UNDERSTANDING AND PREVENTING STEAM TURBINE OVERSPEEDS

by

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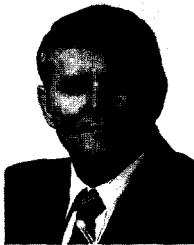
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ABSTRACT

Destructive overspeeding of steam turbines due to poor component design, inadequate maintenance, and improper operating practices are all too common. The total cost of a turbine runaway can run into the millions of dollars and often include lives lost. Determining the root cause of the failure afterwards is confusing. Because so many variables contribute to an uncontrolled turbine, assessing all of them is difficult. Design ideas of the trip systems for the many turbine manufacturers are varied and span many years. Frequently, the causes of an overspeed disaster are not fully understood even after an exhaustive study is completed. Little hard data are available because things happen very fast. The accounts of eyewitnesses are highly unreliable, because so much happened that one individual could not see it all. No one wants to talk about the event due to legal entanglements. Most conclusions must be based on an event reconstruction done by looking at some broken parts, with limited information from data sources and witnesses. The unfortunate result is little movement among either users or manufacturers toward improving equipment to reduce the possibilities of an overspeed episode.

Engineering design and application of the steam turbine and its appendage devices are important factors in the reliability of the machinery train. Lack of training for maintenance and operating personnel are also big factors in the possibility of a runaway. Nearly half the more than forty destructive overspeed failures encountered by the authors occurred during periodic testing activities. Considering that only a few hundred turbines are involved in the authors' experience, overspeed failure represents a major problem.

INTRODUCTION

Over the last few years, planned maintenance (PM), predictive maintenance (PdM), and reliability centered maintenance (RCM) have been developed to promote reliability, reduce cost, and eliminate catastrophic failures. Very little information is available on occurrence frequency or severity of overspeed episodes. However, it is believed that these improvement programs have not reduced steam turbine overspeed problems. Data and experiences of one insurance company in several areas of failure for steam turbines were presented by Clark [1]. Examined more closely herein are the characteristics of a steam turbine's components and evaluation as to possible involvement in loss of control of a machine. These characteristics may seem small but can loom large in some failures. Mostly, U. S. systems design will be discussed, although the principles apply equally to European and Asian designs.

TURBINE DESIGN CHARACTERISTICS

Many design characteristics of the steam turbine can influence the tendencies toward uncontrolled speed and reliability problems. These factors can determine how quickly the turbine can change speed and the maximum speed it can attain. Weaver [2] calls all these characteristics time-constant factors. Other areas beyond the time-constant factors can greatly affect the tendency for a steam turbine to overspeed. Two distinct turbine designs affect the problems encountered as shown in Table 1. Also, the design, construction, and maintenance of appendages to the turbine create problems that may result in the turbine overspeeding.

Table 1. Comparisons of Turbine Designs.

Turbine Parameter	Impulse Turbine	Reaction Turbine
Steam Pressure	Decreases in every nozzle, remains constant in moving and fixed blades	Decreases in every row of fixed and moving blades
Absolute Steam Velocity	(1) Increases in nozzles (2) Decreases in moving blades (3) Practically constant in fixed blades	(1) Increases in fixed blades (2) Decreases in moving blades
Theoretical Efficient Speed	For simple impulse, blade speed should be about 1/2 steam velocity; for velocity—compounded (Curtis) with two rows of moving blades, the blade speed should be about 1/4 steam velocity	For pure reaction, blade speed should be about steam velocity

The velocity of the steam exiting the nozzles is the major factor in the tendency to accelerate the turbine rotor. In simple terms, the velocity-compounded (Curtis) type single stage impulse is easier to overspeed, while the reaction turbine is less likely to overspeed. More detailed discussions of these basic design factors of steam turbines are available in the literature [3, 4].

Mechanical or Generator Drive?

The load that the turbine is driving and how fast it can change is another major factor in loss of control. A turbine driving an electric generator can suffer a 100 percent load loss in a few cycles or about 1/20th of a second as shown in Figure 1 [5]. The generator is probably more sensitive to overspeed damage than the turbine driver. Rotor design of the two pole generators makes the securing of the windings in the slots difficult. With the windings held in place by the varnish and a few wedges in the slots, an overspeed can cause rupturing of the heavy retaining rings at each end of the rotor. This will dislodge the rotor windings, causing major damage to the stator as well. In some failures, the stator copper can stop the rotating assembly almost instantly, causing the shaft to snap, leaving the turbine completely unloaded from that point.

By contrast, an instantaneous 100 percent load loss is less likely to occur in a mechanical drive machine. Surge in a driven compressor or loss of suction in some pumps causes major changes in the load, but generally about 40 to 60 percent, not the 100 percent range. In addition, the load change is a slow process, by comparison, to the electrical load loss. With a poor governing system, these smaller, less abrupt, load changes can still result in an overspeed situation. Centrifugal compressors generally have small rotors with less inertia than the turbine rotor. Occasionally, the effect of "gas mass inertia" on a turbocompressor can be significant. Although the gas flow is interrupted, the gas immediately in front of the machine continues to impart energy to the system, due to its inertia, and can contribute to an overspeed situation. Evaluating the impact of this effect on an overspeed situation is very difficult.

Complete shattering of a gear coupling sleeve, flexing elements of nonlubricated designs, or the spacer on any type coupling for a mechanical drive machine can cause a rapid 100 percent load loss.
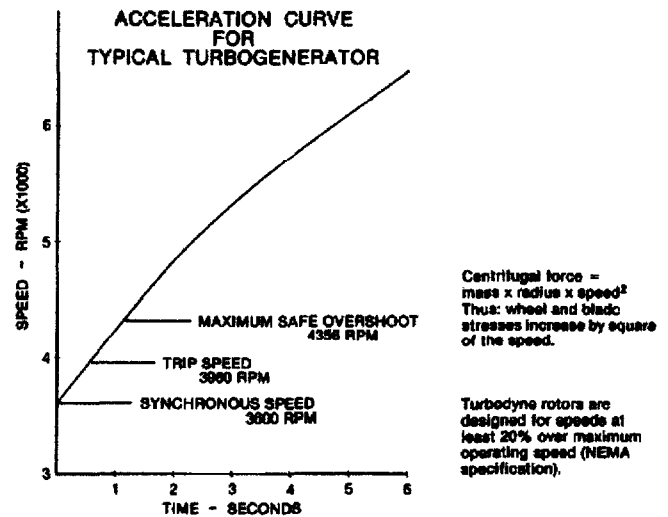

Figure 1. Simplified Acceleration Curve for a Typical Turbine Generator [5].

To reduce problems, correct application and installation of the coupling is vital. Shaft couplings are rated by their manufacturers in several ways: how much torque they can transmit, how much misalignment they can accommodate, and the maximum speed at which they can operate. Still, manufacturers, buyers, or various organizations often use qualifiers, or factors that significantly reduce the published capabilities of couplings, when the published ratings cannot be used at face value. Users and application engineers must be knowledgeable about the process of coupling selection and installation. Avoiding the pitfalls of meaningless or misleading ratings, and the possibility of selecting an undersized coupling, is an important step toward reducing overspeeds [6, 7, 8]. The user should question the veracity of *couplings catalogs* that contain ratings that cannot be used even under the best conditions.

Case Histories

In one episode, an aluminum alloy coupling spacer was used to reduce the overhung moments of a gear coupling in an ammonia plant for a two-case air compressor train, during the early days of understanding (or misunderstanding) rotordynamics. The aluminum spacer, between the turbine and the first compressor case, suffered a fatigue type failure. An early day, custom designed, analog electronic governor retained control of the turbine and an overspeed situation was not encountered, though the load loss was 100 percent and almost instantaneous. This is a good example of how a well designed and upgraded electronic governor can greatly improve the safety of operating a steam turbine.

In another situation, the mechanical/hydraulic governor on a 1500 hp turbine did not control the turbine's speed after the teeth, on the pump half of a grease lubricated gear coupling, wore down. The coupling between the turbine and the pump was a Purchasing Department special bargain. A few hundred dollars were "saved" by purchasing a bar stock, carbon steel, gear coupling that had about two-thirds the torque transmitting capacity of the required forged alloy steel, quality coupling. Reduced coupling capacity resulted in rapid wear of the gear teeth, although a quality lubricant was used. The turbine was severely damaged during the overspeed and had to be replaced.

STEAM TURBINE CONTROL BASICS

Overspeed prevention is best accomplished by not losing control of the turbine. Energy used in a steam turbine is produced in a boiler independent of, and external to, the machine. Control of the steam flow to the nozzles depends on the turbine itself. Understanding the original equipment manufacturer's (OEM)

control design intentions is very important. Speed control systems for steam turbines consist of three basic elements: sensing, transmitting, and correcting. Sensing elements include fly ball weights, electric generators, and positive displacement pumps. Transmitting elements may be mechanical linkage, hydraulic or pneumatic pressure, electrical signals or, as is most common, a combination. Sometimes an amplifying device such as a pilot, convertor, or servomotor is necessary to boost the sensing element signal to a point where it can do useful work. The correcting element of the governors' system is the valve or valves that control the flow of steam to the turbine.

