compressor. Signals from the process are used as measures of load. The load signal, converted to an equivalent speed, is compared with actual speed and the difference, via controller SC, used to regulate compressor speed.

The low pressure turbine throttle valve position is compared with its reference value and the difference, via controller DC, used to reset the high pressure throttle valve. A control strategy such as just described appears to satisfy, qualitatively, all the requirements.

Quantitative design requires a knowledge of the static and dynamic properties of all components of the system. These properties of sensors and controllers may be measured in the laboratory. The dynamic characteristics of control valves, servo motors and associated components would be measured in the field. Flow-lift relations for control valves, must, most likely, be determined experimentally on the operating system. Similarly the performance of the high pressure steam generating system, the intermediate steam system and the turbine-compressor ensemble must also be found by field tests.

Obtaining dynamic data

Dynamic information both in the laboratory and plant is obtained relatively easily by pulse testing methods. This is done by exciting the given component or subsystem by injecting a pulse-like change in an input. Records of input and output are obtained and these data processed to yield frequency response information. A number of publications describe results obtained by testing processing components in this way (8, 9, 10,).

Essentially linear descriptions will usually be found which are adequate for design purposes. Well-known feed-back control system synthesis methods can then be applied and a system with specified performance assembled.

Bode's rules for non-minimum phase systems can usually be applied using Bode plots of the experimental data for the process and control components. The feedback controller functions are chosen once the performance characteristics of the total open loop, excluding the controller, are known. The parameters in the controller function are chosen to satisfy Bode's rules, i.e. the slope of the Bode plot of the complete feed forward open loop (including controller) should be about -1 and the associated phase no more than -150° at gain cross-over (unity gain). Modifications of these rules are discussed by Oldenburger (11).

It should be mentioned that since the undamped natural frequency of turbine-compressor ensembles is in the neighborhood of 1 cycle/sec. dynamic compensation must be provided in a higher frequency range than chemical engineers are accustomed to. Also to avoid the difficulties accompanying pure delay time, stiction, backlash and lost motion in mechanical linkages must be reduced to a minimum. The magnitude of these can be determined by tests on the assembled system before start-up.

Experimental measurements will also yield the information needed to design the components which produce static conversion, such as L and D. Relations useful for developing feedforward control can also be found from experimental measurements.

The normal complement of industrial sensors and instruments does not have the capability of procuring data useful for control system design and cannot be used in experimental studies having this objective.

A special system is mandatory if useful data are to be obtained. The data system need not be complicated but must be versatile and convenient to use.

Selection and installation of sensors

Special attention must be given to the selection and installation of sensors. These must be highly responsive and sensitive. Strain gage transducers have proved extremely useful for measuring pressures and pressure gradients. In various configurations they may be used to measure displacement and strain with great accuracy. Resistance elements in bridge circuits are excellent for sensing temperature and temperature differential. Linear potentiometers may be used for measuring large displacements such as valve stems.

Signal conditioning consists of filtering, amplification and suppression. The latter is quite necessary especially when measuring dynamic responses. Frequently it is more important to measure changes in process variables than absolute values. Suppression enables the experimenter to bias out a known level of the variable and amplify the difference above that level.

Direct writing oscillographs are recommended for recording purposes. Direct writing has the advantages that results can be observed as tests are made and much is learned without extensive data processing. Faulty components and poor data can be recognized immediately.

In the writer's experience such systems have proved entirely reliable and easy to use. Plant environment has never posed serious problems.

Contractors bearing the overall responsibility for turbine compressor systems are well-advised to become conversant with the details of proposed ensembles prior to their installations, especially the control system. Performance characteristics should be requested on all components where this is feasible. Provision for procuring additional data should be incorporated in the final installation and a testing program initiated as soon as possible during or following start-up.

If such service is not offered by the contractor the users should become self-sufficient in this respect. An investment of, say $100, 000, in test equipment and time is small indeed compared to the losses which could incur from a single catastrophic failure. A well executed test program immediately following start-up will also insure attaining optimum operating conditions early in the life of the process, in addition to supplying good performance data useful for future designs.

Because measuring, testing and design methods have been proved in the study of a wide variety of processes, including turbine-compressors, satisfactory performance of designed systems is assured through the combination of experimentation and control system design theory.

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