RELIABLE OVERSPEED PROTECTION FOR INDUSTRIAL DRIVE TURBINES

by

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ABSTRACT

The paper discusses the fundamental relationships between turbine overspeed and the time constant of the rotor. The amount of overspeed is shown to be related not only to the time lag of the emergency tripping devices, but also to the amount of thermal energy stored in piping to the turbine. Methods of protecting against overspeed are described and discussed. The use of redundant and "fail-safe" designs is given as a way of arriving at a system to give reliable protection of the turbine against overspeed.

INTRODUCTION

A large bucket or a section of a wheel which separates from the rotor of an overspeeding steam turbine can penetrate the casing wall. Major damage can result to the turbine and to the driven equipment, with corresponding danger to human life, when excessive turbine overspeed occurs. The control of and the protection from excessive turbine overspeed is, therefore, of major concern to the turbine designer.

It is the purpose of this paper to discuss the various factors taken into account by the turbine designer in arriving at a system to give reliable protection of the turbine against overspeed.

SPEED OF THE TURBINE

The magnitude of the problem of overspeed protection depends on how fast the turbine can change speed. The ability of a turbine to change speed is commonly measured by its time constant, Tc. For an instantaneous step change in position of the turbine valves, and assuming that speed is a first order function of torque, the turbine rotor will change speed in accordance with

$$n = 1 - \epsilon^{-t/Tc} \qquad (1)$$

where:

- n = change in speed, per unit
- ϵ = 2.71828
- t = time, seconds
- Tc = time constant, seconds

From the equation it can be seen that when t = Tc, then $n = 1 - \frac{1}{\epsilon}$ or n = .63. This leads to one definition of rotor time constant.

> "The rotor time constant gives the length of time it takes the rotor to reach 63 percent of its total speed change due to an instantaneous change in turbine valve position."

The rate of change of speed at the instant the valve position is changed can be found by differentiating equation (1) and setting time equal to zero. This results in:

$$\frac{dn}{dt} = \frac{1}{Tc} \qquad (2)$$

From this equation it is seen that dn = 1.0 when dt = Tc. This leads to a second definition of rotor time constant.

> "The rotor time constant gives the length of time it would take the rotor to reach 100 percent of its total speed change due to an instantaneous change in turbine valve position if it continued to change speed at its initial rate."

Figure 1 shows the change in rotor speed versus time for an instantaneous change in turbine inlet valve position. This figure illustrates the basis for the two previous definitions of rotor time constant.

The general definition above can be made more illustrative by having the instantaneous valve opening change corresponding to a full load change. In this case, the following definition applies:

> "The rotor time constant gives the length of time it would take the rotor to reach double speed due to a 100% instantaneous drop in turbine load provided the rotor continued to change speed at its initial rate."

This definition is useful in visualizing and approximating what happens in the very short time between loss of load and activation of controls and safety devices. If it is assumed that

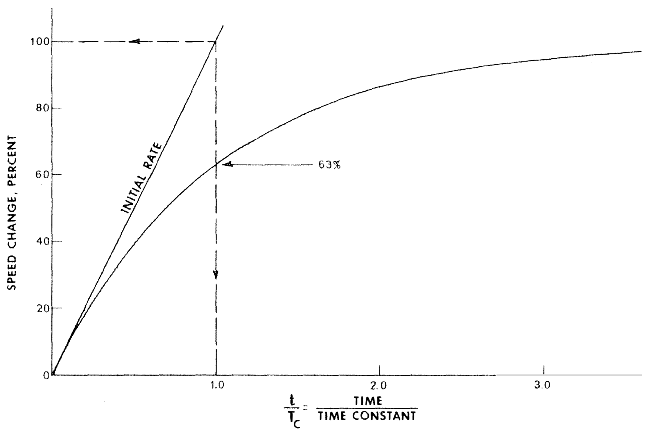

Figure 1. Response of a system with a single time constant, Tc. Change in speed versus time for an instantaneous change in position of the turbine inlet valves.

steam flow is unchanged and that all relationships are linear, then it follows that for an instantaneous change in load and for small values of time:

$$\Delta n = L \frac{t}{Tc} \tag{3}$$

$$\text{or} \quad t = \frac{\Delta n}{L} Tc$$

where:

t = time, seconds

Δn = speed change in time "t," percent

Tc = rotor time constant, seconds

L = instantaneous load change, percent

From equation (3) it can be seen that it will take a rotor only $\frac{Tc}{10}$ seconds to change speed 10% if there is an instantaneous 100% change in load. This is a significant speed change that takes place in a very short time. Therefore, extremely fast speed controls and emergency overspeed tripping systems are required to hold speed rise to a reasonable value on instantaneous loss of full load.