Fundamentals of Inlet Control Valves

The flow of steam into the steam turbine is regulated by inlet control valve(s). Controlling the turbine steam flow with a minimum pressure drop through the control valves is very important. Any pressure drop caused by the steam flow through the control valves is a loss of thermodynamic energy. The valve design will influence response and how well speed control is accomplished.

Single Valve Turbines

In smaller turbines, all the steam flows to the turbine nozzles through a single, double seated valve, characterized by high flows with low lift and low unbalance forces. Quantity of steam passing through the nozzles is regulated by throttling the steam pressure, as shown in Figure 2 [9]. At low loads, the turbine will be very inefficient because of the energy dissipated by excessive throttling. To improve this situation, one or more hand valves can be provided. Each hand valve will remove some nozzles from service, reducing the available nozzle area and limiting the wasteful throttling.
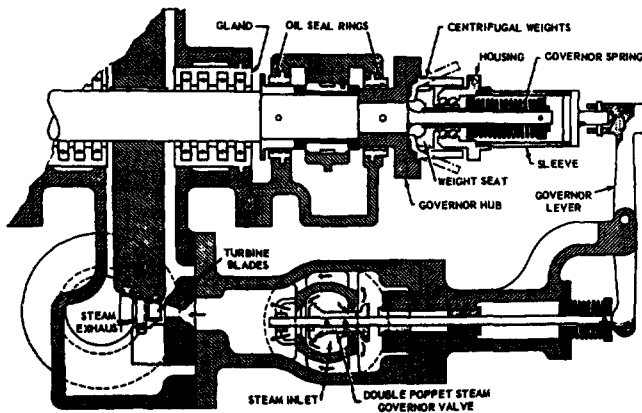
Figure 2. Small Single Stage Velocity-Compounded Turbine with Mechanical Governor [9].

Multivalve Turbines

As the turbine size increases, four to eight valves are normally used to divide the steam flow to nozzle groups. These valves have superior flow-lift characteristics, as compared with the double-seated single valve. Steam flow through each valve is proportional to the lift of the valve from its valve seat. An increase in steam flow through the turbine is accomplished by opening the valve on successive nozzle groups, increasing the nozzle area. Valves function in sequence. Only one valve is throttling or controlling at a time. All other inlet valves that are passing steam are operating at a minimum pressure drop. Two styles of lifting mechanisms are employed to move the valves.

Lifting Bar Type of Control Valves—On some multivalve machines, venturi valves are lifted by a bar mechanism, as shown in Figure 3. The valves have different stem lengths to allow them

to be lifted from their seats, in the proper sequence, as the entire bar is moved. Motive force to move the valves in the closing direction comes only from the weight of the valve itself and the unbalanced steam pressures across it.
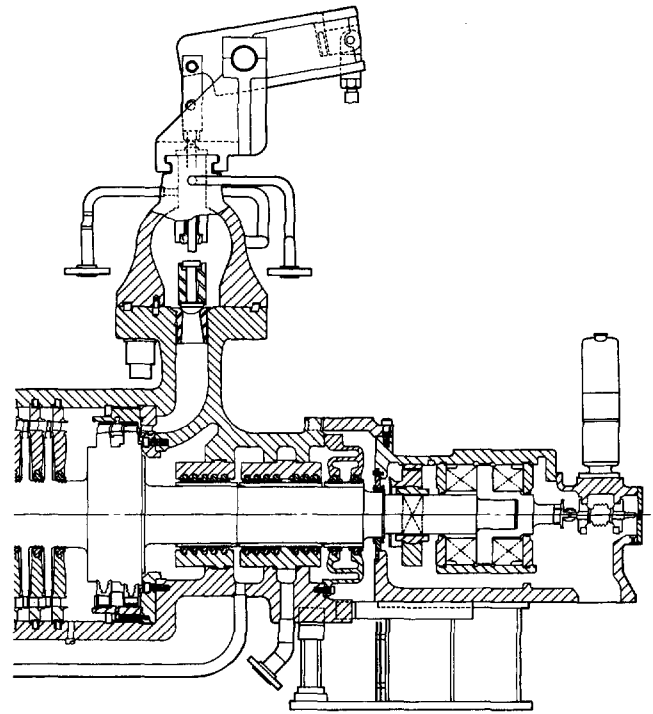
Figure 3. Larger Turbine with Governor Worm Gear Drive and Dished Disk Overspeed Trip [10].

Cam Type of Control Valve—Cams mounted on a shaft are another lifting arrangement for valves as shown in Figure 4. The mechanical advantage of the cam geometry produces a large lifting force. Cams permit varying the rate of valve lift vs angular camshaft travel, so the nonlinearity of the valve-flow-lift characteristic can be corrected by the proper cam shape. The camshaft has a gear pinion keyed to it. A rack or gear segment moved by the hydraulic servomotor provides the torque and positioning means required to operate the camshaft and the valve gear. Cam shapes produce a governing point lift of the valve in about 40 degrees of cam rotation. Cam-lift valve gear uses spring force on the valve stem plus the valve-disk steam load to close the valves. Spring force is enough to counteract the valve steam unbalance force plus one hundred to five hundred additional pound biases at the valve-cracking point lift. In addition, some designs have a pin on the cam that can provide a "knock down" force to aid in closing the valves.
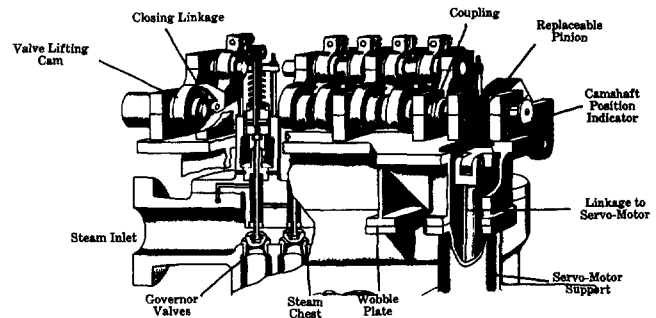
Figure 4. Cam Lift for Multivalve Turbines [10].

In multivalve installations, the force required to operate the valve racks can approach two and a half tons for a mechanical drive turbine. Higher pressure hydraulic oil can be used to provide the motive force but multiplying action slows valve movement response time. The type of valve lifting device can influence the tendency for a machine to approach overspeed.

Governor Classifications

Governors are classified by the National Electrical Manufacturers Association (NEMA) SM-23 into four groups, as shown in Table 2. Speed variation permitted by a governor is also influenced by the inlet valve design, as discussed previously.

Table 2. Characteristics of Governor Classifications.

Class	Max Speed Regulation Percent	Max Speed Rise Percent
A	10	13
B	6	7
C	4	7
D	0.5	7

A partial verbal description of the basic types of governors fitting the above groups is as follows:

● Class A—Mechanical shaft—The familiar fly weight type is mechanically connected to the steam valve as shown in Figure 2. It develops small forces (≈20 in-lbs) to move the governor valve, limiting its use to smaller turbines. Some self-contained hydraulic and electronic governors also fall within this classification.

● Class B—Direct acting orifice—A shaft driven, positive displacement type oil pump delivers pressure to a spring loaded diaphragm connected to the governor valve stem. Since the delivered oil pressure is directly proportional to shaft speed, control can be accomplished. This design is seldom used in modern turbines.

● Class C—Mostly self contained fly weight type governors on some mechanical drive turbines and generator drives as well.

● Class D—Oil relay—Built to use lube oil pressure or a separate, higher governor oil pressure, this design is usually integral to the turbine and designed by the turbine manufacturer. Turbines built in the 1950s to the 1970s used this type of governor. Maintenance of this governor is difficult and time consuming because of the need to correct wear in the many linkages, pins, bell cranks, etc.

● Class D—Precision oil relay—This design uses its own hydraulic oil system and is made by a governor only manufacturer. One U. S. manufacturer dominates the market. It is a better design than the governor made by the OEM, but it is still subject to wear and needs to be checked out on a test stand on one to two year frequencies. A major problem arises, because the mechanical governor operates best at speeds of about 800 rpm to 1200 rpm. Since the turbine generally operates at higher speeds, a step down gear box is required to drive the governor as shown in Figure 3 [10]. In addition, a coupling between the gear system and the governor input shaft is required. Lubrication of the gear mesh, the coupling, and the governor becomes critical. Any vibration problems in the turbine can destroy the gears or the coupling quickly.

● Class D—Electronic—The electronic governor speed measurement signal is generated by a magnetic pickup in proximity to a toothed wheel or gear mounted on the turbine shaft, as shown in Figure 5. An I/P controls the output air (normally three to 15 lb) that goes to the actuator on the inlet steam valve(s) of the turbine. The design can have redundancy and require little

maintenance. Testing and other checks can be done with the steam turbine in full operation, with little chance of losing control and placing the turbine at risk.
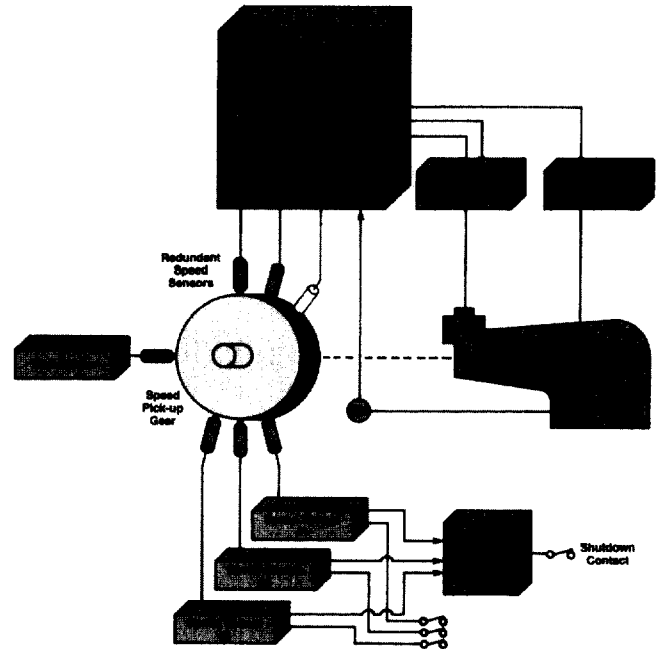

Figure 5. Electronic Governor and Overspeed Trip System [5].

