USING READILY AVAILABLE PLANT DATA TO ASSESS THE IMPACT OF ATTEMPERATOR PERFORMANCE ON HRSG PRESSURE PART LIFE

Robert Anderson
Competitive Power Resources Corp
P O Box 383
Palmetto, Florida 34220 USA

Abstract

Premature failure of heat recovery steam generator tubes and piping caused by poor performance of interstage attemperators is a common occurrence in combined cycle and cogeneration plants. This paper demonstrates how to utilize readily available historical distributed control system (DCS) data to identify damaging thermal transients associated with attemperators. It gives a brief review of heat recovery steam generator (HRSG) heating surface arrangements, attemperator design features, control logic configurations, and operator practices identified as contributors to poor attemperator performance and subsequent pressure part damage. Suggestions for corrective actions are also presented.

Introduction

Malfunction, inappropriate operation, and poor performance of interstage attemperators are responsible for many piping failures and superheater (SH) / reheater (RH) tube failures. The permanent instrumentation supplied with the typical combined cycle and cogeneration plant provides all the data necessary to assess attemperator performance. Unless noted otherwise, comments regarding SH attemperators apply equally to RH attemperators.

Some of the historical DCS data plots developed during assessment of thermal transients in over 30 combined cycle and cogeneration plants are presented in this paper to demonstrate common and damaging attemperator induced thermal transients. Many of these thermal transient assessments were conducted in concert with water chemistry and flow accelerated corrosion assessments (not included in this paper) conducted by Barry Dooley of Structural Integrity Associates. The findings resulting from these assessments are similar across a wide variety of HRSG manufacturers, operators, process details, and countries. The plant variables included in the assessments include:

- 13 HRSG manufacturers
- 5 Gas turbine (GT) manufacturers (11 models – industrial and aero derivative)

- 8 Steam turbine manufacturers
- 2-Shaft and Single-Shaft power generation arrangements
- Cogeneration only arrangements
- River, seawater, cooling tower, air-cooled condenser cooling
- 4000 to 130,000 operating hours
- 90 to 630 starts
- Single pressure, 2-pressure and 3-pressure
- 1-on-1 and 2-on-2 configurations
- Reheat and non-reheat
- With duct burners, without duct burners
- Horizontal gas path, vertical gas path

Attemperator Basics

Figure 1 shows a typical interstage attemperator installation with the spray nozzle and thermal liner located in the steam pipe between the primary and secondary SH. A properly designed, installed, operated, controlled, and maintained attemperator injects a carefully controlled and uniformly distributed spray of very small water droplets into steam exiting the secondary SH only when this spray water is required to limit secondary SH outlet steam temperature within its design limit.

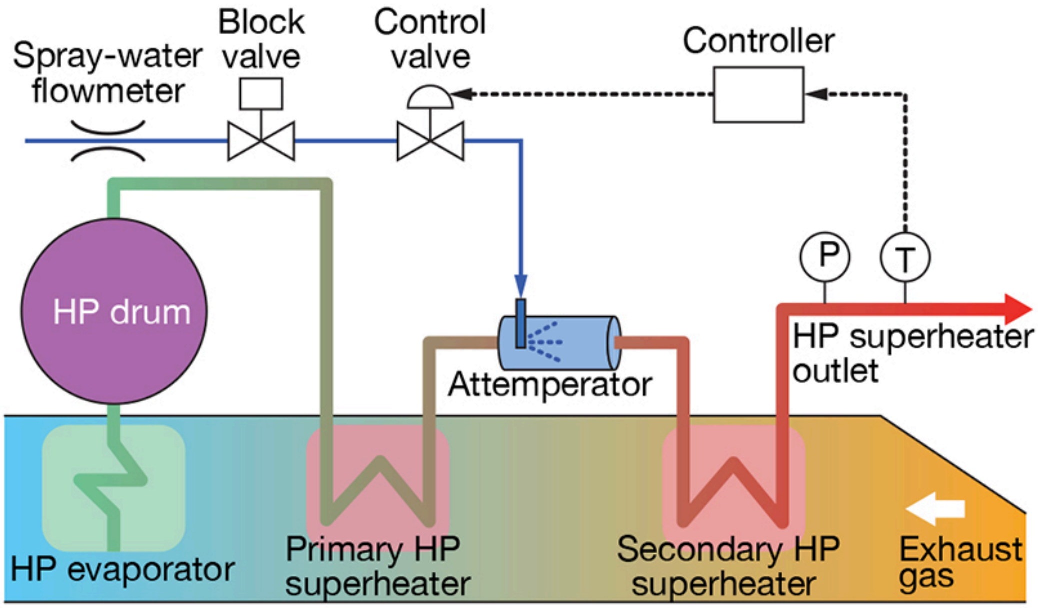

Figure 1 [1]

Basic interstage attemperator arrangement and instrumentation with simple feedback loop control.