ROTOR TIME CONSTANT VALUES

Rotor time constant is a basic measure of rotor response. The rotor time constant can be obtained from the basic equation for horsepower by substituting the relationship of speed and time for acceleration. The equation for rotor time constant is:

$$Tc = 0.619 \left[\frac{N}{1000}\right]^2 \frac{[WR^2]}{H.P.} \tag{4}$$

where:

Tc = rotor time constant, seconds

N = rated speed, rpm

WR^2 = rotor inertia, lbs-ft^2

H.P. = rated horsepower

The rotor inertia, WR^2, is the only factor in this equation that depends on the design of the turbine and, therefore, is not known until the design is complete. The WR^2 for mechanical drive turbines varies over a large range (about 1000/1) but tends to have an inverse relationship with speed. Therefore, the turbine rotor time constant generally falls in the rather narrow range of 2 seconds to 8 seconds with an occasional rotor as fast as 0.5 seconds or as slow as 10 seconds.

Knowledge of the rotor time constant is fundamental to the understanding of the total system performance. The speed of all controls or relays and the speed of the rate of change of load are measured relative to the speed of the rotor. Any change which requires more time than about 1/10 of the rotor time constant can be considered to be slow. A "slow" relay or control tends to be bad while a "slow" load change tends to be good.

LOSS OF LOAD

Loss of load has been assumed to be "instantaneous" in previous discussions. By "instantaneous" it was meant that the change took place in zero time. It is obvious that an instantaneous load change will result in the maximum change in speed. This leads to the question of the difference between speed response to an "instantaneous" load change and to a "sudden" load change.

Figure 2 shows the change in speed for an instantaneous change in load and for several values of linear ramped changes in load, T/Tc.

T = total time for the load change to take place, seconds

Tc = rotor time constant, seconds

t = time, seconds

You will note from these curves that the ramped load introduces a lag in the rotor speed response. This lag in rotor response is approximately equal to one half the ramp time, T, for small values of T/Tc. For example, a system with a rotor time constant of 5 seconds starts at time zero to have the load linearly ramped "suddenly" to a new value of load in one second. The change in speed with time will be approximately the same as if there had been no load change for the first 1/2 second and then the load change took place "instantaneously."

Generator drive machinery can lose full load in a few cycles, or about 1/20 second or less. It is reasonable that these turbines should be designed to protect against the instantaneous loss of full load. Since this loss in load can occur without uncoupling the generator, the rotor time constant includes both the turbine and generator WR^2. This increases the combined rotor time constant and helps minimize the rotor overspeed.

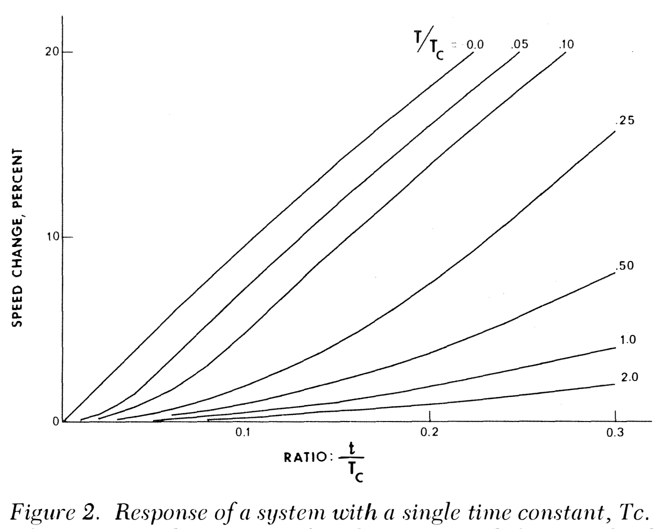

Figure 2. Response of a system with a single time constant, Tc. Change in speed versus time for a linear ramped change in load from zero to maximum time, T.

The mechanical drive turbine coupled to a pump or a compressor is a different matter. The load is not easily disconnected. There are only four ways to achieve sudden loss of load. The first is to throttle the pump or compressor suction. This is not readily accomplished and is only part load reduction at most. A flashing pump or a surging compressor may still require upward of 30% of full load. More importantly, to throttle a suction requires time, probably a minimum of one or two seconds. This is a "slow" action for all except the very slowest, 10 second turbines.

The second method of losing load is to break a coupling. This method is often referred to in conversations on the overspeed subject. It is, however, extremely rare. The author has personal knowledge of only one such coupling break in the past 35 years. The fact that there was no damaging overspeed to the turbine when it occurred leads one to believe that the breaking process must have been "slow." There is no data on the subject.

The third method of losing load is in process upsets. These upsets are generally 20% or less in load occurring over several minutes time. This category of upset is extremely "slow" relative to any turbine rotor time constant and, therefore, of no concern relative to turbine overspeed protection. It is possible that a process upset could result in a surge in the system. Even if such a system surge did dump as much as 70% load in one second, this would still be a "slow" action for all except the very slowest turbines.