The last two governor types as modified [11] are compared in Table 3. Note that the term "minimum governor" has little importance for an electronic governor. With the other governor designs, the turbine must be started on hand control up to about 60 percent turbine design speed when enough force is developed to move the governor valve(s). The electronic design controls speeds over almost the entire range.

Upgrading to a well designed electronic governor to improve speed control is a major step toward preventing a turbine overspeed. The mechanical governors can only sense speed and control the turbine at a predetermined speed. Also, as mechanical/hydraulic control (MHC) systems designed by the OEM become more obsolete, availability of spare parts becomes a compelling reason for modernizing controls.

As outlined in Table 3, the mechanical aspects of the older designs are quite complex, resulting in a mechanical/hydraulic (MHC) governor life reliability expectancy of only one to two years without a major overhaul. These repairs are a big expense and require a process unit downtime. Besides gains in accuracy and response, an electronic/hydraulic conversion (EHC) eliminates many high-maintenance moving components. Fly weights, worm-gear drives, feedback linkages, levers, pin joints, rod-end bearings, and motor-speed changers are not used. Labor-intensive activities, such as frequent mechanical tunings and instrument calibrations, are not needed.

Digital-based controls offer further advantages over analog-electronic controls by allowing online diagnostics and subsequent modifications to the control system. Another digital advantage is the availability of fault-tolerant control forms that use voting between multiple microprocessors. Finally, microprocessors are not subjected to drifting like discrete analog components are.

Additional improvements should be done when converting to an electronic governor. Replacing the shaft-driven lube oil pump with an electric motor driven pump increases reliability by correcting the marginal performance of the original pump at low speeds. Upgrading the overspeed trip system with an electronic one, based on a two-out-of-three voting hierarchy between microprocessors,

Table 3. Comparative Features of Turbine-Drive Governing Systems (Modified from [11]).

Features	Mechanical/Hydraulic (with Electric Positioner) MHC	Electro/Hydraulic EHC
Speed-set point control	Spring loading device (must be able to produce up to 400 lb)	Relay contacts—analog or digital signals
Active mechanical components (servo sections)	Servo piston, speed piston. Position limiter, manual overrides, overspeed trip device	Servo pistons, relay valves, actuator
Active mechanical components (governing section)	Reducing gears, support bearings, support bearing, speed springs, speed weights and lever assembly, manual speed setter and limiting assembly, positioner drive mechanisms, position limit switches, position feedback assembly.	None
Installation and maintenance	Many moving and wearing parts requiring regular attention. Complex piping systems.	Simplified piping, additional speed pickups, and additional solenoid trips
Speed sensor	Reduction gears, weights, and springs	Noncontacting magnetic pickup or permanent magnet alternator
Speed sensor limits	About 1,200 rpm, requires reduction gears for higher turbine speeds	Essentially unlimited
Range changes	Various combinations of weights and springs	Screw adjustments of a potentiometer, software changes
Usable speed control	60% to 100%	5% to 105%—Criticals blocked out
Transition from startup to operating conditions.	Mechanical transfer from a servo positioner limiter to speed governor.	Electrical actuator controls servo position in response to governor commands, either on a speed control or position limit.
Control oil required	Yes—5 to 10 gpm to a governor pilot valve	Yes—0.05 gpm to an actuator

is also a major step. This arrangement will permit testing of overspeed circuits without overspeeding the turbine, reducing the risks for the turbine and personnel during this activity.

Case History

Failures of the precision oil relay governor gear drive on a turbine (10,000 hp, 8450 rpm, 110 psig to condensing) driving a hydrogen recycle compressor on a hydrocracker occurred three times during the first year of operation. The gear reducer was precision manufactured with silver babbitt bearings, but alignment problems to the turbine rotor caused the failures. In addition, the mechanical/hydraulic governor did not properly control the turbine during unit upsets even when the gear box was functioning correctly. A conversion was made to an inhouse designed and built analog-electronic governor in about four days time. The electronic governor eliminated the need for the gear drive and improved control under all conditions. No further governor problems were encountered with the driver of the compressor train.

OVERSPEED TRIP SYSTEMS

The overall control scheme of a steam turbine consists of an operating speed governor, as discussed previously, and a separate shutdown system to prevent overspeed. Such a system for a mechanical (MHC) governor is shown in Figure 6 [10]. In the event the speed regulating governor fails to control the speed of the turbine, a fraction of a second is all the reaction time available to sense an overspeed, start, and complete the correction process. Overspeed shutdown systems generally begin to function at 110 percent of design speed and *must* function to stop the steam flow in less than one second completely if no damage is to occur, as shown in Figure 1.

Trip valves separate from the governor valve(s) should shutoff the steam supply completely. Speed measurements, set points, and steam control mechanisms for the speed regulating governor and the overspeed trip should be independent of each other [9, 10, 11, 12]. The standard shutdown system consists of three sections:

- A weight, in the main shaft or in a stub attached to it, with spring tension resisting outward movement of the weight. When shaft speed exceeds a desired safe level of about 10 percent, centrifugal action throws the weight outward about ⅛ in against—

- A latching device, either hydraulic or mechanical, which releases and allows a spring to close—
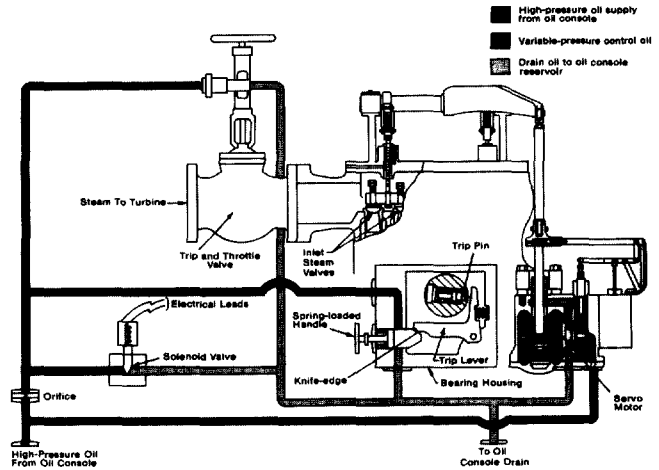
- A special emergency stop valve(s)

Figure 6. Overspeed Trip System—Large Turbine [10].

Overspeed Sensing Devices

Some overspeed sensing devices are:

- Pivoted lever weight—The lever weight is pivoted about a stud and is limited in its movement by a spring as shown in Figure 7 [9]. This type device is used for moderate speeds (up to 4500 rpm) and generally will be found on single stage turbines. Pivoted lever devices can have about ±3 percent spread between the trip speed on one attempt, and the subsequent trip speed on another attempt. Use of a Teflon® bushing on the lever weight pivot and a stainless steel material for the stud is recommended.

- Pin type weight—The pin type weight trip is used for higher speed (>3600 rpm) single stage trips and on the larger multistage turbines as a standard, as shown in Figures 5, 8, and 9. Movement of the trip pin is limited to about 1/8 to 3/16 in. The tripping speed is adjusted by changing the compression on the emergency spring adjusting screw opposite the pin in most systems. In this scheme, the moment balance on the pin is not changed during normal operation. Usually, the weight balance adjustment nut is on the pin side of the shaft and opposite the spring compression nut adjustment. In other schemes, the emergency weight adjusting screw is for altering the balance of the pin weight to change tripping speed. The spring compression is not used to set the tripping speed under this arrangement. A pin type trip of a cartridge design is shown in Figure 9. This permits setting of the trip off the turbine by using an air motor drive, microswitch contacts on the striking arm, and an electronic tachometer. This is a highly desirable feature on a turbine because of the time and labor savings from not having to run the turbine for adjustments.
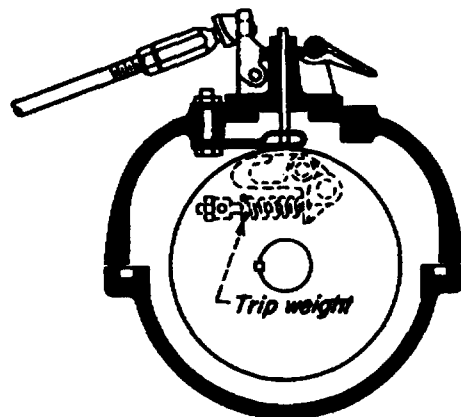

