TURBINE OVERSPEED SYSTEMS AND REQUIRED RESPONSE TIMES

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ABSTRACT

Turbine overspeed events lead to the most dangerous and costly of catastrophic turbine failures. Traditionally, overspeed protection systems were designed and provided by the original equipment manufacturers (OEMs)—few options were available either in terms of testing or modifying these systems. With new turbines, and with the retrofit of older systems, it is more common to see third party systems playing this critical protective role. However users often do not appreciate the overspeed protection system response time that is required to adequately protect their equipment.

This tutorial will explain the basic components of a turbine trip system, and consider sources of variability in the response times of those components. Industry standards, API 612 (2005) and API 670 (2000), will be used to show how to calculate the required overspeed protection system response time. Users will walk out of this tutorial with a basic understanding of what comprises a steam turbine trip system, a greater appreciation of the affect of the system response time, and an understanding of how to verify that their system has the adequate response time to protect their turbines, plants, and personnel.

INTRODUCTION

A turbine overspeed event can lead to catastrophic failures that may generate unanticipated, uncontained liberation of turbine blades and components, release high pressure and temperature steam, and result in secondary damage including process piping damage, chemical release, fires, and missile damage. Its causes and prevention are common topics of presentations, articles, sales brochures, as well discussions with operators, societies, manufacturers, and insurance carriers.

New technology has brought solutions to some of the shortcomings of traditional mechanical protection mechanisms and brought a host of new possibilities, choices, and potential problems with overspeed protection systems. It has also brought the possibility to better understand, analyze, and measure the performance needed and delivered by protection systems. Industry guidelines are being expanded to incorporate the new technologies as well as the need to increase the owner/operators understanding of their system. The newest challenge facing owners and operators is understanding the response requirements of their systems and the testing that is required to assure the system is adequately protected.

This tutorial will present the traditional protection means, the new systems, guidelines for system designs that exist today, and new information that is being added. The authors will look at these new guidelines in detail and provide examples of several different systems. This tutorial will use industrial standards, API 612 (2005) and 670 (2000), to assist with educating turbine users on specific timing requirements and testing necessary to safely protect their turbines and their driven loads. Since the entire steam turbine trip system must respond fast enough to safely protect the turbine and its driven load (compressor, generator, pump), the timing of each basic trip system component will be reviewed. The authors will break down the components of the total system that determine the magnitude of an overspeed event. They will also discuss some of the testing options and tradeoffs that are now available. One objective is to demonstrate how to calculate and verify the required component response times based on the required total system trip time.

It is becoming common to find steam turbines that have been or are being upgraded with new controls and/or related shutdown systems. This tutorial will focus on what turbine response times are required to adequately protect turbines of all sizes. Examples based on real turbines will be used to compare the differences between turbine shutdown systems scan rates and response times.

Cost and in some cases the recommendations of OEMs may encourage the adoption of different test methodologies. Since testing of these trip systems has become a large discussion topic throughout the Turbomachinery Laboratory's Turbomachinery Symposium's discussion groups, basic testing procedures will be compared in an effort to educate users on best practices.

MECHANICAL OVERSPEED SENSING SYSTEMS

Traditional mechanical overspeed sensing systems relied on the balance between the centripetal force imparted on a weight by the rotation of the shaft opposed to a spring with a known and specific constant. At the designed overspeed trip point, this balance position of this weight allows it to physically contact with a lever that releases a valve that causes the trip oil header to drain and lose pressure. The spring, bolt, and trip lever can be seen in Figure 1.

Figure 1. Mechanical Overspeed Trip Mechanism Photo. (Courtesy Younie, 2009)

Figure 2 does not show manual trips or trip overrides for testing purposes that are associated with this system but, essentially there is a direct, if complex, mechanical connection between the trip bolt and the trip oil header dump valve.

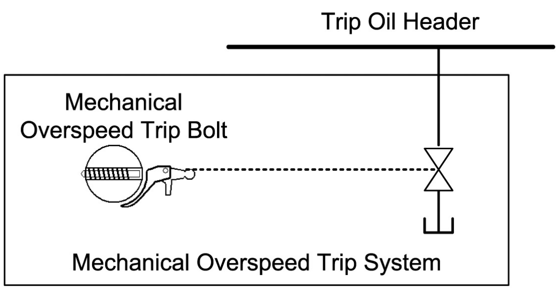

Figure 2. Mechanical Overspeed Trip System Schematic (Simplified).

The loss of trip oil header pressure may directly move a trip mechanism or may operate through a hydraulic circuit to cause stop valves to close. In general, the trip oil system follows a common design regardless of the sensing mechanism although the trip mechanism designs on the steam valve can vary. The trip systems will be described in more detail later.

Since the mechanism is not easily observed, it is possible for a significant problem to go undetected. The author is familiar with a case where the assembly was built incorrectly but the system appeared to function properly for years (the bolt was not striking the trip lever in the correct location). The risk of the failure was arguably higher because the system was not operating as it was designed to.

On many systems there is only a single bolt. Because the contact with the lever occurs over a relatively limited angle there is an inherent variability and a worst case delay of approximately 15mS (if the trip is at 4000 rpm). There are some designs where the head of the bolt is designed to provide contact with the trip lever over a wider angular range.

Some mechanical trip systems incorporate two bolts. This provides redundancy in the system and allows testing without compromise of protection. In those cases where there are two bolts, they can be configured 180 degrees apart from each other on the shaft such that response latency issue is reduced by half.

The response delay is not the primary issue with the mechanical trip systems. The greatest concern is variability in the trip point and response delays caused by the tendency for these systems to stick. Most users with mechanical trip systems will be familiar with the phrase, “let’s wait for a little bit” or “let’s give it a little time” during tests. This is not an option in a real overspeed event. Some systems allow an operator to lock out the trip and exercise the bolt periodically to “clean” the components and reduce the possibility of sticking. The quality of lubrication oil that is used in the trip system and the possibility of fluid contamination, or incompatibility of components with the fluid, can affect the response of mechanical overspeed protection systems.

Additionally, mechanical wear or changes in the spring constant in these systems can affect the setpoint. Multiple tests of these systems are required as the test itself could affect the mechanism. If subsequent tests result in increasing trip point, the assembly must be repaired. Generally this requires replacement of components, recalibration, and retesting. As mentioned previously, the components are also susceptible to sticking. In those systems where oil can be injected behind the bolt, it is possible to correlate the test pressure to the theoretical speed at which the trip would occur. But, in general, testing these systems requires bringing the turbine up to the overspeed trip point (multiple times), which causes stress to turbine rotors and generators.

ELECTRONIC OVERSPEED DETECTION SYSTEMS

Electronic overspeed detection systems are common today. They are provided as new equipment from OEMs and as retrofit systems. Today, API 612 (2005) and 670 (2000) refer exclusively to electronic overspeed detection systems.