The fourth is to have a rupture of a compressor discharge line. It is conceivable that in the worst case this could result in a load loss approaching 100%. This loss in load will not be instantaneous but will require time which is a function of the system size and pressure levels.

From the above discussion, it appears reasonably conservative that mechanical drive turbines be designed to protect against excessive overspeed due to a sudden loss of 100% load with the sudden load change occurring at a uniform rate in one second. Since this loss in load will occur without uncoupling the driven load, the rotor time constant will include both the turbine and the driven load WR^2.

DESIGN OVERSPEED

Overspeed protective devices are generally set to operate at about 110% of rated speed. If the speed governor functions as specified by NEMA Class D, it will limit the maximum speed of the turbine to 107% rated speed when 100% load is suddenly reduced to zero. Therefore, as long as the speed governor functions properly, the action of the overspeed trip will occur only as a result of a slow speed test. This test demonstrates the function of the parts but does not represent the maximum speed excursion that will occur under the actual emergency tripping conditions. It is important that the maximum speed excursion not damage the rotor. The safe allowable maximum speed will vary from rotor design to design, but a usual reasonable maximum would be 125% rated speed. At this speed all rotor stresses will be 56% above normal with corresponding reductions in design factors of safety.

OVERSPEED DUE TO TIME LAG

There are two basic sources of overspeed beyond the 110% normal emergency tripping speed setting. One of these is time lag in the mechanism used to close the trip throttle valves to the turbine. The other is the energy available in the steam stored between the closing valve and the turbine.

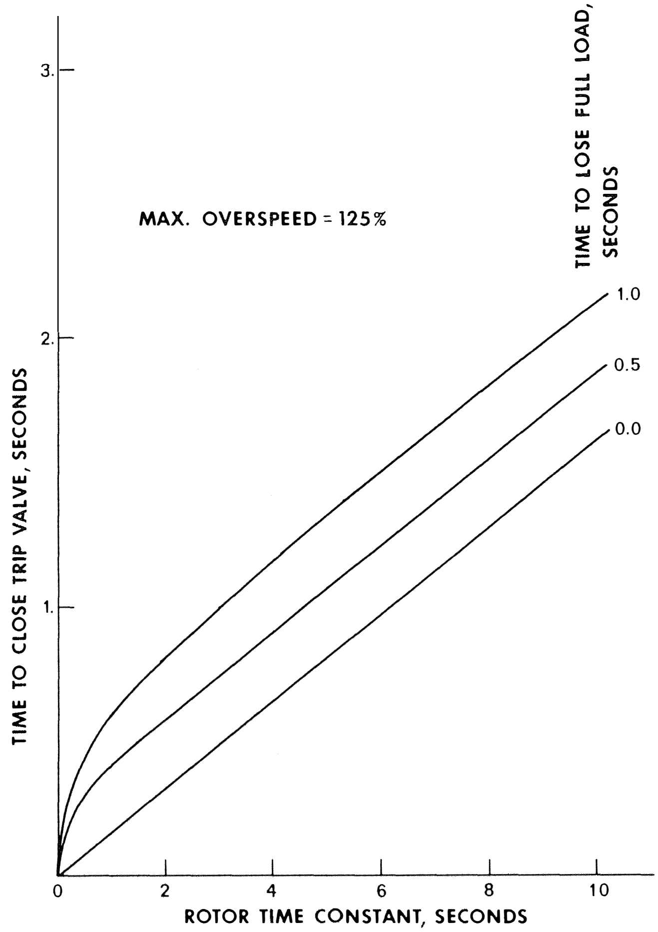

Figure 3. Time required for a system with a rotor time constant, Tc, to accelerate from 110% to 125% speed on "sudden" loss of 100% load.

Figure 3 shows the time in which the emergency governor must actuate and the trip throttle must close in order to limit the maximum overspeed to 125% rated speed. These curves are based on the emergency governor being set to operate at 110% rated speed and they assume that there is no overspeeding from energy of stored steam. They are also based on the conservative assumption that the governing valves are inactive and that the only speed limitation is by the action of the trip throttle valve closing. Curves are shown for instantaneous loss of 100% load and for a uniform rate of load loss to give 100% loss in both 0.5 seconds and 1.0 seconds.

There is no technical problem in providing adequate overspeed protection for rotors having time constants of two seconds or greater. However, for smaller rotor time constants, the reduced closing times required become troublesome. Special designs are required to obtain closing times of less than about 0.3 seconds.