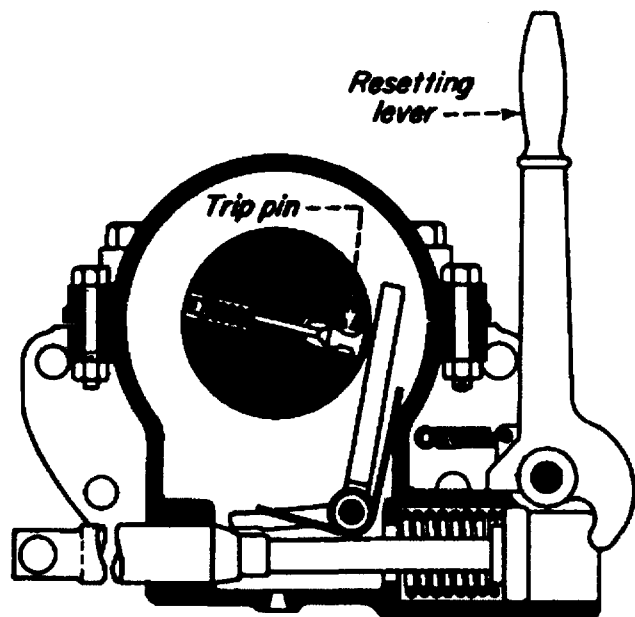Figure 7. Pivoted Lever Overspeed Sensing Device [9].
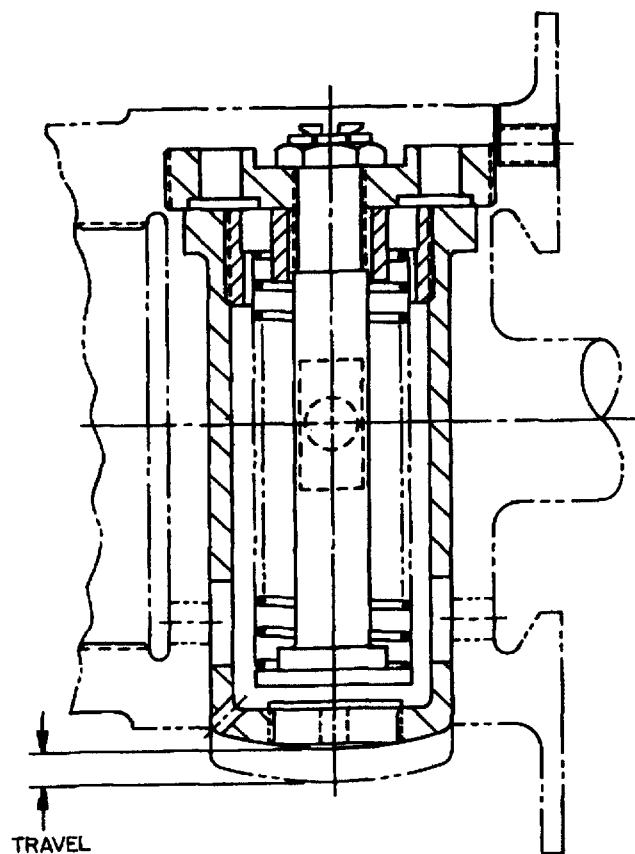
Figure 8. Pin Type Overspeed Sensing Device [9].

Figure 9. Cartridge Type Overspeed Trip Pin.

Trip speed repeatability of the pin type is about ±2 percent. Its major disadvantage is that only a limited amount of pin mass can be obtained in higher speed machines. The diameter of the hole that can be drilled in the shaft limits the pin weight. Also, the spring-weight combination can become unstable at higher trip speeds.

• Eccentric ring design—Similar in concept to the pin type weight. The major advantage of this design is that it permits considerable mass to be added in striking the trip lever of the latching system. This assures that the trip lever is moved quickly. Trip speed adjustments are made by changing the balance of the ring and pin assembly. The design is seen in some Japanese manufactured machines and in at least one U. S. made turbine.

• Dished disk—In this patented design for higher speeds (above 8000 rpm), a dished disk is mounted on the shaft. A series of weights is located around the circumference of the disk to increase the mass as shown in Figures 3 and 10. Dynamic balance of the assembly is inherently stable. At trip speed, centrifugal forces cause the disk to "snap" to the tripped position and strike a trip lever in an axial direction, rather than radial as on the lever or pin designs. The trip speed repeatability is very good, about ±1 percent or less spread.

• Electronic type—Electronic trip systems do not require a mechanical sensing weight and are desirable in higher speed units. An electronic system measures pulses from the governor toothed wheel on the turbine shaft and puts those pulses through a frequency/voltage converter. That output goes to a comparator that switches at the set point and is coupled to a power amplifier that trips an electric relay. The relay opens a solenoid valve that dumps the trip oil header pressure as shown in Figure 5. From here, the system is the same as the mechanical system. Repeatability range is excellent, less than ±1 percent.
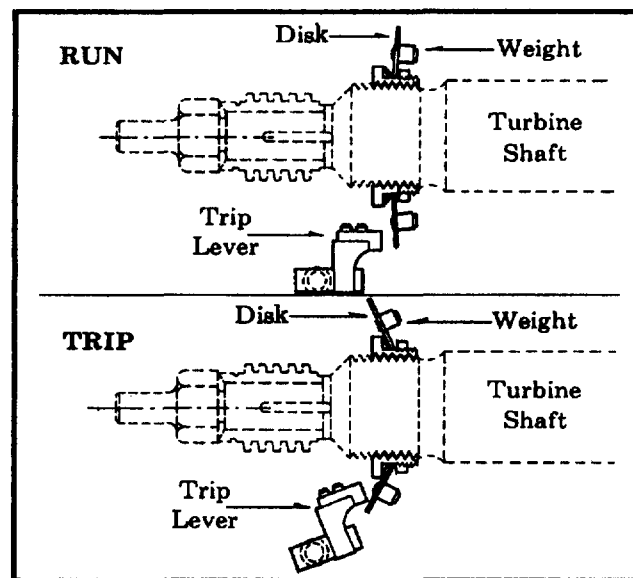

Figure 10. Dished Disk Speed Sensing System [10].

Pros and Cons of the Sensing Systems

The speed sensing systems discussed have previously been used by various manufacturers for many years. While speed regulating governors have undergone continuous improvement, very few changes have been made in the trip sensing devices. As the amount of force that the overspeed trip device must exert to actuate the linkage and/or trip valve increases, so does the stress on the parts. They wear faster, break, or otherwise become inoperable. Some manufacturers use hydraulic relay valves that are floating pistons or dump valves. The intent is to permit the trip mechanism to exert forces of a small amount and still actuate trip valves, requiring large forces to complete the tripping action. Experience suggests that the relay valve is more of a problem than the one it supposedly corrects and, therefore, should not be used.

All of the designs, except the electronic type, require extra shaft overhangs that can be detrimental because of vibration or impact on

critical speed locations on higher speed machines. Long connecting piping runs use lube oil as hydraulic fluid, a fire hazard. Electronic systems require only short shaft overhangs, can be easily tested without running the machine, and give a high degree of repeatability. Wires go directly to the solenoid valve, so much shorter trip header piping is needed. It does, however, require an additional reliable power source and may require explosion-proof classification.

Case Histories

A recent investigation into an overspeed situation that involved a fatality and serious injuries to two other individuals revealed that the governor proper of the turbine had been upgraded from the original design but not the trip system. An all new turbine from the same manufacturer was examined. The overspeed trip system and the butterfly type stop valve were identical in all respects to the one on the nearly 50-year-old turbine involved in the mishap. This lack of improvement in the systems over that many years does not speak well of the machinery industry's and the collective process industry owner's attention to safety matters.

Two of the more destructive runaways encountered by the authors were caused by stuck hydraulic relay valves as discussed under the pin type trip. A stuck hydraulic relay valve prevented closing of the steam inlet valve in a two-case ammonia plant refrigeration train driven by a steam turbine. The relay valve problem was not properly corrected during the investigation of the first runaway. One of those details that can become large, the relay valve struck again, causing another destructive overspeed of the same machine three years later.

Since the hydraulic relay valve used in the trip system is dead ended, a small amount of contamination of the oil can cause the piston to stick. The oil pressure on the trip/throttle valve or the nonreturn valves will not be dumped, and the trip/throttle valve will remain open.

Trip Tolerances

Most of the trip systems have about 20 to 24 trips available before the system components must be disassembled for inspection and repairs. Often, these "good trips" are expended in striving for accuracy in setting the overspeed trip and other testing before the unit is placed in service. To avoid this potential problem, trip repeat tolerance limits should not be too tight.

No single standard for trip tolerances exists. The total system is not designed for greater accuracy than the repeatability on the trip sensing weight. One source, API-617 (Centrifugal Compressors for Petroleum, Chemical, and Gas Service industries, Sixth Edition) calls for trip speeds of 115 percent of *rated* speed for compressor drives. This is 110 percent of maximum continuous speed. Tolerances are not explained clearly. Using the ±2 percent tolerance of a pin type overspeed trip gives ranges as shown in Table 4.

Table 4. Trip Tolerances.

Operating Speed	Trip Speed	Speed Tolerance
3,600	3,960	± 70 rpm (spread 140)
6,000	6,600	± 120 rpm (spread 240)
9,000	9,900	± 180 rpm (spread 360)
etc.		