The electronic overspeed sensing systems generally use a toothed wheel (or gear) to sense speed. Either active or passive probes detect the passage of the teeth and then a digital logic solver is used to determine the shaft rpm based on these signals. Figure 3 shows a functional schematic of an electronic overspeed trip system.

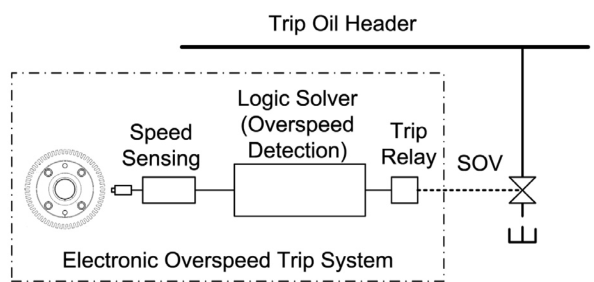

Figure 3. Electronic Overspeed Trip System Schematic.

API 612 (2002) 12.3.2.1 states, “An overspeed detection system based on three independent measuring circuits and two-out-of-three voting logic in accordance with API Std 670 shall be supplied.” So a sensing assembly may look something like Figure 4. Since the same gear may also be used for the control speed sending, many of the brackets look like Figure 5.

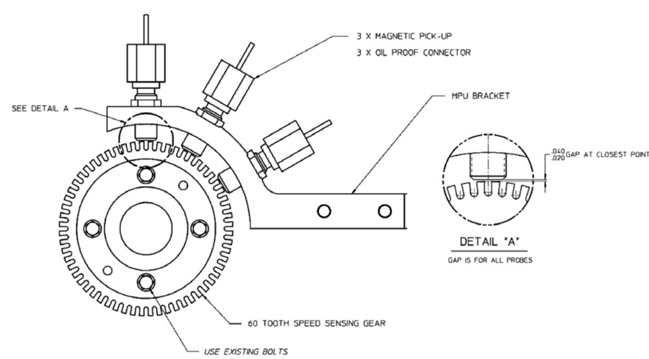

Figure 4. Speed Sensing Gear and Bracket with Three Probes. (Courtesy Woodward)

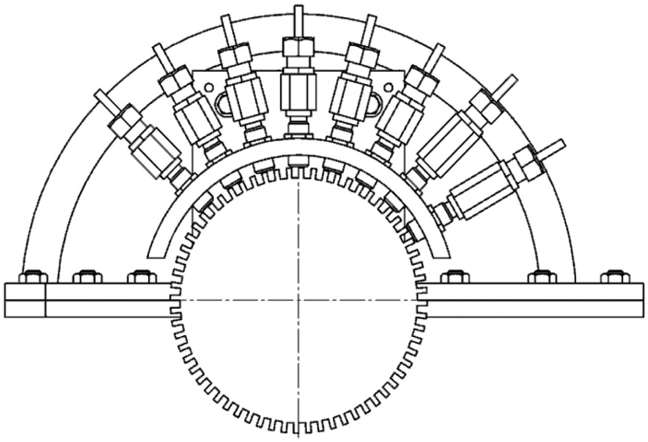

Figure 5. Speed Sensing Gear and Bracket with Multiple Probes. (Courtesy Woodward)

The speed of response of the detection of an overspeed event is obviously important. API 670 (2000) 5.4.8.4b. states, "Unless otherwise specified, the system shall sense an overspeed event and change the state of its output relays within 40 milliseconds …" but it goes on to note that "40mS response time may not be adequate in all cases to keep the rotor speed from exceeding the maximum allowed for the machine …"

This 40mS includes the time for each channel to detect speed, compare it to an overspeed setpoint, vote the results, and then to output a trip command. This functionality is shown, versus time, in Figure 6.

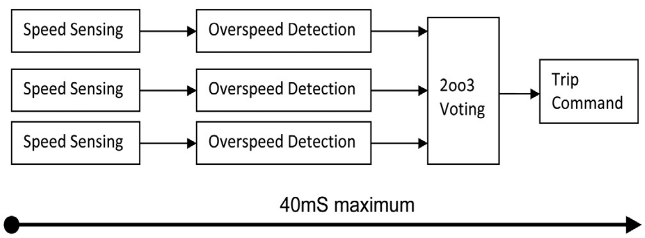

Figure 6. Functional Block Diagram of Electronic Overspeed Protection System.

The trip command typically involves the deenergization of a trip relay, which may take around 10mS. Assuming the overspeed detection and voting can be accomplished in a negligible amount of time, only 30mS is left for speed sensing. But what if the actual overspeed rpm was reached just after the speed sensing routine started? Then the overspeed would not be detected until the speed sensing routing was executed the next time. In reality the speed sensing routine in this example would have to run every 15mS. The speed sensing algorithm must be accomplished both accurately and quickly.

The speed sensing routine can be broken into a data collection (or sampling) phase and a speed calculation phase (Figure 7). In the data collection phase, time between pulses is measured and/or pulses are counted over a fixed time. In the speed calculation phase, those data are used to calculate the turbine rpm. The correct calculation requires that the correct number of sensing gear teeth and proper gear ratio, if applicable, are entered into the overspeed protection device.

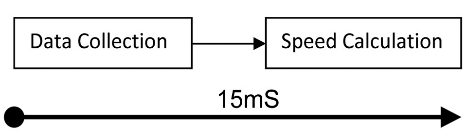

Figure 7. Functional Block Diagram of Speed Sensing Routine.

NOTE: At 4000 rpm, one shaft rotation requires 15mS so calculating speed on a once per revolution basis requires too much sampling time.

Different sensing algorithms have tradeoffs primarily between accuracy and response time. Measurements on a per tooth basis can be quick but are susceptible to noise due to variation in the tooth spacing. Counting the number of teeth in a fixed time yields a very poor resolution with the sampling time available. The maximum number of sample points in the shortest possible time is desired but there are practical limits on how many gear teeth can be used and on the frequencies that can be sensed practically.

There is no simple speed sensing algorithm that provides the performance required. Fast and accurate sampling hardware is required and, ultimately, some form of averaging is needed. Very advanced speed sensing systems model once per revolution variations and can produce a very accurate speed measurement on a per tooth basis. But the use of this technology is not common on overspeed protection systems that the author is familiar with.

Although faults in the logic solver cards are among the most common reported faults in the overspeed systems, the use of redundant and triple modular redundant systems mean that detected faults do not necessarily mean a compromise of the system protection or response. Further, these systems typically employ means for testing individual circuits to confirm availability and detect faults and allow for their repair during extended periods of operation. These features are generally needed to meet safety integrity levels (SIL), which are being specified more commonly today for protection systems. SIL relate to the reliability and risk of dangerous failure of a safety system. The definition of an application of SIL requirements is specified as part of IEC 61508/61511.