The fact that it requires some time to lose load is particularly significant to the potential overspeed of turbine rotors with low time constants. You can see from Figure 3 that if there could be an instantaneous loss of full load on a 1/2 second turbine rotor, you would have to close the throttle trip valve in 0.08 seconds in order to limit the overspeed to 125%. To achieve this small a value of total tripping time is beyond present experience. A conservative estimate for mechanical drive turbines is that it will take at least one second to lose full load. With this load loss you would have to close the throttle trip valve in 0.44 seconds in order to limit the overspeed to 125%. This value of total tripping time is practical to achieve.

OVERSPEED DUE TO STORED ENERGY

The previous discussion of overspeed due to time lag in closing various relays and the trip throttle valve makes the assumption that when the valve is closed, all steam to the turbine is cut off. This is not true. There is stored energy in the steam that exists in the piping and turbine casing between the trip throttle valve and the turbine nozzle plate. This steam, as well as any steam or water that exists in the casing itself and its extraction openings, or connected piping and storage volumes, acts to drive the turbine rotor after the trip throttle valve is closed.

The change in speed of the rotor, due to stored steam and water, can be found by equating the energy in the steam and water to the change in kinetic energy of the rotor. The following equation shows the maximum speed that can be reached above trip speed due only to stored energy of steam and water. This equation is based on a turbine which has 75% efficiency at rated steam conditions and rated load. The inlet steam pressure and the turbine efficiency both decrease as the stored steam is used. The average effective efficiency of the stored steam utilization is approximately 60%.

$$\left(\frac{Nf}{Nt}\right)^2 = 1 + \frac{2.72\,(BTU)}{WR^2(Nt/1000)^2} \qquad (5)$$

where:

BTU = total stored energy, BTU

WR^2 = rotor inertia, lbs-ft

Nt = trip speed, RPM

Nf = maximum speed, RPM

If it is considered that only the inlet piping has stored energy, then the above equation can be simplified to show the number of feet of piping that is required to store enough energy to overspeed the turbine. The following equation shows the piping length required to overspeed from a trip speed of 110% rated speed to a maximum speed of 115% rated speed.

$$P = 0.095\ TcV \qquad (6)$$

where:

P = length of inlet pipe between the trip throttle valve and turbine, feet

Tc = turbine rotor time constant, seconds

V = inlet piping steam velocity, ft/sec

For a usual inlet piping velocity of 150 ft/sec and for a one second time constant rotor, it would require the equivalent of 14 feet of piping between the trip valve and the turbine first stage nozzles to overspeed the turbine from 110% trip speed to 115% maximum speed.

The steam chest volume plus the volume of the piping itself should be considered as the total volume of the equivalent length of piping.

This result indicates that additional overspeed due to stored energy can be appreciable if the trip throttle valve is mounted off the turbine, or, as in some designs, there is piping between the governing valves in the turbine steam chest and the first stage nozzles.

Stored energy in extraction lines can also be appreciable. Non-return valves in extraction lines should be mounted with a minimum of piping between them and the turbine casing. In general, the total stored energy from all sources should be restricted to give an additional overspeed of the turbine of not more than 8% or 9% so that when it is combined with the overspeed due to time lag of the relays and valves, the total overspeed will not exceed 125%.

A major potential source of stored energy in extraction openings is water. Water has almost 100 times the stored energy of an equal volume of steam at 50 psig. This ratio becomes less as the pressure increases such that at 400 psig water has only 20 times the stored energy of an equal volume of steam. In any event, the high energy level of water emphasizes the importance of proper design of extraction piping to provide adequate drainage and to avoid pockets where water could accumulate.

OVERSPEED PROTECTION

There should always be at least two independent methods of protecting the turbine against accidental overspeed. The speed governor usually acts as one of the methods. The speed governor is designed not only to control turbine speed of load, but also to limit the turbine speed in case of sudden loss of load. The rate of response of these controls may be determined by either of these requirements. The rate of response of the controls for the "slow" turbine is determined by the stability requirements of the system. The "fast" turbine may have the rate of response of its controls determined by the need to limit overspeed on sudden loss of load.

The emergency governor (emergency trip) acts as a second independent system to limit excessive overspeed. The emergency governor is set to operate at a higher speed than would be allowed by the speed governor. Therefore, it is expected to operate only on malfunction of the speed governor. The emergency governor function may be either mechanical or electrical.

The mechanical emergency governor has been used for many years and is probably the most common application today. It is used when only a single emergency trip function is supplied. The electrical emergency governor has experienced increased usage in recent years. Both single and dual electrical emergency governors have been applied along with the mechanical emergency governor to give redundant tripping systems. A system of two duplicate but independent emergency tripping systems is also used with no mechanical emergency governor. When only electrical emergency governors are used, it is good practice to supply two independent emergency tripping systems.