Two or three repeats within these ranges are adequate. More trips can damage the bolt, linkage, and/or trip/throttle valve, thereby reducing the long term reliability of the overall system.

Other Trip Systems

Solenoid valves can be used to shut the turbine down in response to low lube oil pressure, remote push buttons, or abnormal process conditions as shown in Figure 5.

Latching Systems

An all mechanical overspeed system, shown in Figure 11, is frequently used on single-valve small machines. A trip pin is mounted in the turbine shaft with its center of gravity slightly off center and possible movement of 1/8 to 3/16 in. When a predetermined speed is reached, the unbalanced plunger or pin overcomes a spring force. As it moves outward about 1/16 in, it strikes the trip-lever, causing release of a spring loaded butterfly valve that stops the steam flow.
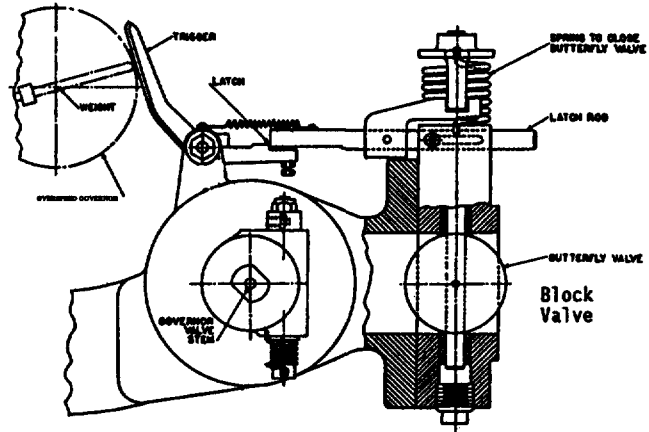

Figure 11. Typical Overspeed Trip System—Small Turbine.

On mechanical/hydraulic systems, the unbalanced plunger or pin strikes the trip lever and causes the draining of the trip circuit oil pressure, as shown in Figure 6. When the trip circuit oil pressure drops below about 45 to 50 percent of normal pressure, a piston-spring combination in the trip/throttle valve becomes unbalanced. Force of a spring and the steam pressure above the valve disk slams shut the trip/throttle valve. The dished disk system accomplishes the striking of the oil release device by movement in an axial direction rather than radial; the remainder of the action is identical. All these actions in the valve *must* occur within about one quarter second, so that the total time of action for the three components of the trip system does not exceed about one second.

Misalignment between the centerline of the trip lever and the trip plunger or pin in the axial plane can cause the system to become inoperative. As the trip pin or plunger moves out, it strikes empty air.

On some turbines, the inlet steam valves can also be closed besides the trip/throttle valve, when any device attempts to trip the machine. This closing action is slow and should not be relied upon to prevent an overspeed if a 100 percent load loss is encountered. It is, however, a step in the right direction.

Case History

The drain line on the trip/throttle valve hydraulic piston must be large enough that the oil moves instantaneously from under the trip piston. In one case, a cost cutting measure by the turbine manufacturer involved reducing the drain oil tubing size from one inch to 5/8 inch in diameter (less than two feet of drain line). The greater restriction to the oil flowing from the latching cylinder increased the trip system functioning time up to about seven seconds. The extra time required to trip could have caused serious damage to the turbine if the situation had not been detected and corrected on the manufacturer's test stand before running of the turbine.

STOP VALVES

The valve that closes off the main steam flow can be one of several different designs:

● Spring loaded butterfly valves (small turbines, Figure 11). Because these valves are not positive shutoff devices, they are sometimes placed *between* the governor valve and the steam chest to reduce how much steam is trapped there after a trip off. This location also simplifies the mechanical latching linkage.

● A spring loaded single or double seated poppet type valve (small to medium turbines).

● An hydraulically latched single seated valve designed by the turbine manufacturer (OEM) to be in the same casting as the single governor or control valve. These valves are generally found in medium turbines of 1000 hp to 2500 hp range.

● A specially designed trip/throttle valve in a separate casting to the turbine governor or control valve(s), as in Figures 12 and 13. It is spring loaded but hydraulically latched, instead of mechanically released (larger machines from all U.S. turbine manufacturers).

● Extraction system nonreturn valves are of a check valve style bolted on the flanges of the turbine shell and activated by the same hydraulic system that closes the trip/throttle valve, as shown in Figure 14.
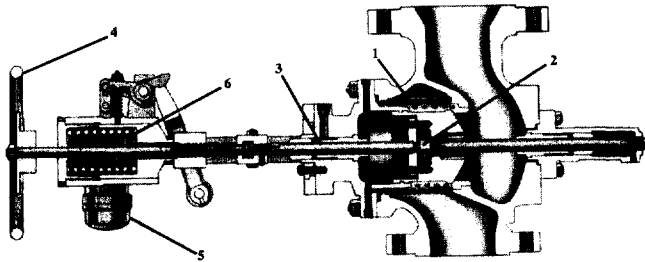

Figure 12. Hydraulic Latch Type Trip/Throttle Valve [15].
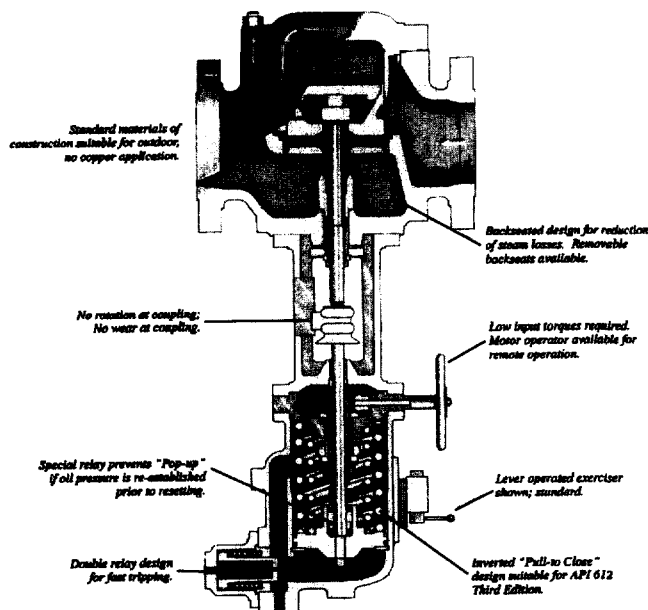

Figure 13. Fully Oil Operated Trip/Throttle Valve Pull onto Seat—Built in Exerciser [15].

No matter what the design, these valves must close completely to stop the steam flow in much less than one second, including the
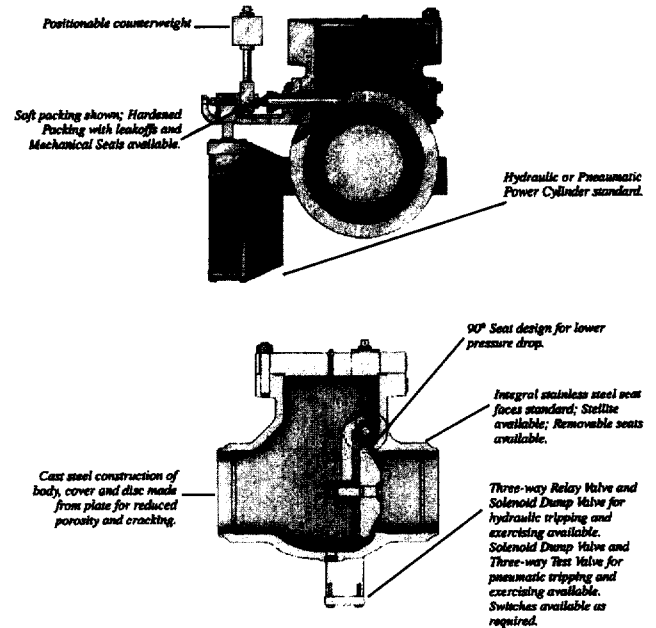

Figure 14. Extraction-Induction/Extraction Swing Disk Nonreturn Valve [15].

speed sensing and initiation of tripping action. To reduce the steam trapped between the trip/throttle valve and the turbine nozzles, the trip/throttle valve should be placed as close as possible to the steam chest. *No extraneous piping should be permitted.*

BASIC TRIP/THROTTLE VALVE DESIGNS

The design idea of the older standard trip/throttle valves, as shown in Figure 12, is that of a globe valve with a stem nut mounted in a frame or bracket that is free to move. A trip/throttle valve mechanism has two separate and distinct functions. First, it acts as a quick closing valve when tripped manually or automatically by the overspeed governor or other safety devices. Second, it provides hand throttling capability for starting and bringing the turbine up to speed. The steam inlet is above, and the steam outlet is below the valve seat. In addition to the compressive force of the spring, steam line pressure positions and holds the valve tight on the seat after the valve is tripped. The designs were originally intended to be installed on machines placed indoors. With little redesign of the valve and its components to compensate for the more hostile environment, most installations are currently located outdoors. Because of the dual functions required of the valve—the tripping action and the throttling action—the stem must be in two pieces in most designs. The stem of the steam shutoff part of the valve does not rotate; it only slides to fulfill the tripping action needed. The upper portion of the trip/throttle valve stem has rotary motion so that it can be positioned within the spring-loaded, hydraulically positioned stem nut to permit throttling. A change of direction and rotation occurs within the split coupling. A hardened steel button or thrust bearing separates the ends of the two stems. In some larger valves, a ball type thrust bearing is used. Keeping the two stems aligned is difficult whatever the valve size [13, 14].