In general, systems specifically designed for overspeed protection and compliant with API 670 (2000) are neither a major cause for concern nor an issue in the response of the overall protection system. Overall, an electronic overspeed protection system should be considered superior to a mechanical system for machine protection. Replacing a mechanical system with an electronic overspeed protection system has been recommended by OEMs and insurance carriers to improve system protection.

SOLENOID OPERATED VALVES (SOV)

The interface between the overspeed detection system, or logic solver, and the hydraulic trip oil header is a solenoid dump valve. API 612 (2005) 12.3.3.1 requires a minimum of two solenoid operated valves in the shutdown system.

An electronic overspeed sensing system will typically deenergize a solenoid valve or multiple solenoid valves to dump the trip oil. Solenoid valve responses can vary greatly depending on size, design, operating function, fluid media, temperature, inlet pressure and pressure drop. A very fast solenoid valve may have an operating time on the order of 10mS but reliability and robust design must also be considered in the selection. Response times in trip systems may range from 10s to 100s of mS depending on the size, complexity, operating pressure, and other requirements of the system.

For a given solenoid valve, its response time might significantly be affected by maintenance, replacement of coils, and addition of components such as flyback, or noise-suppression, diodes, or interposing relays. It is a common practice in the retrofit of mechanical control systems to add flyback diodes to suppress contact arcing of relays in a new control system. Ideally, these are placed as physically close to the SOV or relay coils as practical as indicated in Figure 8. Sometimes these diodes are incorporated directly into cable connectors and may not be visible or apparent. In some relays, these diodes are integral to the relay housing. The dynamic affects of this modification may not be recognized and may be significant.

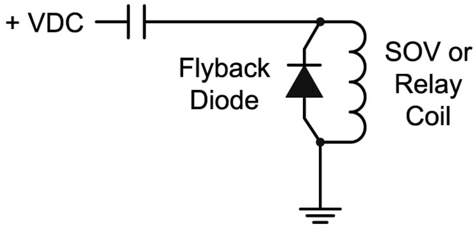

Figure 8. Flyback Diode Schematic.

In a test of a trip system, the author monitored the voltage across the trip solenoid coil and the pressure in the trip header. In this test, the trip header was capped off within the trip system so the volume of the trip header was very small and there were no other devices connected to the header. A solenoid valve with and without diode noise suppression was tested. The time between deenergizing the trip solenoid and the first indication of the start of the pressure drop in the trip oil header was monitored. To understand the absolute response of the solenoid valve, the response of the pressure transmitter must be considered and other factors such as the test pressure and temperature need to be matched to the operational values. Even so, it is valid to attribute the difference in the responses recorded to the difference in the noise suppression mechanism.

Multiple tests were performed with each configuration and typical results are shown in Figures 9 (no noise suppression diode) and 10 (with noise suppression diode). In this particular case, the solenoid response time was increased from approximately 140mS to 220mS by the addition of the noise suppression diode. The increase of 80mS is significant in an overspeed protection system.

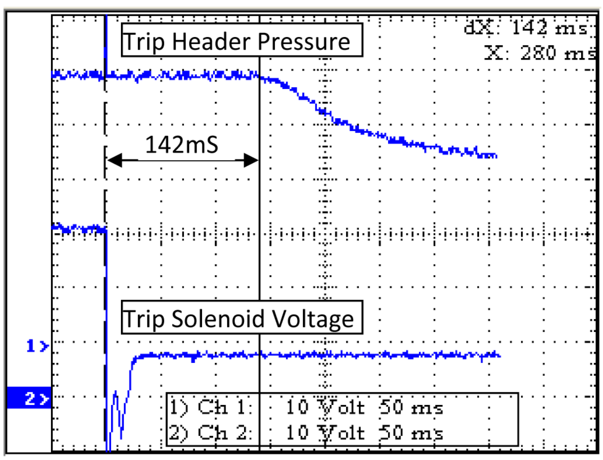

Figure 9. Solenoid Response with No Noise Suppression.

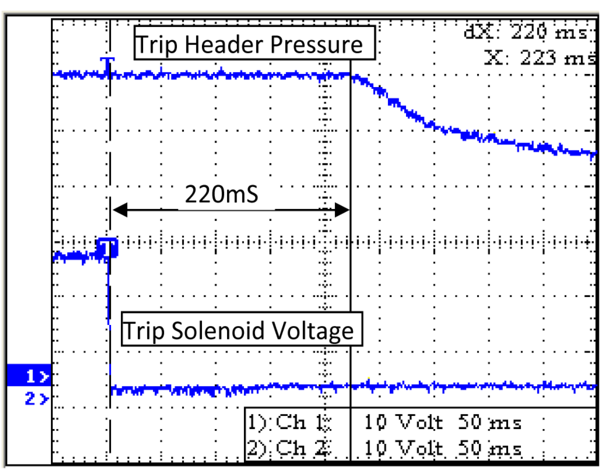

Figure 10. Solenoid Response with Diode Noise Suppression.

A metal oxide varistor (MOV) can limit the voltage across the relay contacts but dissipates the energy of the collapsing magnetic field from the solenoids coil more quickly than a diode. Tests with metal oxide varistors of 25 V and 56 V yielded response times of approximately 147mS and 142mS, respectively. The solenoid valve response with the MOVs was nearly as fast as with no noise suppression and much better than with the diode noise suppression.

In a test with a faster acting SOV, the response time with the diode was 26mS and without the diode was 9mS. With the MOVs the response was around 10mS.

Just picking the fastest SOV is not the answer. There was some concern that the faster SOV had tighter tolerances on the internal components and that it may be more susceptible to contamination of the hydraulic supply. The system must be both fast and reliable.

Problems with the solenoid valves account for a significant number of faults in overspeed protection systems. In one study, more faults were attributed to SOVs than sticking trip valves and faults in the electronic cards were only slightly more common (Hesler, 2004). In several different studies, problems with SOVs are rated with stop valves and faults in electronic cards as the leading root cause of overspeed protection system failures. With this known likelihood of faults, system redundancy, a periodic test schedule, and preventive maintenance programs become a necessity.

THE TRIP OIL SYSTEM

Once a mechanical or solenoid operated trip oil system dump valve opens, pressure held in the trip oil system is bled off rapidly. When the trip oil system pressure drops, trip or shutoff valves, and in some cases control valves, are forced closed. The actual mechanism that closes the valves can take a variety of forms. In some cases a hydraulic trip relay is used to block the oil that would have held an actuator open and open a path to drain to the actuator cylinder. System return spring forces would then force the actuator and therefore the valve closed. In another design, a similar hydraulic trip relay would be used but in a double acting actuator it would port oil to drive the actuator closed. Another trip method utilizes an actuator connected directly to the trip oil system. As trip oil pressure is lost, a spring moves the actuator to mechanically unlatch a trip valve that shuts off steam to the turbine. Spring forces then drive the steam valve closed when it is unlatched. A total overspeed protection example is shown in Figure 11.