MECHANICAL TRIP SYSTEM

The total mechanical system is one in which a spring loaded plunger, mounted in the turbine shaft, is actuated by overspeed to move a system of linkage and levers to unlatch the throttle valve and allow it to close. This system is adequate for low rated turbines where the forces involved are small and the distances required for the linkage system are short. This system becomes increasingly penalized by frictional and inertial forces as the size and loading of the turbine becomes larger. In these larger turbines, the mechanical trip system is, in fact, a mechanical-hydraulic system. A typical trip system of this kind is shown in Figure 4.

The primary action of the system is that the emergency trip plunger overcomes the spring force at trip speed and snaps out to strike and open the hydraulic trip valve. This lowers the supply oil pressure to the trip throttle valve which unlatches and allows the valve to close. The same rapid closing action can be obtained on the trip throttle valve by any other fast acting

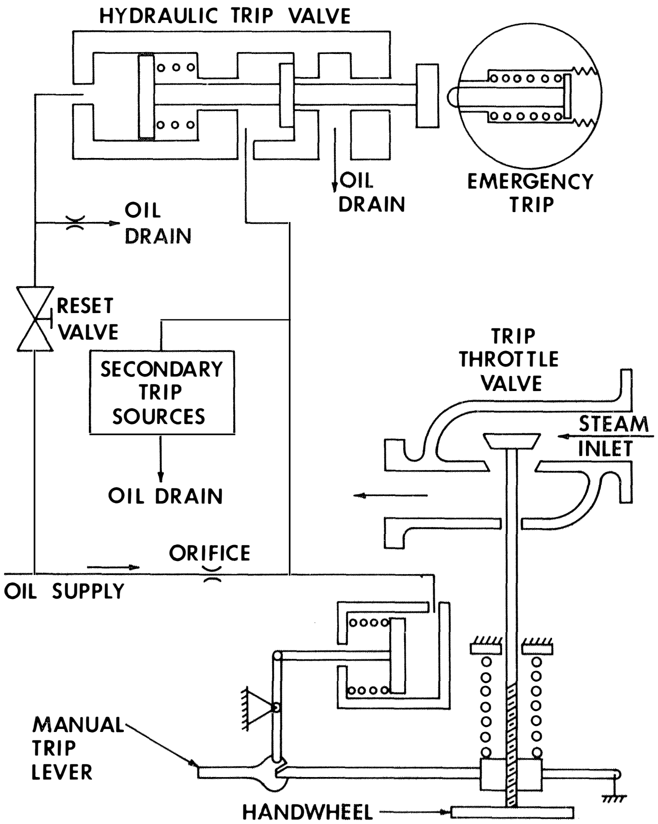

Figure 4. Schematic representation of a simple mechanical-hydraulic emergency trip system.

oil dump valve that is placed in the oil line between the hydraulic trip valve and the trip throttle valve. These possible additional sources of tripping the turbine are identified on Figure 4 as Secondary Trip Sources. The two most common secondary trip functions are for low lube oil pressure and manual tripping. Either of these can be done by a direct acting valve or by a solenoid actuated valve. A manual trip remote from the turbine must be solenoid actuated. A manual trip lever can also be built into the trip throttle valve as a totally mechanical function rather than a combination hydraulic-mechanical function.

The hydraulic trip valve is held closed against a spring by unbalanced oil pressure across the valve. Once the valve has tripped opened, the spring will hold it open. It can be closed by opening the reset valve to put oil pressure on the small piston built into the hydraulic trip valve. There is an orificed line to drain between the reset valve and the hydraulic trip valve to prevent accidental pressure existing on the piston when the reset valve is closed. The reset valve can be operated either manually or remotely.

The mechanical emergency governor has been used for many years and has been developed to a high degree of reliability. It has, however, certain disadvantages which are:

Trip settings cannot be changed in operation.

Can be tested only at relatively high speeds.

Redundancy is not feasible.

It has moving and wearing parts.

The plunger and spring design of the emergency trip is such that higher trip speeds require increasingly smaller plunger weights and relatively larger spring gradients. Thus, as the trip speed increases, the sensitivity of the trip increases. By sensitivity, is meant the change in trip speed for a given movement of the plunger. In some high speed trips, the trip speed will change over 100 rpm for less than one mil movement of the plunger. The high speed trips are, therefore, more difficult to manufacture and to set, and will have more spread in their trip settings than lower speed trips.

The electric trip system does not have the disadvantages which are listed for the mechanical emergency governor and, of course, it has no plunger and spring for a primary source of speed sensing. Like all designs, it has its own evaluation which will be discussed below.

ELECTRIC TRIP SYSTEM

The electric trip system is different from the mechanical trip system in that the functions of the emergency trip and the hydraulic trip valve are performed electrically. The other parts in the system can be the same. A single electrical trip system in its simplest form is shown in Figure 5. An oil actuated stop valve is shown in this figure, but a trip throttle valve as shown in Figure 4 could have been used.