Four trip/throttle valve design variations are used: two concerning the direction of the closing action and two, the method of holding the movable stem nut in its operating position.

● Disk is pushed onto seat design—In the larger valve sizes, the closing force on the valve stems and split coupling are frequently not adequately designed to withstand the impact load generated by this high closing force and any misalignment. Frequently, damage occurs to stems or the split coupling with the plug pushed onto a seat design. This can cause sticking or retarded movement of the steam valve plug, as shown in Figure 12.

• Disk is pulled onto seat design—The design that "pulls" the plug onto the seat without a lower guide is the preferable design, since the two stems and the split coupling operate in tension. This seems to limit the mechanical damage to the valve components during the closing action [13, 14]. Details of construction are shown in Figure 13. Also, fewer tight fitting stem areas where steam deposits can collect exist.

• Latch type stem nut holder design—In this design, the bracket is spring loaded to push it in one direction and has a knife edge latch mechanism to hold it and the stem nut in the proper position as shown in Figure 12. When the valve is called upon to act as a throttle valve, the stem nut is latched in place and operates in the manner of a conventional globe valve, permitting raising and lowering of the plug. When the valve is called upon to act as a trip valve, the stem nut and bracket are released from their operating position on the knife edges by a small hydraulic piston. Spring force pushes the stem nut downward to close the valve. "Hang up" of the hydraulic release piston in its cylinder caused by corrosion is frequently encountered. Since this is a dead ended point of the hydraulic system, accumulation of water and sludge is difficult to predict or prevent, causing a major problem [16].

Because of the extensive sliding and pivoting actions needed to release the stem nut for the tripping action, lubrication of the valve upper works is critical. The grease used tends to attract dirt and grit and/or retain moisture near tight fitting components. Exposure to high steam temperature also "cooks" out the lubricant, leaving "soap" or base material. All these problems can cause binding of the linkages.

The latch type trip/throttle valve cannot be exercised readily. By using a special gag, the valve can be moved to do some testing.

• Piston-type stem nut holder design—Another type of trip/throttle valve does not have the knife edge latch device. A larger oil cylinder that holds the stem nut directly is substituted. This is a globe type valve of inverted construction with the operating mechanism below the disk and a semibalanced disk arrangement. In this valve, the force for closing the valve is provided by a main spring above the oil piston and the steam pressure above the disk. The details are shown in Figure 13.

After the valve has been tripped shut, it is reset by turning the hand wheel clockwise. The rotation of the screw spindle will raise the main piston and compress the spring. The hand wheel will be turned until the piston comes to rest against the cover and stop in an upward direction.

This design admits oil through an oil inlet connection and orifice to the main oil cylinder with a relay valve. When the oil supply pressure is less than that required to reset the valve (generally about 45 to 50 percent of trip header pressure), the relay valve is unseated. The chamber below the main piston is opened to drain. When the oil pressure is increased to the reseated pressure, it overcomes the force of the relay spring, thus seating the relay valve and closing the passage from below the piston to the drain. Pressure then builds up on the main cylinder.

To open the valve, the hand wheel is turned counterclockwise. The oil pressure will hold the main piston against the cover, and the rotation of the screw spindle will lift the pilot valve off the seat. After the pilot valve moves its full stroke, it contacts the disk flange and further movement unseats the disk. The valve should be backed off about two turns from the wide-open position.

A mechanical or test bypass on the hydraulic piston should be installed, so that the valve can be exercised to approximately one-third closed position as frequently as needed. Free motion of this distance virtually guarantees positive tripping action in an emergency. The test bypass is so arranged that motion beyond one third travel cannot be achieved in the test mode, since the hydraulic piston covers the leakage port. Tripping action can take place when in the test mode. The details of how this can be accomplished are shown in Figure 13. Some small percentage of valve "hangups"

take place in the last one-third of the plug travel. Any exerciser device might not move the plug far enough to clear the area of obstruction. However, even this limited movement can give some assurance that the hold of steam deposits is broken on the valve parts. It is a very worthwhile feature on a trip/throttle valve. Limit switches for lights to show the valve is open, closed, or in the test position should also be installed.

Case History

During trip setting and testing activities, a turbine (2550 hp, 8610 rpm, 415 psig inlet steam to 185 psig exhaust, single control valve, forged rotor) went to overspeed. At first, no cause of the problem could be identified. The trip valve functioned perfectly during investigative tests. Finally, someone noted that when the right hand was used to relatch the knife edge as intended by the manufacturer, the valve worked. When relatched with the left hand, distortion of the connecting linkage took place and jammed the knife edge linkage. As luck would have it, a left-handed operator was conducting the test that resulted in the overspeed. Further analysis of the failure determined that the linkage connecting the hydraulic trip piston and the knife-edge latch holding the stem nut bushing was constructed very flimsily. The linkage could easily be distorted to the point that the trip piston could not unlatch the valve on every test. The trip/throttle valve was designed and manufactured by the turbine manufacturer. A survey of that same manufacturer's installations in the refinery revealed that the flimsy linkage could disable the tripping action on five more machines. Replacing all of the trip/throttle valves was necessary. In only a few cases have a poorly designed trip/throttle valve contributed so directly to a destructive overspeed. Obviously, a lack of training for operating and maintenance personnel also contributed. A training film was made as a result of the episode [17].

Nonreturn Valves

Extraction-induction/extraction connections remove steam from intermediate stages of a steam turbine for process uses, heating boiler feedwater or to drive auxiliary turbines. The steam flow to the lower pressure sections of the turbine is controlled by additional valve racks. Design considerations allow a minimum cooling steam flow to the low pressure end of the turbine, even when the readmission valves and the valve racks are completely closed. This cooling steam flow is more than enough volume to overspeed an unloaded turbine. A swing disk check or nonreturn valve is installed in the extraction or induction/extraction line. The valves are the equivalent of the trip/throttle valve on the extraction systems. If the nonreturn valve does not close after a turbine trip, steam will flow back into the tripped turbine and cause an overspeed condition.

The nonreturn valve, like the trip/throttle valve, has a dual function:

• It functions as a free swinging check valve to prevent reverse flow of process steam into the extraction stages of the turbine during normal operating conditions.

• It serves as a positive closing device that prevents reverse flow of steam into the extraction stages of the turbine when:

  • The emergency or overspeed governor action is started and closes the inlet steam stop valve.

  • The turbine is manually taken offline because of operational needs.

In older, smaller turbines, these valves can be simple check valves, actuated only by gravity and the back pressure in the extraction line. To raise the reliability of nonreturn valves, power-assisted valves, as shown in Figure 15, are used in larger turbines. A lever arm with a positionable counterweight and a hydraulic power cylinder provides the needed closing force. The hydraulic or

pneumatic power-assisted device is tied into the main overspeed trip system hydraulics.
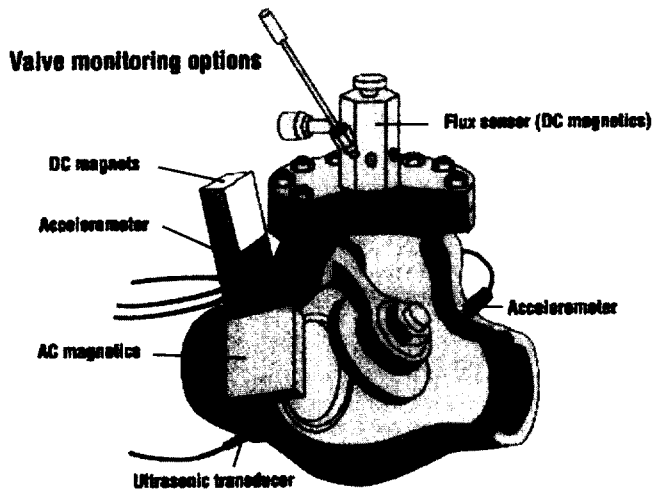

Figure 15. Nonreturn Valve Monitoring Options [18, 19].

Valves that have a weighted arm to aid in providing a closing force also have a negative aspect in that they must use a hydraulic relay valve to begin the power-assist feature. This relay can stick, as discussed earlier, and reduce the reliability of the nonreturn valve.

Frequent inspection and exercising of nonreturn valves is vital to assure that they are in perfect working order. It is even more difficult to test these valves than it is to test the trip device and the trip/throttle valve. The valves can be disabled by steam deposits on the valve stems, the most frequent cause of failure to close. The deposits may cause the valve stems to bend. Stem packing that is too tight may bind up the weight-arm.

Many turbine design engineers recommend the installation of two power-assisted check valves in series. Only a few higher speed machines currently have this feature. Questions exist about whether that series of positions would increase the reliability of the system *without* other concurrent major improvements.