The quantity of spray water injected should never be more than can be completely evaporated before the largest (and slowest to evaporate) water droplets arrive at the first pipe tee branch or elbow downstream of the attemperator. Properly designed, maintained and controlled attemperators always maintain a minimum of 50°F (28°C) superheat at the attemperator outlet under all operating conditions. Moderate “overspray” conditions result when attemperator outlet steam temperature decreases below 50°F (28°C) superheat. Severe overspray conditions result when attemperator outlet steam temperature is permitted to decrease to saturation temperature. The most effective attemperator control logic schemes prevent attemperator operation during unit startup and shutdown when exhaust gas temperature (EGT) or steam flow are below predetermined minimum values necessary for proper attemperator performance.

Attemperator Arrangements

Interstage attemperators are found in various locations. Many older units with attemperators installed in steam pipes at the top of the HRSG have no drains to prevent leaking spray water from entering the SH tubes. In 2007 the ASME Boiler and Pressure Vessel Code added the requirement for an automatically controlled low-point drain between attemperators and SH tubes. Most pre-2007 HRSGs with attemperators installed in steam piping at the bottom of the HRSG have drains in these pipes. However, these drains are sometimes too small, or routed in such a way, so that drain flow rates are too slow to remove all water before steam flow carries it into the tubes. Figures 2 though 6 show some typical attemperator arrangements.