In the electrical system, the speed is sensed by a magnetic pickup located in proximity to the teeth of a wheel mounted on the turbine shaft. The magnetic pickup generates a voltage and its frequency is a function of speed and the number of teeth on the wheel. The relay tachometer converts the alternating portion of this input signal into a D.C. analogue voltage which is proportional to speed. The D.C. analogue voltage is compared to a "set" voltage, adjustable from the power source, and a switch is actuated when the "set" voltage is reached or exceeded. The switch actuates the solenoid valve to dump the oil and close the trip throttle valve. Once the relay tachometer has actuated the trip switch, it will remain in that position until a reset button is pushed and a predetermined lower speed has been reached.

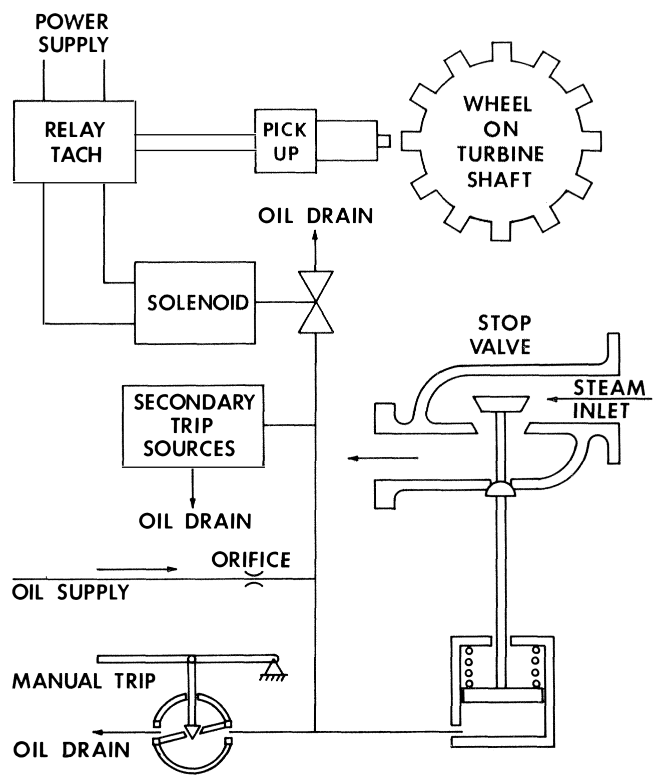

Figure 5. Schematic representation of a simple electric-hydraulic emergency trip system.

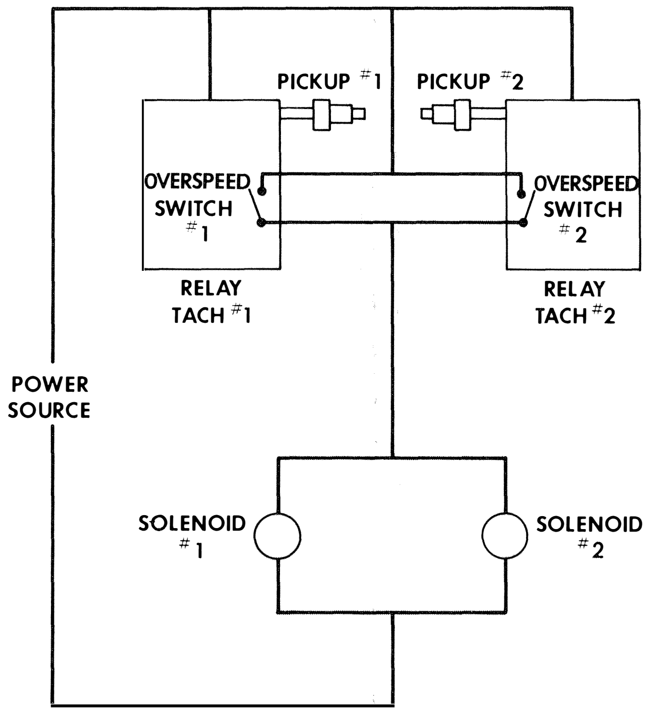

Figure 6. Schematic of dual electrical system for overspeed trip.