Experience with nonreturn valves has not been good, because a high steam velocity through them is required to raise the check disk enough to reduce the pressure drop across the valve. Wear of hinge pins, disk studs, hydraulic or pneumatic pistons, and other mechanical parts is rapid because of flow impacting. In the past, the primary diagnostic technique has been to disassemble the valve or to await failure of the valve to function properly. The Electric Power Research Institute (EPRI), the Institute of Nuclear Power Operations (INTO), and the U. S. Nuclear Regulatory Commission (NRC) recently funded research into nonintrusive valve diagnosis. Results of the studies offer considerable improvement in the ability to monitor the condition of a critical valve [18, 19]. As shown in Figure 15, accelerometers are attached externally on the nonreturn valve to detect structure-borne vibrations caused by impacting and rattling of valve parts or flow noise. Digital filtering of the data identifies the source of the noise. The sensors can withstand temperatures up to 550°F. This approach may be the best way and perhaps the only way to detect the mechanical condition of a nonreturn valve while in service.

Case Histories

In one overspeed case, the lagging or blanket insulation on the turbine casing sagged enough to interfere with the movement of the weight arm arc on the extraction nonreturn valve. A casual inspection should have noted the problem. When the turbine tripped, steam continued to flow through the jammed open nonreturn valve. Enough energy was available to take the machine

to overspeed. During the subsequent failure investigation, it was determined that operating personnel were not aware of the existence, location, or the function of the nonreturn valve. Again, a lack of training of operating and maintenance personnel set the stage for the runaway.

In another situation, a fourteen stage (400 psig, 700°F inlet steam, 150 psig and 50 psig extractions, condensing) 7500 KW turbogenerator was destroyed during a heavy rainstorm. The trip system appeared to function correctly and in a timely manner after an electrical problem developed. However, a section from the middle of the rotor exited the casing. No stretching of the rotor wheel bores took place as occurs in an overspeed. This stretching is the "litmus test" of an overspeed. The two extraction lines literally ran all over the plant to serve process needs. Insulation and trap maintenance on steam lines is a very low priority item in most plants. This fact permitted the rapid accumulation of condensate during the sudden high rainfall. The turbine problem appeared to have been caused by large quantities of condensate that had collected in the lines during the chilling deluge of the rainstorm. The condensate pockets in the extraction lines flowed backwards during the tripping process. A vacuum present in the condenser flashed some condensate into steam but not all of it. After the trip off, the back flowing condensate "slugged" the rotor, breaking the shaft at the first wheel of each of the two extraction sections. The disk of the 150 psig nonreturn valve immediately in front of the first rotor shaft break was damaged and did not seat properly. It could not be determined if the disk had been damaged during the trip off or was damaged before the event occurred. Again, testing and inspection of the nonreturn valves contributed to rotor damage. Poor maintenance of the steam piping system also was a major contributor.

Vacuum Breakers

When a condensing turbine is tripped, a vacuum condition can exist in the turbine casing up to the inlet valves. Rapid pressure drops will cause any condensate present in the casing, and/or the connected piping in admission/extraction machines, to flash into steam that will reenter the turbine's steam path. With enough condensate present, the flash steam will accelerate the turbine and an overspeed condition can be reached. To prevent this condition from occurring, it is good practice and required by most industry standards to install a vacuum breaker activated by the primary trip system on the turbine exhaust.

OPERATING PROBLEMS

Serious doubts exist if most trip/throttle device designs can remain on the line for more than two years and still be free to operate in an emergency, for several reasons as discussed below.

Lack of Training

The original design ideas of older trip/throttle valves were based on the premise that the valve would be exercised and/or tested through its full travel, on a three to six months (with a one year maximum) interval. Refineries and petrochemical plants do not currently operate in this mode. A lack of training and understanding by both maintenance and operating personnel has made all parties afraid to do any kind of overspeed testing for fear of losing control of the turbine and causing a facility outage. The result is that testing is simply not done as frequently as it should be. The fears of testing are based more on the older and less reliable mechanical governor and trip systems than on the newer all electronic designs.

Water in the Lube Oil

In one wrecked turbine, water in the lube oil was a major contributor to the failure. Water in the lube oil, even at low levels, generates sludge in the oil and corrosion of the metallic parts of the governor control system and the overspeed trip system. Sticking

action of the overspeed sensing pin mechanism and the trip/throttle valve latching mechanism frequently occurs [16].

Steam Quality

Many refinery and chemical process units have waste heat recovery systems. Feedwater quality supplied to these boilers is often questionable. Boilers frequently have steam drums with poor moisture internal separation systems that permit carryover. Dirty steam caused by poor operating practices and/or poorly designed generation equipment, can also cause the sticking of the trip/throttle valve. Without proper operating and maintenance practices, these marginal boiler designs can cause catastrophic machinery failures.

The biggest problem with all of the trip/throttle valve designs is that the valve stem must be free to close rapidly, but it also must limit leakage from the steam path along the stem. This is done with soft packing sometimes or with a hardened steel leak-off bushing that permits controlled leakage between the stem and the bushing. Steam deposits present a problem either in the design that pulls the valve plug on the seat or the pushing on the seat design. The pushing force valve design quite often has an upper and lower guide bushing with tight fitting clearances as shown in Figure 12. Both designs are subject to movement retardation due to a collection of steam deposits on the stem that must enter a tight fitting hardened bushing. In addition, the guide sleeve for both designs can warp and offer restraint to plug movement.

Case History

The screen shown in Figure 16 is from the trip/throttle valve of a turbine (10,000 hp, 5100 rpm, topping, 415 psig to 115 psig) running in tandem with a Frame 5 gas turbine. Both machines drive a catalytic cracking unit two-case wet gas compressor train. The steam turbine used steam generated in two waste heat boilers by exhaust gases from this gas turbine and another one. The turbine was just downstream of the waste heat boiler. The boiler steam drums had very poorly designed dry pipes. Moisture separation internals in the steam drum consisted of a small area of a wire-mesh-screen over the dry pipe. Moisture and solids carryovers were very high. Steam quality analyzers were installed on the two waste heat boilers outlet pipes. The analyzers alerted operating personnel to the carryover problems, but no corrective actions were taken. Poor steam drum internals and a lack of training for the operating personnel to respond to correct the carryover lead to this problem. It was necessary to shutdown the entire process unit to clean the steam system. A runaway situation was avoided only because the flywheel effect of the gas turbine on the train prevented it. Operating independently, this turbine would have been a catastrophe looking for an opportunity to happen.

The effective area of the screens (or perforated plate) should be at least twice the cross sectional area of the turbine inlet connection. The strainer should be capable of withstanding a pressure differential at least equal to 25 percent of the *maximum* inlet pressure.

PROBABLE OVERSPEED
SEQUENCE OF EVENTS

If both the governor system and the overspeed system should fail to function correctly, the turbine can go to destructively high speeds. Investigation into the base cause is vital. Every case is different and must be investigated carefully. The sequence of events in a runaway as described by Sohre [20], is probably the best discussion and guide to failure investigation available. His experiences and comments, paraphrased and summarized for brevity are as follows.

● The turbine load is 100 percent lost when the generator trips electrically, or for a mechanical drive machine, when the coupling between it and the driven machine breaks. A compressor goes into a violent surge or the pump loses suction causing considerable (50
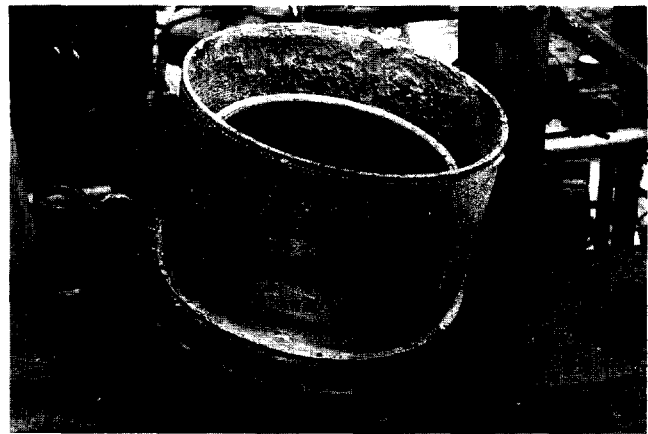

Figure 16. Trip/Throttle Valve Screen with Heavy Carryover Deposits.

percent to 80 percent) load to be lost. With a greatly reduced load imposed on it, the turbine starts to accelerate rapidly. As little as a 10 percent loss, coupled with a poor control system, have triggered a runaway.

● The overspeed pin weight or the latching device can be frozen, due to sludge from the lube oil and/or corrosion, and fails to initiate the valve closing action.

● If the overspeed trip pin does actuate, the trip/throttle valve *and* the governor valve, *or* the extraction nonreturn valve, may not close completely because of steam deposits on their stems or other moving parts. The "coupling" block connecting the rotating and sliding stems of the trip/throttle valve is subject to damage in normal use and can cause problems with alignment of the two stems.

● As the turbine speeds up past about 120 percent of operating speed, critical speeds are encountered and severe vibration occurs. What happens next is dependent upon the turbine manufacturer's design and what the weak link in the design is. At 125 percent of operating, all rotor stresses are 56 percent above normal. In some designs, white metal bearing failures and exhaust end and interstage seal rubs occur.

● In some designs, cast iron housings with weak bolting fail at a much lower speed than cast steel housings. The bearing housing cap is especially susceptible to fracture. The housing design represents part of the bearing stiffness discussed in all rotordynamics papers.

● As the turbine approaches twice normal speed, the blading/wheel stress level is about four times normal.