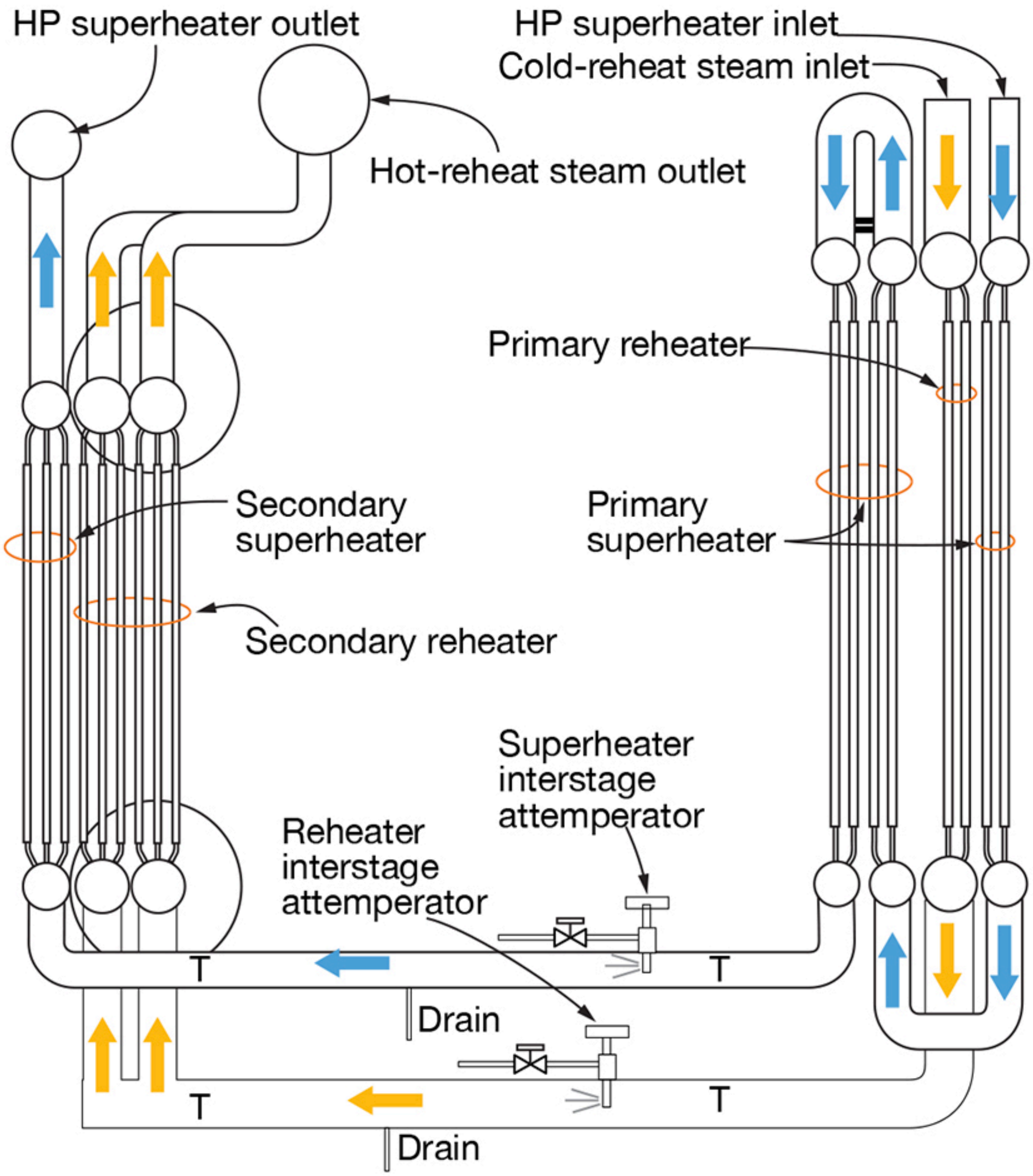

Figure 2 [1]

The SH and RH attemperators are both located in nominally horizontal steam pipes at the bottom of this horizontal gas path HRSG.

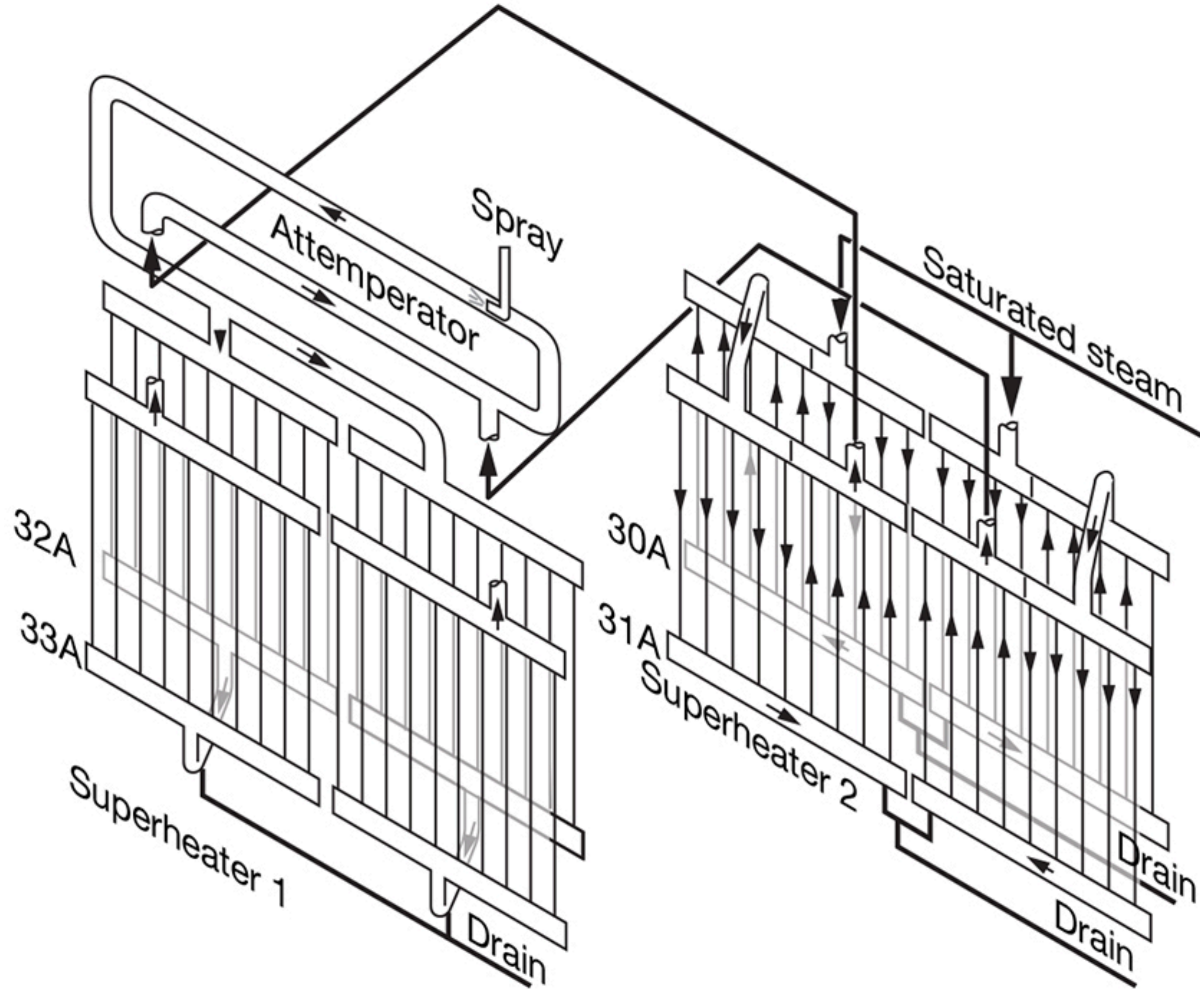

Figure 3 [1]