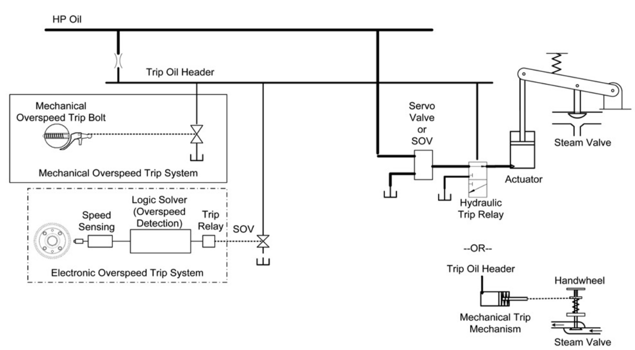

Figure 11. Overspeed Trip System.

The response of this system significantly depends on its design. Obviously, the valve must be sized correctly to rapidly dump the trip oil system pressure. In addition to response time, valve selection and placement affect the total delay associated with the control system response. Changes in the piping design can have a significant effect on the response time. One example showed how the response of a trip system with 20 ft of piping between a trip cylinder and the drain could be improved from 100ms to about 30ms by shortening the piping to 2 ft (Jacoby, 2008).

Viscosity is a factor in the system response. Although it is generally assumed systems will be up to operating temperatures before the overspeed system would be required, operators should not overlook potential issues with changes to fluid or fluid quality. Again, issues with contamination and compatibility can affect the system response and even prevent it from functioning at all.

The oil in the trip system is typically considered a noncompressible fluid but there is a study that shows, in low pressure trip oil systems, it is possible for gases trapped in the trip oil to come out of solution and significantly increase trip delay time by affecting the rate of pressure decay in the trip oil header (Cote, 2008).

STEAM VALVE CLOSURE TIME

Once the trip oil header pressure is lost, the steam valve or valves are driven closed either by spring force or by hydraulic force. These valves close very quickly. Valve closure times are on the order of 100s of mS. Systems from the OEM should comply with the API recommendations that prevent overspeed. But changes to the actuation systems or steam valves should be studied carefully and preferably performance should be tested to ensure response times are adequate.

Greater problems are encountered when the valves stick. This can slow the valve closure or prevent the valve from closing. Valve stems can become stuck for a variety of reasons including deposits on the valve or valve stem, problems with packing, bushings, or use of improper lubrication. Damage to, improper manufacture of, or use of improper materials in the valve or valve seat could allow steam to continue to enter the turbine even after the valve is closed. This could cause conditions where an overspeed cannot be prevented. Periodic testing can prevent some of the causes of valve sticking and allow operators to discover a problem under controlled conditions.

THE TOTAL OVERSPEED PROTECTION SYSTEM

This is a system and there are a variety of areas where problems can occur. One thing makes response time testing difficult is that the system, when operating properly and even sometimes when not, is very fast. It is so fast that a user would not likely be able to detect a fault short of a total failure to operate. For this reason, there is a strong argument to instrument the trip system in such a way that the total system response can be measured. During tests a change in the response could indicate a degradation that might compromise system protection or point out a failing component. The author is not aware of any user that is currently doing this today. This seems strange in light of the great emphasis placed on testing the setpoint. In some cases, system response times have been tested when put into service and this would still be recommended as a baseline if changes are ever made to the system. But these response time data are not being used to evaluate the systems present status.

REQUIRED RESPONSE TIME

The authors have broken the overspeed system into components and examined issues, tradeoffs, and potential problems with each of the component elements. But, how fast must the overall system respond?

- From API 612, Sixth Edition, November 2005:

12.3.1.1 "… The system shall prevent the turbine rotor speed from exceeding 127% of the rated speed on an instantaneous, complete loss of coupled inertia load while operating at the rated conditions. …" It goes on to state, "… In the event of loss of load without loss of coupled inertia, and unless otherwise specified …, the system shall prevent the speed from exceeding 120% of rated speed …"

The burden of meeting this requirement is placed on the turbine vendor but it is important for the owner/operator of the equipment to understand and be able to validate how this is accomplished. This is particularly important as maintenance, replacement, or retrofit of original equipment provided may affect the system.

ASME PTC 20.2 (1965), "Overspeed Trip Systems for Steam Turbine-Generator Units", Section 5 discusses computation of emergency overspeed. This calculation methodology is referred to both in API 612 and API 670:

- From API 612, Sixth Edition, November 2005:

In section 12.3.4.7, "… The calculation methodology shall be in accordance with ASME PTC 20.2."

- From API 670, Fourth Edition, December 2000:

5.4.8.4 b "… Response time must consider complete system dynamics … as outlined in ASME PTC 20.2-1965."

In the upcoming version of API 612, these formulas will be incorporated as a new Annex, L.

Essentially these equations take the total energy in the turbine at the time the trip speed is reached. This includes the rotor kinetic energy and the in-train energy with consideration of the efficiency of the conversion of that energy. To this the additional steam energy that is admitted during the overspeed trip system detection and response delays and the energy admitted while the stop valve(s) is closing is added. This total energy is converted to the equivalent kinetic energy without consideration of losses.

EQUATIONS FOR CALCULATING THE MAXIMUM ROTOR SPEED

Acceleration (rpm/s):

$$\alpha_t = k \times \frac{P_{g(\max)}}{N_T \times WR^2} \tag{1}$$

Rotor kinetic energy:

$$E_T = k_2 \times WR^2 \times N_T{}^2 \tag{2}$$

Ctrl time delay energy:

$$\Delta E_s = T_s \times P_{g(\max)} \tag{3}$$

Stop valve closure:

$$\Delta E_v = f \times T_v \times P_{g(\max)} \tag{4}$$

NOTE: In more complex systems where multiple stop valves are employed, such as reheat steam turbines, these must be considered too.