● The shaft rotational speed of an impulse design will level out just below twice design speeds because of the limiting exiting velocity of the steam from the nozzles and the increased friction of the rubs, failed bearings, or broken bearing housings. Centrifugal force at this speed causes severe yielding of blading dovetails. Disk dovetail grooves at the periphery of the disk on a forged rotor design and the disk bores of a stacked rotor design begin to yield. The disk bores increase by 1/2 in or more, disengaging the keys that transfer torque from the disks to the shaft.

● The turbine disks yield and/or disintegrate (built up rotor design). This massive yielding is the "litmus test" of overspeed. Without the yielding, the cause of failure is not overspeed. The disks now spin on the shaft free of any drag from the damaged bearings or bearing brackets.

● The turbine shaft breaks, most generally inboard of the exhaust end bearing.

• Instead of the single shaft break discussed above, multiple shaft fractures may occur if a critical speed and the accompanying vibration levels are encountered. Sometimes, the turbine shaft has broken into as many as five pieces.

• Penetration of the turbine casing by blading and disk fragments occurs, especially in the exhaust section where the blading is longer and heavier. The casing in this area is frequently made of cast iron that ruptures in fragments.

• Major components of the rotor may exit the casing. Tendencies are for disks to separate from the shaft and become airborne independently.

• The severe vibration causes lube oil line fractures, and fires are started as the leaking oil sprays on hot steam piping.

• If stuck open trip/throttle or stop valves were a factor in the initiation of the overspeed, they often close during all the vibration. This fact can create confusion in the later investigation into the accident as the closed valve is inconsistent with the runaway.

PERIODIC TESTING OF
THE TURBINE TRIP SYSTEM

The preventive maintenance program in a plant must be good enough to prevent a destructed and uncontrolled overspeed occurrence. In the past, the emphasis has been focused on periodic testing of the protective devices. A paradox exists in the need for periodic testing. To ensure the protection of the turbine and operating personnel from risks during normal operation, an uncoupled run for setting the trip device places the turbine and personnel at risk. Often, inspection and overhaul of emergency shutoff valves and overspeed trip systems are not done at the time of a dismantled inspection of a steam turbine. Contractors, even original equipment manufacturers, do this service only when it is specified in the contract bid given to them by the steam turbine owner. It is not a standard item.

A complete dismantling and refurbishing of the trip/throttle valve and any nonreturn valves should be done at least every three years or more often, if experience suggests. This is a more frequent interval than the turbine is opened and inspected. A detailed examination of a given plant's maintenance practices may save much grief later. To ensure that enough power is available to drive the turbine to overspeed, the turbine is unloaded by reducing the electrical load to zero for a generator or by uncoupling the load for a mechanical drive machine. In either case, it takes very little steam to take the machine to destructive speeds. A well-written instruction sheet that documents the periodic testing procedures should be used during the periodic testing.

Many overspeeds take place during testing because personnel are not properly trained. Ignorance is the base cause. Good training aids are available [19]. The tendency during investigations is to place the blame on the trip system rather than the personnel involved. Placing the blame on an individual who may have been severely injured or killed during an overspeed situation is difficult emotionally for the investigating group. While it is true that equipment can be defective, most runaways are a result of personnel error. When slow rolling, test running, or overspeed testing the turbine while uncoupled from the load, control should be by the *trip/throttle valve only* or a bypass around the inlet valve. The slightest movement of the governor valve in an uncoupled situation can result in tremendous changes of speed. A sticking governor valve or linkage, which was not corrected before testing, can be very dangerous. To prevent maximum steam flow to the trip/throttle valve, a one inch bypass line, with a globe valve around the steam turbine inlet block valve (not the trip/throttle valve) should be installed. This bypass valve can be used for a turbine warm up or overspeed testing. To prove out the governors' system, limit the maximum speed with the hand wheel of the trip/throttle valve or the bypass line and control the speed *downward* with the governors' system.

Two accurate and independent means of determining the correct turbine speed should be used. By registering confusing readings, resonating reed and strobe light tachometers have contributed to several overspeeds. They should not be used for testing.

CONCLUSIONS

Operating flexibility and increased reliability of centrifugal compressor trains and critical pumps requires substantial upgrading of the controls for the steam and gas turbine drivers. Most of the failure-to-trip and resulting overspeed conditions can be attributed to seven basic problems.

• Steam deposits on the trip/throttle valve stem (or stems)

• Lubrication deposits (i.e., soaps, dirt, detergents, etc.) in the top works of the trip/throttle valve exposed to the elements

• Mechanical failures of the trip/throttle valve resulting from bent stems, damaged split couplings, etc. This comment is particularly true for the push on the seat designs.

• Galling of the piston in the hydraulically latched cylinder

• Jamming of the screw spindle (Figure 13) in the fully oil operated cylinder-type valve design due to the use of excessive force by operating personnel

• Lack of training for both operating and maintenance personnel

• Lack of a rigid testing program to detect problems early

• Failure to upgrade turbine components, such as the mechanical/hydraulic governor, to match owners' expectations of extended run times and longer machinery overhaul intervals that exceed the manufacturer's initial design ideas

RECOMMENDATIONS

Experiences with various turbine installations suggest that the following are good engineering design practices for the current turbines and trip/throttle valves.

• Mechanical/hydraulic control (MHC) governors should be upgraded to electronic hydraulic control (EHC) governors.

• The trip/throttle valve should be upgraded with the following guidelines:

  • The "pull on" plug design is less likely to be damaged in service than the "push on" type trip/throttle valves.

  • A built in "exerciser" on the fully oil operated trip/throttle valve is preferable. Even limited movement of the valve stem is better than doing nothing over run extended periods.

•. The present overspeed sensing devices should be upgraded for greater reliability. The dished disk and some eccentric ring designs are adequate. Most of the pin and offset pivot designs should be upgraded to electronic devices when the EHC governor upgrade is done.

• Training and testing—Intensive training should be done for operating and maintenance personnel on how the current trip systems function, including testing programs. It is very important that these protective devices function when needed. The hot, wet atmosphere around a turbine causes corrosion and minerals in the steam plate out on moving parts. In addition, normal wear and tear will cause parts to stick or malfunction. For these reasons, all of the various steam turbine protective devices must be cleaned, calibrated, and checked at regular intervals.

• Hydraulic relays in the trip system should not be permitted.

• Coupling selection for mechanical drive trains should be done very carefully to ensure an adequate rating.

• Long term, the whole function of a trip/throttle valve should be reviewed. The push for dual functions of the trip/throttle valve has reduced the present valve design's value as a positive shutoff

against overspeed. For larger turbines (500 hp and up), hydraulically operated, positive closing, single function stop valves should be used. This would be essentially an unbalanced main valve and a pilot valve held open by hydraulic forces generated in a large cylinder. When oil pressure is released through action of the relay valves, closing or tripping action results. Closing forces are a combination of both the steam and spring component. A back seating device should be provided for the valve in the open position so that greater clearances can be used between the hardened stems and bushings without concern for steam leakage and deposits. This mechanism would not have the problem of coupling a rotating and stationary stem together. The stem could be one long length and alignment would not be a problem.

While there is need for improving the governor and overspeed trip's systems, the major problem area is the training and understanding of the personnel that operate steam turbines. In these days of instant solutions, steam turbines still require constant attention while operating and a high level of maintenance expertise. If good practices are not followed, a failure is almost sure to result. Poor steam quality, water or dirt in the lube oil, poor maintenance of the overspeed trip system proper, and the governors' system combine to create failure conditions. These comments are summarized in Table 5.

Table 5. Factors Related to Steam Turbine Overspeed Problems.

Factor	Short Term Fix	Long Term Solution
Steam turbine loses load rapidly	(1) Use vibration analysis to detect coupling problems to prevent coupling failure (2) Proper coupling, surge protection instrument maintenance (3) Operate process to avoid unit upsets and compressor surge (4) Intensify training for all phases above	(1) Evaluate coupling applications carefully (2) Improve condition monitoring systems where needed (3) Upgrade surge protection system to best available technology (4) Upgrade process control system where needed (5) Upgrade electrical distribution system as needed
Failure of overspeed sensor	(1) Frequent testing and training (2) Improved maintenance and training	Upgrade governor to electronic and overspeed sensor to dished-disk or electronic
Trip valve fails to close completely	(1) Improve steam quality/training (2) Frequent testing/exercising valve (3) Improve maintenance/training	(1) Steam quality—install continuous monitors and alarms on steam, feedwater, condensate systems, etc.; upgrade boiler internal steam separation equipment; upgrade steam process control systems. (2) Replace trip valve with improved design to permit some exercising (3) Install parallel trip valves to permit testing while turbine is in normal operation
Extraction nonreturn valves fail to close	(1) Frequent exercise/training (2) Improve maintenance/training	(1) Replace nonreturn valve with improved design (2) Install monitoring equipment as per EPRI practices
Overspeed occurs during uncoupled/low load testing	(1) Increased training (2) Detailed written procedures (3) Use small manual bypass valve for speed control during test	(1) Upgrade to well designed electronic governor and trip system that can be tested without unloading turbine.
Governor system fails	(1) Improve steam quality/operator training (2) Improve maintenance/training	(1) Upgrade to well designed electronic governor system and decrease acting time of governor valves.

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