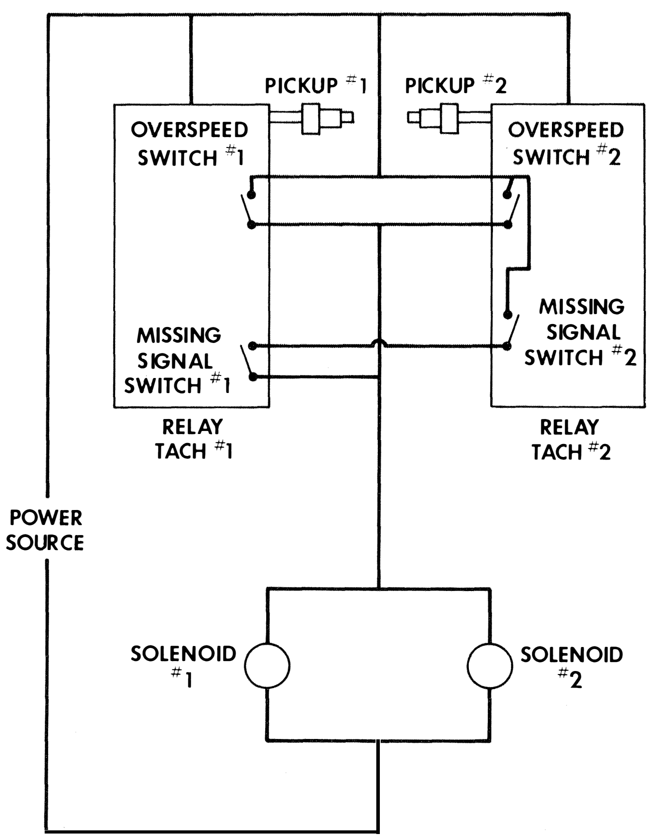

Figure 7. Schematic of dual electrical system for overspeed trip modified to safeguard against missing frequency signals.

This fundamentally simple electrical system can be used with great flexibility to supply redundant, fail-safe tripping systems with features to detect faults in the system and to test its function.

DUAL ELECTRIC SYSTEMS

It is good engineering practice to have the additional reliability of a dual electrical trip system. The arrangement of such a dual system is shown in simple form in Figure 6. Both magnetic pickups are generally mounted to sense speed from the same toothed wheel on the turbine shaft. The two switches and the two solenoids are both arranged in parallel so that the signal from either pickup #1 or pickup #2 may actuate either or both solenoids. The solenoids act to dump the oil and close the turbine stop valve. With this parallel arrangement, a failure may occur in any single electrical part and still allow the proper functioning of the trip system.

The two solenoid trip valves may both be used to actuate (trip closed) the same inlet stop valve or they may each actuate a separate stop valve. General practice is to use a single stop valve, but dual stop valves have been used where maximum protection against overspeed is required or dual steam sources are employed.

FAIL-SAFE

All turbine overspeed trip designs should be made to "fail-safe." The turbine manufacturer considers the design to be "fail-safe" when a failure in the system causes the turbine to shut down. This gives maximum protection to the turbine. Consideration of possible failure modes makes necessary additional circuits in the simplified dual electrical system shown in Figure 6.

MISSING SPEED SIGNAL

A circuit is introduced to notify and to protect against missing speed signals. A schematic of how this is done is shown in Figure 7. When there is no or very low frequency signal from the pickup, the corresponding "missing signal" switch is closed. The two missing signal switches are placed in series with each other but in parallel with the overspeed switches. Therefore, when both missing signal switches are closed, a circuit is completed that energizes the solenoids and trips the turbine stop valve. A single missing signal will not trip the stop valve. However, when that missing signal switch is closed, it also closes a circuit to give an alarm (not shown) to notify operating personnel that there is a malfunction in that circuit.

LOSS OF POWER

The solenoid should be de-energized to trip in order to meet the turbine manufacturer's "fail-safe" criteria. Then any failure of the power system, the wiring up to the solenoid or the solenoid itself will cause the solenoid to actuate and trip the turbine stop valve. A careful review of both solenoid and actuating switches should be made to see that they can accommodate the full energizing current to the solenoid on a continuous basis.

This "fail-safe" mode of operation raises the possibility of a system malfunction in which the actuating switches which have been improperly selected fail to operate because they have become fused shut by the continuous passing of solenoid actuating current. Concern with this possibility has resulted in trip systems in which the solenoids are designed to energize to trip. When this is done, it is necessary to add a "loss of power" circuit which will independently dump the oil and close the turbine stop valve whenever the power source to the regular solenoid trips is interrupted. While this circuit protects from failure of the power system, it does not protect from failure in the wiring up to the solenoid or failure of the solenoid itself.

LOW OIL SUPPLY

The trip system should be "fail-safe" not only electrically but also hydraulically. The stop valve is spring loaded and requires oil pressure to open. Loss of oil supply pressure will cause the trip throttle or stop valve to close. Tripping the turbine out of service as a result of low oil pressure is accomplished by either or both of the following methods:

The turbine stop valve may be equipped with an oil relay valve that latches at normal supply oil pressure but which opens at about 35 psig, dumping the oil under the piston to drain, and closing the stop valve.