This pre-2007 SH attemperator is installed at the top of the HRSG in steam piping with no drain to prevent spray water from flowing into the tubes.

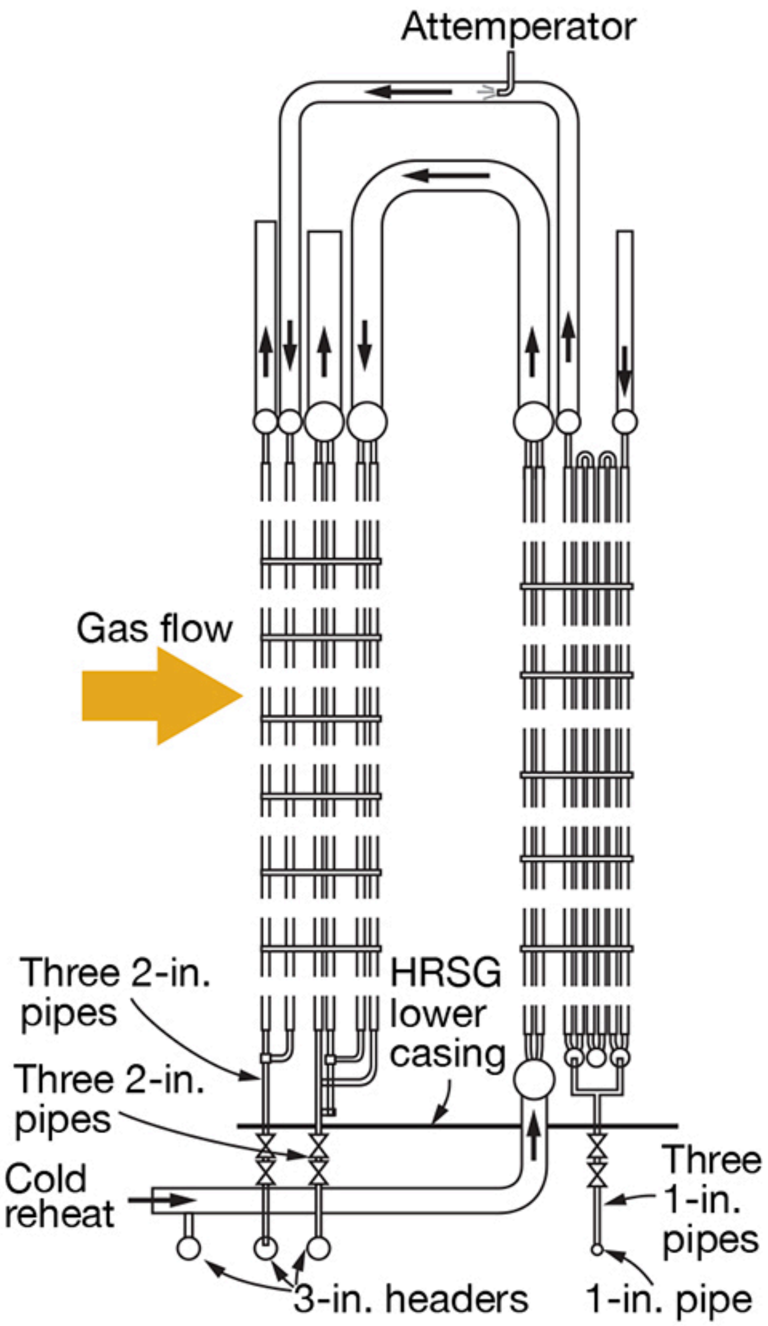

Figure 4 [1]

The SH and RH attemperators are both located in nominally horizontal steam pipes at the top of this horizontal gas path HRSG with no drain to prevent spray water from flowing into the tubes.