Trapped energy:

$$\Delta E_e = k_3 \eta \left(\sum W_{1i} u_{1i} - \sum W_{2i} u_{2i} - \sum (W_{1i} - W_{2i}) h_{2i} \right) \tag{5}$$

Total rotor energy:

$$E_{\max} = E_T + E_s + E_v + E_e \tag{6}$$

Maximum rotor speed (rpm):

$$N_{\max} = \sqrt{\frac{E_{\max}}{k_2 \times WR^2}} \tag{7}$$

where:

$P_{g(\max)}$	=	Turbine rated power
N_T	=	Setpoint of overspeed trip device (rpm)
WR^2	=	Rotational inertia of turbine (uncoupled)
Ts	=	Signal time delay (seconds)
T_v	=	Closure time for stop valve (seconds)
f	=	Fraction of maximum steam flow that passes through the stop valve during closure period
h	=	Steam turbine efficiency
W_{1i}	=	Mass of steam and condensate contained within each "i" space inside the turbine when the turbine is operating at its maximum output
u_{1i}	=	Internal energy for each of the steam W_{1i} masses, estimated at the actual pressures and temperatures that exist at the various "i" spaces when operating at maximum output

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W_{2i} = Weight of steam in the "i" spaces defined for W_{1i} after expansion has ceased

u_{2i} = Internal energies for the W_{2i} masses of steam in the "i" spaces after isentropic expansion

h_{2i} = Enthalpies of the W_{2i} masses of steam after isentropic expansion

Constants and Units

Where independent of the system of measurement, units are shown in the equations above.

- In SI units:
  Power is in (kW), Energy is in (kW-sec), Rotational inertia is in (kg-m^2)
  k = 9.1189×10^4 (rpm^2-kg-m^2/[kW-s])
  k_2 = 5.49×10^{-6} (kW -sec-min^2/[kg-m^2])
  k_3 = 1.0 (kW-sec/kJ)
- In US customary units for a generator drive:
  Power is in (kW), Energy is in (kW-sec), Rotational inertia is in (lb-ft^2)
  k = 2.164×10^6 (rpm^2-lb-ft^2/[kW-s])
  k_2 = 2.31×10^{-7} (kW -sec-min^2/[lb-ft^2])
  k_3 = 1.055 kW-sec/BTU)
- In US customary units for a mechanical drive:
  Power is in (hp), Energy is in (hp-sec), Rotational inertia is in (lb-ft^2)
  k = 1.614×10^6 (rpm^2-lb-ft^2/[hp-s])
  k_2 = 3.10×10^{-7} (hp-sec-min^2/[lb-ft^2])
  k_3 = 1.415 (hp-sec/BTU)

Acceleration (Equation 1)

Although not needed for the calculation of the maximum rotor speed, the equation for instantaneous acceleration at the moment of the load loss is included. This could potentially be used as the basis of some anticipatory trip logic.

Rotor Kinetic Energy (Equation 2)

This calculates the kinetic energy of the rotor at the overspeed trip point. This is needed as one component of the total system energy that will be used to calculate the maximum overspeed.

Ctrl Time Delay Energy (Equation 3)

The ctrl time delay energy equation recognizes Ts, the time interval between overspeed trip action and start of emergency stop valve closure. Ts is the sum of the time delays of the individual components of the overspeed protection system. Ts = speed sensing time + detection time + electronic output circuit response time + solenoid response time + hydraulic system response time. Particular care must be taken to understand the total additive affects of system delays.

During this time the full energy of maximum power (in the worst case) will continue to be admitted into the turbine.

Stop Valve Closure (Equation 4)

The stop valve closure equations recognize Tv, the time between the end of Ts and the moment when the respective emergency stop valve is closed. With the flow characteristics of the valve considered, the amount of energy that passes through the valve while it is closing can be calculated.

In some cases there are multiple valves in different locations so the various closing times, flow characteristics, and energy available must be used for each valve and the totals summed together.

Trapped Energy (Equation 5)

Even after the steam valves are closed, the steam behind the valves is still expanding through the turbine. So the turbine speed continues to increase until this in-train energy has been converted to kinetic energy. In some cases this may be simply approximated by taking the volume of the piping between the stop valve and the turbine. Depending on the turbine design this might be considered negligible. In the case of more complex turbines this may involve multiple spaces with steam at various energy levels and detailed knowledge of the efficiency of the turbine. Generally this would require data from the OEM. Neglecting operation at off-rated conditions, this energy will remain constant unless significant modifications are made to the turbine. These factors should be considered during turbine uprates or modifications to improve efficiency.

Total Rotor Energy (Equation 6)

All the sources of energy are summed together to get the total energy in the system.

Maximum Rotor Speed (Equation 7)

The total energy and the rotor inertia are used to calculate the maximum rotor speed that could be achieved. The calculated maximum rotor speed can be compared to the limits recommended by API or to the OEM specified limits to determine if the overspeed protection systems response time is adequate.

The examples that follow will provide some sense of response times required for several different steam turbines. This will also highlight the affects of changing the control response time to provide a better appreciation for how much, or little, variation is acceptable and safe.

EXAMPLES

Example 1

A 22 MW turbine with a rotor inertia of 17,882 lb-ft^2 (753 kg-m^2) and a trip speed of 4250 rpm.

- Acceleration (rpm/s):

$$\alpha_t = k \times \frac{P_{g(\max)}}{N_T \times WR^2} \tag{1}$$

Description	Symbol	Value	Units
Conversion factor, accel	k	1.614E+06	RPM^2-lb-ft^2/(HP-s)
Turbine max net power	Pgmax	300268.1	HP
Trip speed	Nt	4000	RPM
Turbine rotor inertia	WR_2	308000	lb-ft^2
Acceleration, initial	Alpha	393.4	RPM/sec

The instantaneous acceleration is 626.7 rpm/sec.

- Rotor kinetic energy:

$$E_T = k_2 \times WR^2 \times N_T^{\ 2} \tag{2}$$

Description	Symbol	Value	Units
Conversion factor, Et	K_2	3.10E-07	
Kinetic Energy at trip speed	Et	1527680	HP-sec

The rotor kinetic energy at trip speed is 100,128 hp-sec (74,695 kW-sec).

- Ctrl time delay energy:

$$\Delta E_s = T_s \times P_{g(\max)} \tag{3}$$

If the control time delay is 0.25 seconds:

Description	Symbol	Value	Units
Control time delay	Ts	0.1	sec
Control Delay energy	Delta_Es	30026.81	HP-sec

The energy added during the control time delay is 7372.654 hp-sec (5500 kW-sec).

- Stop valve closure:

$$\Delta E_v = f \times T_v \times P_{g(\max)} \tag{4}$$

NOTE: In more complex systems where multiple stop valves are employed, such as reheat steam turbines, these must be considered, too.

If the stop valve closure time is 0.7 seconds and the flow characteristic is 0.5:

Description	Symbol	Value	Units
Valve flow characteristic	f	0.84	fraction
Valve closure time	Tv	0.29	sec
Power (hp)	Pg(hp)	80428.95	HP
Valve delay energy (sv)	Delta_Ev	19592.49	HP-sec

The energy added during the stop valve closure is 10,321.72 hp-sec (7700 kW-sec).