The lube oil pressure is monitored by a pressure sensitive switch which closes at a set pressure below the normal safe valve to actuate the solenoid valve and to dump the oil and close the turbine stop valve. The low lube oil pressure system also includes an alarm which is activated at a higher pressure than will actuate the tripping mechanism.

OTHER ALARM AND TRIP SYSTEMS

Any number of other trip or alarm and trip systems may be used. These systems can be all placed in parallel so that any one of them will actuate the solenoids to dump the oil and close the turbine stop valve. A listing of the various factors that are usually considered for purposes of alarm or trip is given in API 612.

STOP VALVE

The "fail-safe" design feature of the stop valve has already been discussed. All of the previous discussion of systems and their speed of response is based on the assumption that the stop valve will go closed when it gets the signal to do so. The fact that this does not always happen has led to the occasional use of dual stop valves. The primary reason that stop valves do not close is that they have become fouled with carry-over in the steam. There are two design features which protect against fouling. One of these is the stem seal. This design feature is depicted in the stop valve sketch in Figure 5. A small valve becomes solidly seated when the stop valve is full open. It takes the full load of the oil pressure on the stop valve piston. When the stop valve is open there is no steam leakage along the valve stem and, therefore, there is no chance of fouling deposit in the close clearances between the valve stem and its bushing which is outside the regular steam path.

A second design feature is that of being able to "exercise" the stop valve during normal turbine operation. The stop valve is designed so that it can be moved through about the upper 30% of its stroke. This "exercising" should be done on a regular monthly basis to give some assurance that deposits have not built up excessively on that portion of the valve stem which is exposed to the inlet steam and that the trip system operates normally.

A schematic of the exercise feature is shown in Figure 8. The "Block" valve is closed first to isolate the stop valve. The "Exercise" valve is then opened and the solenoid trip is actuated by electrically simulating a speed increase. This drops the pressure under the piston and allows it to move until it covers the opening in the cylinder wall that is piped to the exercise valve. A reversal of this procedure restores the stop valve to its normal function. This procedure not only protects against valve stem fouling, but it also gives a test of the complete trip system including relay tachometer, trip solenoid and stop valve, with exception of the speed pickup.

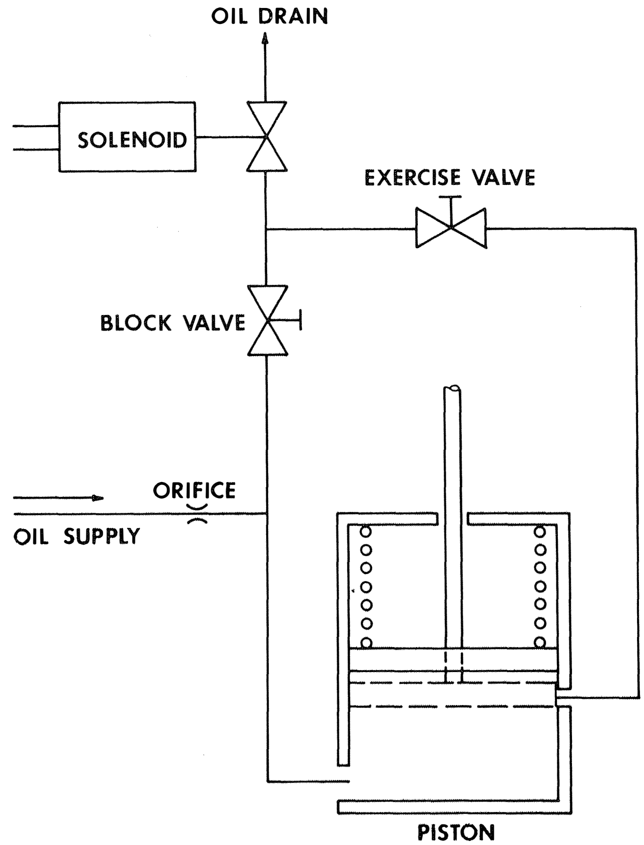

Figure 8. Schematic of stop valve "exercise" system.

SUMMARY

To give reliable protection of the turbine against overspeed requires fast acting, dependable designs of tripping systems that can be tested with the turbine in operation.

The speed with which the emergency trip system must act to limit overspeed to a safe value is a function of the acceleration of the turbine on loss of load.

The acceleration of the turbine on loss of 100% load is a function not only of the time constant of the rotor, but also of the rate at which the load is lost.

Stored energy in steam and water contained in the piping to the turbine can cause appreciable overspeed and should be considered in the design.

The emergency trip system should take into account all possible modes of failure and should be designed to "fail-safe."

Redundant electrical hydraulic systems are available that can provide protection not only against overspeed, but also can act to shut down the turbine for any specified malfunction of either the turbine or the process in which it is used.