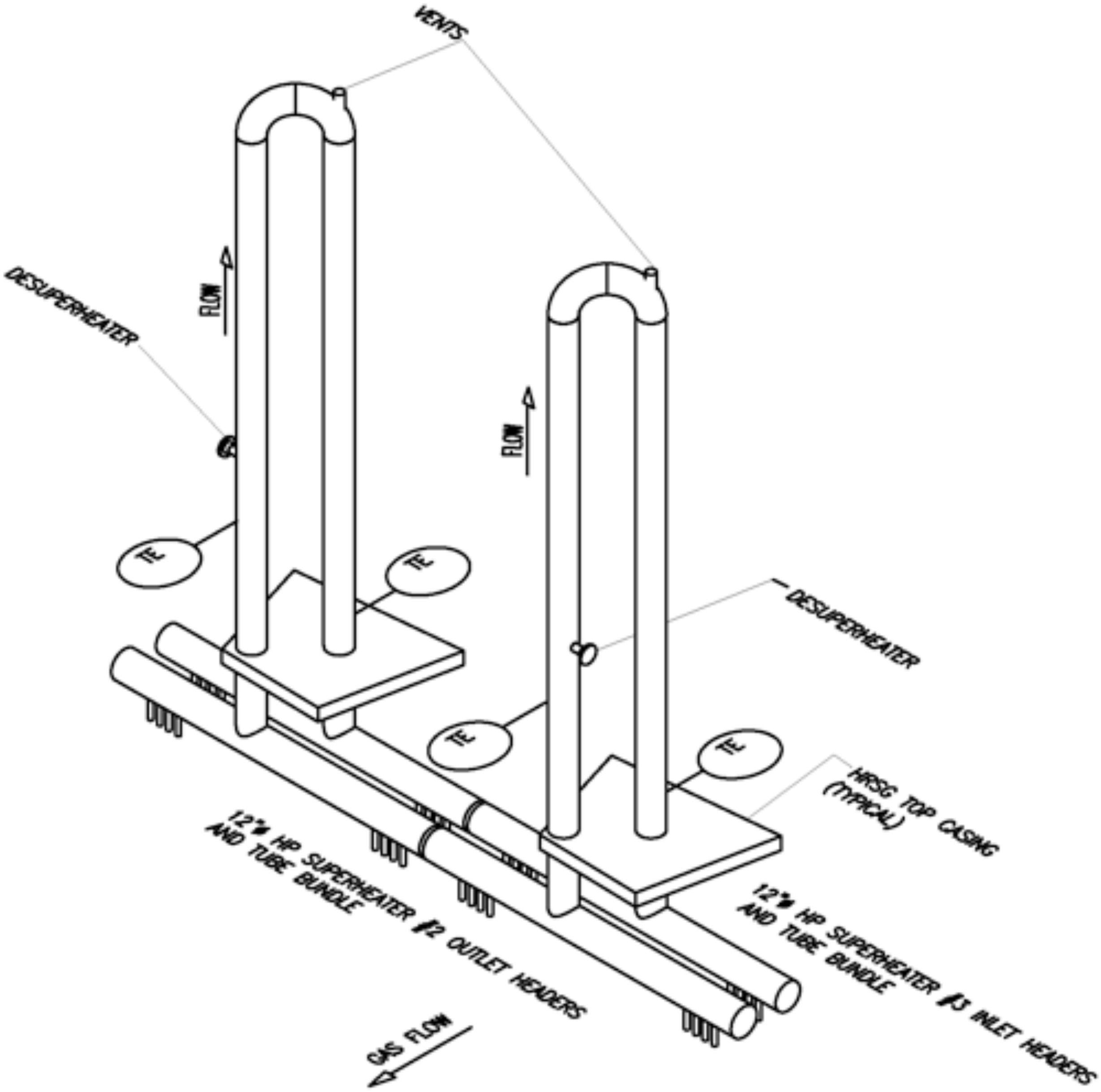

Figure 5 [2]

This top located attemperator arrangement is found in many pre-2007 HRSGs due to its compact design. It is no longer installed due to the inability to install a drain between the attemperator and primary SH.