- Trapped energy:

$$\Delta E_e = k_3 \eta \left(\sum W_{1i} u_{1i} - \sum W_{2i} u_{2i} - \sum (W_{1i} - W_{2i}) h_{2i} \right) \tag{5}$$

In this example there is only one area of trapped energy to consider:

Description	Symbol	Value	Units	
Conversion factor, Delta Ee	K_3	1.41485	HP-sec/BTU	
Turbine isentropic efficiency	Eta	0.75	fraction	
Weight of fluid 1 start	W_11	2.636134	lbm	Steam Chest
Weight of fluid 1 end	W_21	0.007689	lbm	
Internal Energy of fluid 1 start	U_11	1619.3	BTU/lbm	
Internal Energy of fluid 1 end	U_21	1288.24	BTU/lbm	
Enthalpy of fluid 1 end	H_21	1288.24	BTU/lbm	
	sum_W1i_U1i	4268.691		
	sum_W2i_U2i	9.904921		
	sum_W1i_W2i_H2i	3386.068		
Trapped energy	Delta_Ee	926.0742	HP-sec	

The total trapped energy is 926.07 hp-sec (691 kW-sec).

- Total rotor energy:

$$E_{\max} = E_T + E_s + E_v + E_e \tag{6}$$

Description	Symbol	Value	Units
Total Rotor Energy	Emax	118748	HP-sec

So the total rotor energy is 118,748 hp-sec. (88,586 kW-sec).

- Maximum rotor speed (rpm):

$$N_{\max} = \sqrt{\frac{E_{\max}}{k_2 \times WR^2}} \tag{7}$$

Description	Symbol	Value	Units
Conversion Factor, Nmax	K_4	3.23E+06	
Max Attained Rotor Speed	Nmax	4631.346	RPM

Converting the total rotor energy into speed, gives the maximum rotor speed as 4631 rpm.

Rotor Max Allowable Speed	Nmxa	4675	RPM

If a 10 percent rise above the trip speed is allowed, then this response is acceptable.

It is interesting to look at the effects of varying the control delay from 0 to 1 second:

Control Delay time (sec)	Energy added during Control Delay Delta_Es (HP-sec)	Maximum Rotor Speed (RPM)	Comments
0	0.00	4485.27	With no control delay the speed rise is approximately 235 RPM.
0.35	10321.72 (7700 kW-sec)	4688.51	A control delay of 0.35 seconds results in a maximum speed in excess of the maximum allowable speed.
0.75	22117.97 (16500 kW-sec)	4910.48	A 0.75 second delay results in a maximum speed in excess of the 127% maximum limit (4907 RPM).
1	29490.62 (22000 kW-sec)	5044.26	A 1 second delay results in a speed rise of 794 RPM in excess of the trip speed. If we assumed this trip speed was 110% of normal maximum operating speed, then this equates to > 130% rated speed.

Shown graphically:

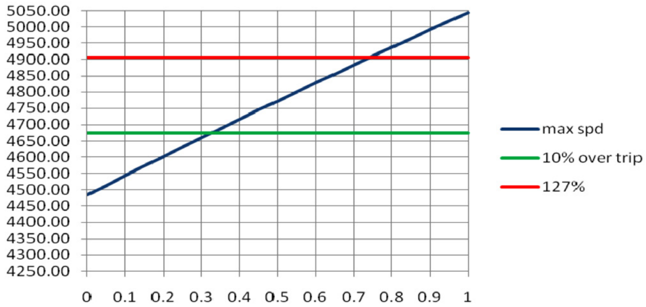

Figure 12. Maximum Rotor Speed (RPM) Versus Control Delay Time (Sec).

NOTE: Control delay refers to the total delay from the actual trip point being reached and the stop valves starting to close. This includes the response delays associated with overspeed detection, solenoid actuation, trip header dump, and any mechanical or hydraulic interfaces to trip valves.

The point of this exercise is to illustrate that conditions change very rapidly for variations in the system response that would not be detectable to an operator without specific instrumentation. This variation in the response of the trip system, if noticed at all, might be considered a minor "sticking" and be ignored during testing where the trip speed was being checked. The results in a real fault situation could be catastrophic.

It is also interesting to note that although the maximum speed rise is not precisely proportional to the control delay time, it is not a bad approximation to use the initial acceleration times the control delay to predict the maximum rotor speed.

Example 2

A 224 MW turbine with a rotor inertia of 308,000 lb-ft^2 (12979 kg-m^2) and a trip speed of 4000 rpm.

- Acceleration (rpm/s):

$$\alpha_t = k \times \frac{P_{g(\max)}}{N_T \times WR^2} \tag{1}$$

Description	Symbol	Value	Units
Conversion factor, accel	k	1.614E+06	RPM^2-lb-ft^2/(HP-s)
Turbine max net power	Pgmax	300268.1	HP
Trip speed	Nt	4000	RPM
Turbine rotor inertia	WR_2	308000	lb-ft^2
Acceleration, initial	Alpha	393.4	RPM/sec

The instantaneous acceleration is 393.6 rpm/sec.

- Rotor kinetic energy:

$$E_T = k_2 \times WR^2 \times N_T^{\,2} \tag{2}$$

Description	Symbol	Value	Units
Conversion factor, Et	K_2	3.10E-07	
Kinetic Energy at trip speed	Et	1527680	HP-sec

The rotor kinetic energy at trip speed is 1,527,680 hp-sec (1,139,649 kW-sec).

- Ctrl time delay energy:

$$\Delta E_s = T_s \times P_{g(\max)} \tag{3}$$

If the control time delay is 0.25 seconds:

Description	Symbol	Value	Units
Control time delay	Ts	0.1	sec
Control Delay energy	Delta_Es	30026.81	HP-sec

The energy added during the control time delay is 30,026.81 hp-sec (22,400 kW-sec).

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- Stop valve closure:

$$\Delta E_v = f \times T_v \times P_{g(\max)} \tag{4}$$

NOTE: In more complex systems where multiple stop valves are employed, such as reheat steam turbines, these must be considered, too.

The following represents the stop valve and the energy input to the high-pressure (HP) section:

Description	Symbol	Value	Units
Valve flow characteristic	f	0.84	fraction
Valve closure time	Tv	0.29	sec
Power (hp)	Pg(hp)	80428.95	HP
Valve delay energy (sv)	Delta_Ev	19592.49	HP-sec

The following represents the reheat stop valve and the energy input to the intermediate-pressure (IP) and low-pressure (LP) sections:

Description	Symbol	Value	Units
Valve flow characteristic	f	0.88	fraction
Valve closure time	Tv	0.13	sec
Power (ip lp)	Pg(ip lp)	219839.1	HP
Valve delay energy (rsv)	Delta_Ev	25149.6	HP-sec

The total energy added during the stop valve and reheat stop valve closure is 19,592.49 + 25,149.6 = 44,742.09 hp-sec (14,616 + 18,762 = 333,778 kW-sec).