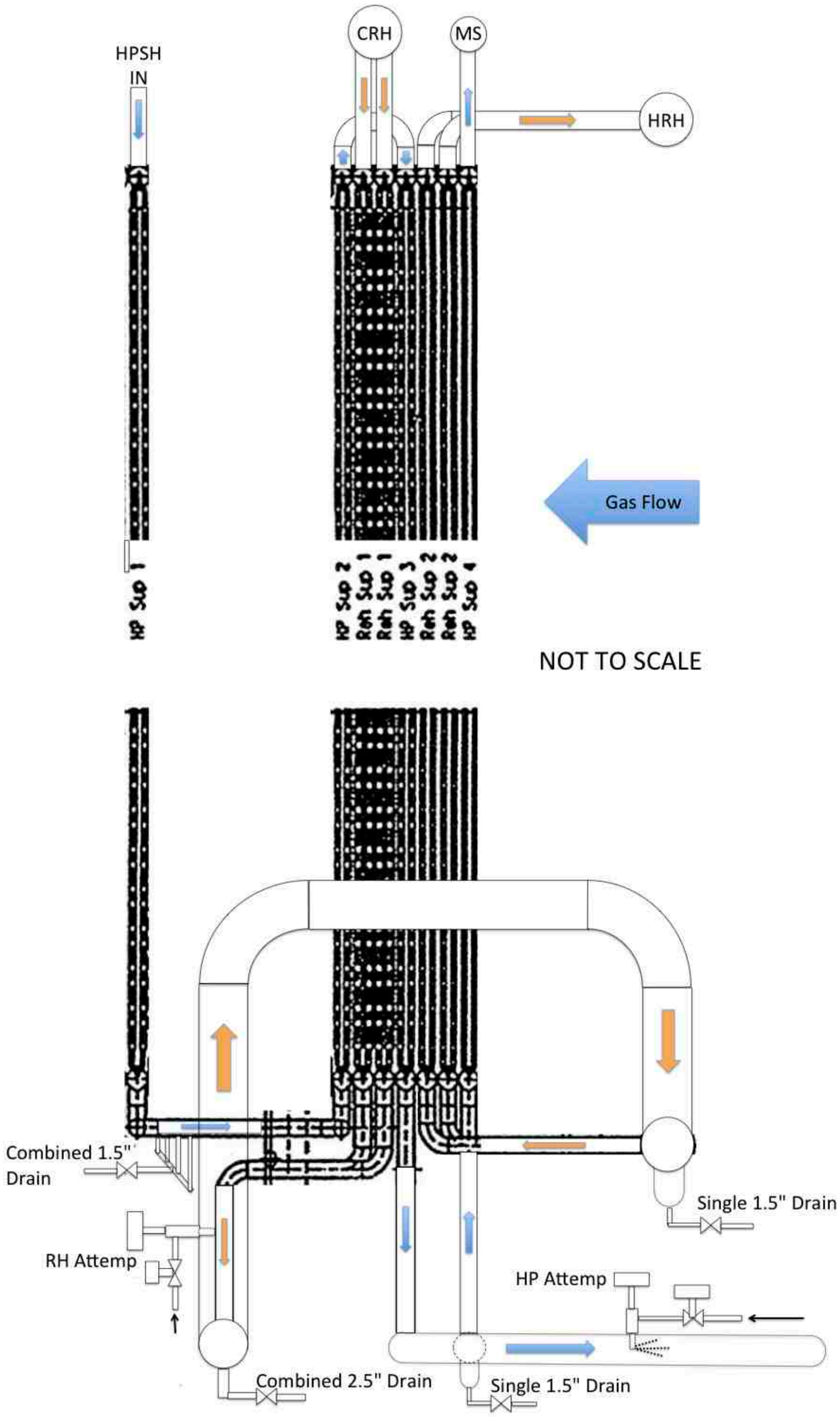

Figure 6 [3]

The SH attemperator is located in a nominally horizontal steam pipe loop and the RH attemperator is located in a vertical pipe steam pipe loop at the bottom of this HRSG.

Attemperator Induced Pressure Part Damage

Malfunction and/or improper operation of attemperators cause rapid quenching of downstream components. These large thermal transients result in high fatigue life expenditure and premature failure of tube-to-header connections, steam pipe elbows, steam pipe tee branch connections, and steam pipe girth welds. These transients also result in both temporary and permanent distortion of steam pipes with resulting damage to pipe supports and redistribution of piping loads. Severe quenching of tubes in-line with secondary SH inlet nozzles often causes these tubes to shrink so much that the material exceeds its yield stress and permanently stretch in length. Now being longer than neighboring tubes when returned to the same temperature, these tubes must bow out of line and are obvious during visual inspections. Figures 7 though 12 show examples of typical damage induced by attemperators.

Figure 7 [1]

Crack at extrados of steam pipe elbow downstream of RH attemperator

Figure 8 [1]

Thermal fatigue crack in SH tube at the toe of the tube-to-header weld. This crack was caused by attemperator spray water repeatedly flowing back into the primary SH after shutdown. This HRSG used an attemperator configuration like that shown in Figure 5.

Figure 9 [1]

Ductile overload failure of SH tubes. These failures were caused by a large quantity of spray water entering a few hot SH tubes to produce a severe quench. Ductile overload failures typically occur immediately after the water enters the tubes and do not require repeated exposure to large thermal transients like the fatigue cracks shown in Figure 8.

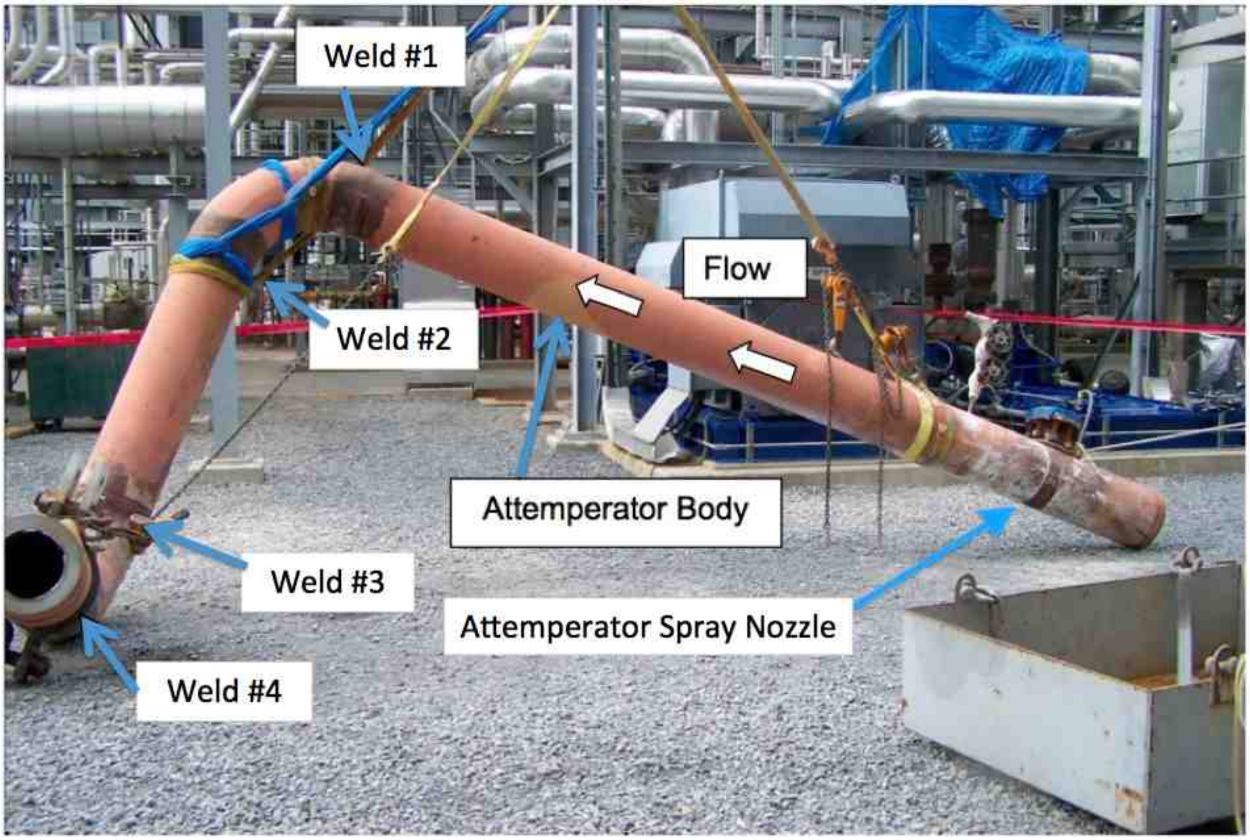

Figure 10 [3]

All 4 of these welds down stream of the SH attemperator were cracked due to repeated overspray during startup and low-load operation.

Figure 11 [3]

Permanently stretched tubes located in-line with secondary SH inlet nozzle. Permanent elongation of tubes results when a few tubes are severely quenched while their neighbors remain hot.

Figure 12 [3]

This primary RH inlet manifold was permanently deformed by attemperator spray water flowing into the hot manifold after shutdown. This manifold was installed in the arrangement shown in Figure 6.