- Trapped energy:

$$\Delta E_e = k_3 \eta \left(\sum W_{1i} u_{1i} - \sum W_{2i} u_{2i} - \sum (W_{1i} - W_{2i}) h_{2i} \right) \tag{5}$$

In this example there are three principle volumes to consider:
• Between stop valve and first stage nozzle,
• Between reheat stop valves and reheat turbine, and
• Crossover pipes between IP and LP turbines:

Description	Symbol	Value	Units
Conversion factor, Delta Ee	K_3	1.41485	HP-sec/BTU
Turbine isentropic efficiency	Eta	0.8	fraction
Weight of fluid 1 start	W_11	150	lbm
Weight of fluid 1 end	W_21	43.5	lbm
Internal Energy of fluid 1 start	U_11	1328	BTU/lbm
Internal Energy of fluid 1 end	U_21	1172	BTU/lbm
Enthalpy of fluid 1 end	H_21	1275	BTU/lbm
Weight of fluid 2 start	W_12	175	lbm
Weight of fluid 2 end	W_22	0.47	lbm
Internal Energy of fluid 2 start	U_12	1364	BTU/lbm
Internal Energy of fluid 2 end	U_22	857	BTU/lbm
Enthalpy of fluid 2 end	H_22	925	BTU/lbm
Weight of fluid 3 start	W_13	130	lbm
Weight of fluid 3 end	W_23	0.5	lbm
Internal Energy of fluid 3 start	U_13	1183	BTU/lbm
Internal Energy of fluid 3 end	U_23	936	BTU/lbm
Enthalpy of fluid 3 end	H_23	965	BTU/lbm
	sum_W1i_U1i	591690	
	sum_W2i_U2i	51852.79	
	sum_W1i_W2i_H2i	422195.3	
Trapped energy	Delta_Ee	133156.6	HP-sec

Additionally, extractions and turbine shell contribute 29,625 hp-sec (22,100 kW-sec) (from manufacturer). The total trapped energy is 162,781 hp-sec (121,435 kW-sec).

- Total rotor energy:

$$E_{\max} = E_T + E_s + E_v + E_e \tag{6}$$

Total Rotor Energy	Emax	1765230	HP-sec

So the total rotor energy is 181,6603 hp-sec (1,355,186 kW-sec).

- Maximum rotor speed (rpm):

$$N_{\max} = \sqrt{\frac{E_{\max}}{k_2 \times WR^2}} \tag{7}$$

Description	Symbol	Value	Units
Conversion Factor, Nmax	K_4	3.23E+06	
Max Attained Rotor Speed	Nmax	4302.556	RPM

Converting the total rotor energy into speed, gives the maximum rotor speed as 4622 rpm.

Rotor Max Allowable Speed	Nmxa	4400	RPM

If a 10 percent rise above the trip speed is allowed, then this response is acceptable. NOTE: 127 percent of rated speed is 4572 rpm.

Example 3

A 5.2 MW turbine with a rotor inertia of 1836 lb-ft^2 (77.4 kg-m^2) and a trip speed of 4158 rpm.

- Acceleration (rpm/s):

$$\alpha_t = k \times \frac{P_{g(\max)}}{N_T \times WR^2} \tag{1}$$

Description	Symbol	Value	Units
Conversion factor, accel	k	1.614E+06	RPM^2-lb-ft^2/(HP-s)
Turbine max net power	Pgmax	7000	HP
Trip speed	Nt	4158	RPM
Turbine rotor inertia	WR_2	1836	lb-ft^2
Acceleration, initial	Alpha	1479.9	RPM/sec

The instantaneous acceleration is 1480.9 rpm/sec.

- Rotor kinetic energy:

$$E_T = k_2 \times WR^2 \times N_T^{\,2} \tag{2}$$

Description	Symbol	Value	Units
Conversion factor, Et	K_2	3.10E-07	
Kinetic Energy at trip speed	Et	9840.19	HP-sec

The rotor kinetic energy at trip speed is 9840 hp-sec (7341 kW-sec).

- Ctrl time delay energy:

$$\Delta E_s = T_s \times P_{g(\max)} \tag{3}$$

If the control time delay is 0.1 seconds:

Description	Symbol	Value	Units
Control time delay	Ts	0.1	sec
Control Delay energy	Delta_Es	700	HP-sec

The energy added during the control time delay is 700 hp-sec (522 kW-sec).

- Stop valve closure:

$$\Delta E_v = f \times T_v \times P_{g(\max)} \tag{4}$$

NOTE: In more complex systems where multiple stop valves are employed, such as reheat steam turbines, these must be considered, too.

If the stop valve closure time is 0.4 seconds and the flow characteristic is 0.5:

Description	Symbol	Value	Units	
Valve flow characteristic	f	0.5	fraction	Linear valve = 0.5
Valve closure time	Tv	0.4	sec	
Valve delay energy	Delta_Ev	1400	HP-sec	

The energy added during the stop valve closure is 1400 hp-sec (1044 kW-sec).

- Trapped energy:

$$\Delta E_e = k_3 \eta \left(\sum W_{1i} u_{1i} - \sum W_{2i} u_{2i} - \sum (W_{1i} - W_{2i}) h_{2i} \right) \tag{5}$$

In this example trapped energy is negligible. The total trapped energy is 0 hp-sec (0 kW-sec).

- Total rotor energy:

$$E_{\max} = E_T + E_s + E_v + E_e \tag{6}$$

Description	Symbol	Value	Units
Total Rotor Energy	Emax	11940	HP-sec

So the total rotor energy is 118,748 hp-sec (88,586 kW-sec).

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- Maximum rotor speed (rpm):

$$N_{\max} = \sqrt{\frac{E_{\max}}{k_2 \times WR^2}} \quad (7)$$

Description	Symbol	Value	Units
Conversion Factor, Nmax	K_4	3.23E+06	
Max Attained Rotor Speed	Nmax	4583.218	RPM

Converting the total rotor energy into speed, gives the maximum rotor speed as 4631 rpm.

Rotor Max Allowable Speed	Nmxa	4573.8	RPM

If a 10 percent rise above the trip speed is allowed, then this response is not acceptable.

If the control time is varied the following maximum rotor speeds:

Control Delay time (sec)	Energy added during Control Delay Delta_Es (HP-sec)	Maximum Rotor Speed (RPM)
0	0.00 (0 kW-sec)	4446.80
0.05	350.00 (261 kW-sec)	4515.51
0.1	700.00 (522 kW-sec)	4583.18
0.15	1050.00 (783 kW-sec)	4649.87
0.2	1400.00 (1044 kW-sec)	4715.62
0.25	1750.00 (1306 kW-sec)	4780.46
0.3	2100.00 (1567 kW-sec)	4844.43

With this smaller turbine, it is interesting to note that a control delay time of only 0.3 seconds would result in the 127 percent speed being exceeded.

TEST METHODOLOGY

Regular, periodic testing is the only way to ensure a protection system is working. In some cases, without exercising the system, mechanical components may become stuck rendering the protection system inoperable. Regular testing of stop or trip valves is generally recommended on a weekly or monthly basis where possible. Additionally it is recommended to shut down with the trip valve(s) whenever possible.

One of the largest owner/operators of power generation equipment requires annual testing on large machines. For mechanical bolts a minimum of two tests are required. If the trip speed of the second test is higher than the first, then a third test is required. Electronic systems can be tested at lower speed. Where there are exceptions to this rule the specific test requirements would have to be recommended by the turbine OEM.

Another very large owner/operator allows low speed trips for testing but requires a "full stress" (full speed) test after any front standard work that could affect speed sensors, wiring, and gap.

Specific test recommendations will come from OEMs and insurance carriers and each owner/operator will establish test requirements for a specific unit. There are a number of studies that discuss testing frequency based on recorded failure probability either across the industry or for a given site. These procedures should not be considered static. As more experience is gained with the unit or as the unit is modified, these procedures, test scope, and test intervals, should be reexamined. Also, as root causes of incidents of all units are examined and shared, new best practices may be recognized.

Low Speed Testing

On generator applications, the full overspeed test point exerts tremendous forces on the rotating components of the turbine and generator. This can lead to increased maintenance costs in the long run so there is some incentive to perform the overspeed system test at a lower rpm.

On compressor or pump applications, it is often (but not always) physically impossible to achieve the overspeed setpoint with the compressors or pumps connected. So users perform overspeed tests with the units uncoupled. Uncoupling and recoupling is a labor intensive process. When uncoupled, the rotational inertia of the system is greatly reduced. This means the control dynamics for the normal system and the uncoupled system are different. This may require entering a different set of dynamics or retuning the governor system for testing (and making sure the original governing system dynamics are restored when the test is complete). In some cases special functions have been added to the governing system to support additional modes for uncoupled testing. Again, there is a risk that the modes of operation be specified or selected correctly.

In both of the above cases there are compelling reasons to perform at least some of the tests of the overspeed protection system at a lower speed. But, insurance carriers and corporate testing requirements must also be met. Some sites require validation of the trip system functionality at full speed but then allow lower speed tests provided no changes are made in the system, particularly with the speed pickup, brackets, or sensing gear that could affect the ability to properly sense speed. There are some specific case examples that demonstrate the value of full speed testing.

Trip System Testing While Running

Many units run for long periods of time, multiple years, between outages. So there may be extended periods of time where actual trip testing cannot be performed. To ensure reliability and availability of the overspeed protection system, periodic partial system testing is performed. This may include confirming mechanical/hydraulic operation of a part of the trip system or performing some simulated test on part of the electronic system while the trip circuit is partially blocked. Ideally the systems should be designed with enough redundancy that protection is not compromised during the test process (but this is not always the case).

In some systems it is possible to do some level of valve testing either at rated or reduced load conditions. While this is valuable, and in some cases necessary to ensure the operation of the valve, it does not guarantee a proper seating of the valve. So there is still a case for a full functional trip test (although this need not be from full trip speed).

Risks of Testing

An overspeed event requires three components:

- Loss of load,
- Failure of the control, and
- Failure of the overspeed protections system.

During an overspeed test the unit is not loaded and the control may be partially compromised by operating above normal operational speeds and, where driving equipment is uncoupled, with a significantly reduced inertia. The overspeed test requires putting the protection system into an abnormal state—either by changing setpoints, overriding normal governor limits, and possibly operating with a different set of system dynamics. So the last line of defense may be the very protection system that is being tested to confirm its functionality. A significant number of overspeed incidents, by some estimates nearly half, occur during testing. Obviously some preliminary testing is required prior to performing a full overspeed test. This may include performing a low speed test prior to full speed testing.

After a successful test is performed it is equally important that the system be restored to the normal operating state. A second set of tests, such as simulated speed testing if the setpoint was changed, should be performed to confirm the system is correctly restored to its normal operating conditions.

INCIDENTS AND ROOT CAUSE ANALYSIS

In one case a boiler feedpump turbine (BFPT) was tested at less than full overspeed trip for years. The setpoint was dialed down for the test and then reset. A safety audit required a full speed test. At 8000 rpm the unit did not trip and an emergency trip was selected. Next year the trip failed again. The third year the controls group got involved. The trip system electronics and program appeared to be correct. When the turbine was tested, above 6200 rpm the overspeed device started to miss pulses and the sensed speed dropped but not enough to trigger loss of speed sensed logic. Work was done by the OEM on the front standard and on the sensing wheel and the unit worked. Although the exact cause of the problem was not identified, one thought is that the rotor shifted slightly at high speeds and that only in those conditions would the speed signal be partially lost. Tests with a frequency generator to simulate speed would not have exposed this problem.

In another case a BFPT was tripped for no apparent reason. Just prior to this, another BFPT had tripped and the distributed control systems (DCS) increased the demand to the second BFPT. It picked up speed and tripped. It was not immediately obvious that an overspeed had occurred. On analysis of the data it was recognized that it tripped at the mechanical overspeed trip setpoint. The electronic governor setpoint was scaled as 0 to 10,000 rpm when the operating range was only around 8000 rpm. The controllers protected overspeed were set to 105 percent of this total range (not 105 percent of the normal maximum operating point). On this unit additional instrumentation was added to annunciate the overspeed bolt trip and the controller was rescaled to the proper operating range.

It is critical that the industry look to and share incidents and root cause analysis to determine how to prevent the repetition of mistakes.

RESPONSE TIME TESTING

API 612 (2005) 16.3.4.6 states, "The response time of the overspeed shutdown system shall be recorded to confirm compliance with the requirements of 12.3.1.1." But this is in a list of optional tests that may be performed as shop tests and not a regular test requirement.

A great deal of emphasis is placed on overspeed testing but this is done under controlled conditions. Speed is raised slowly to the trip point. This confirms the overspeed trip setpoint but does nothing to confirm the dynamic response of the trip system. This factor is as critical as the setpoint.

The authors have shown that the response time of the entire trip system is critical to the protection of the turbine and that relatively small variations (undetectable to an operator) are very significant. Since the timing of these electrohydraulic circuits is so important and the systems so complex, the need to periodically record the trip time and then trend those response times to identify any progressive degradation in the system seems obvious. In the past, achieving a resolution of 1mS to evaluate the response time of a trip system would require specialized equipment but today this is readily available from sequence of event (SOE) cards in DCS systems and programmable logic controller (PLC) systems.

Periodic trip system response time testing, as with the overspeed setpoint testing, is needed to ensure the overspeed prevention system is operating as designed and that the required level of equipment and personnel protection is provided.

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