Monitoring Attemperator Performance

Overspray conditions, spray water leakage, manual reduction of the attemperator control setpoint, and/or operation of the attemperator when it is inappropriate to do so are responsible for most attemperator induced pressure part damage. These conditions can be easily observed by evaluation of readily available DCS data. The DCS data points useful in diagnosing SH attemperator performance are:

- GT Load
- GT EGT
- Secondary SH Outlet Temperature

- Attemperator Outlet Temperature
- Attemperator Inlet Temperature
- SH Saturation Temperature
- Drum Pressure
- Spray Water Flow
- Spray Control Valve Position
- Spray Block Valve Position

The DCS data points useful in diagnosing RH attemperator performance are:

- GT Load
- GT EGT
- Secondary RH Outlet Temperature
- Attemperator Outlet Temperature
- Attemperator Inlet Temperature
- RH Saturation Temperature
- RH Pressure
- Spray Water Flow
- Spray Control Valve Position
- Spray Block Valve Position

Plotting these points during layup, startup, shutdown, load changes, and low-load operation permits identification of conditions likely to cause attemperator induced pressure part damage. Figures 13 through 19 show DCS plots from several units with examples of good and bad attemperator performance.

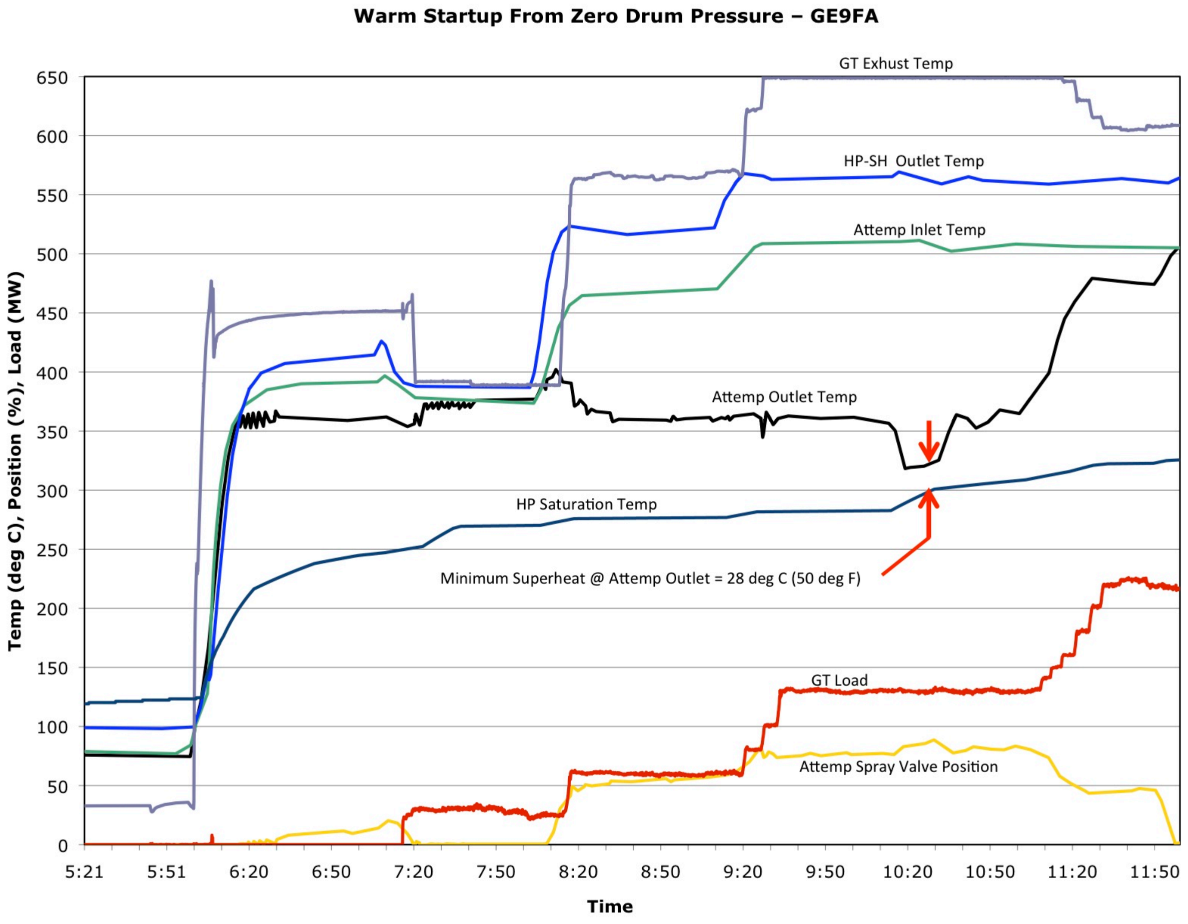

Figure 13 [3]

The SH attemperator performed as it should during startup of this GE 9FA GT

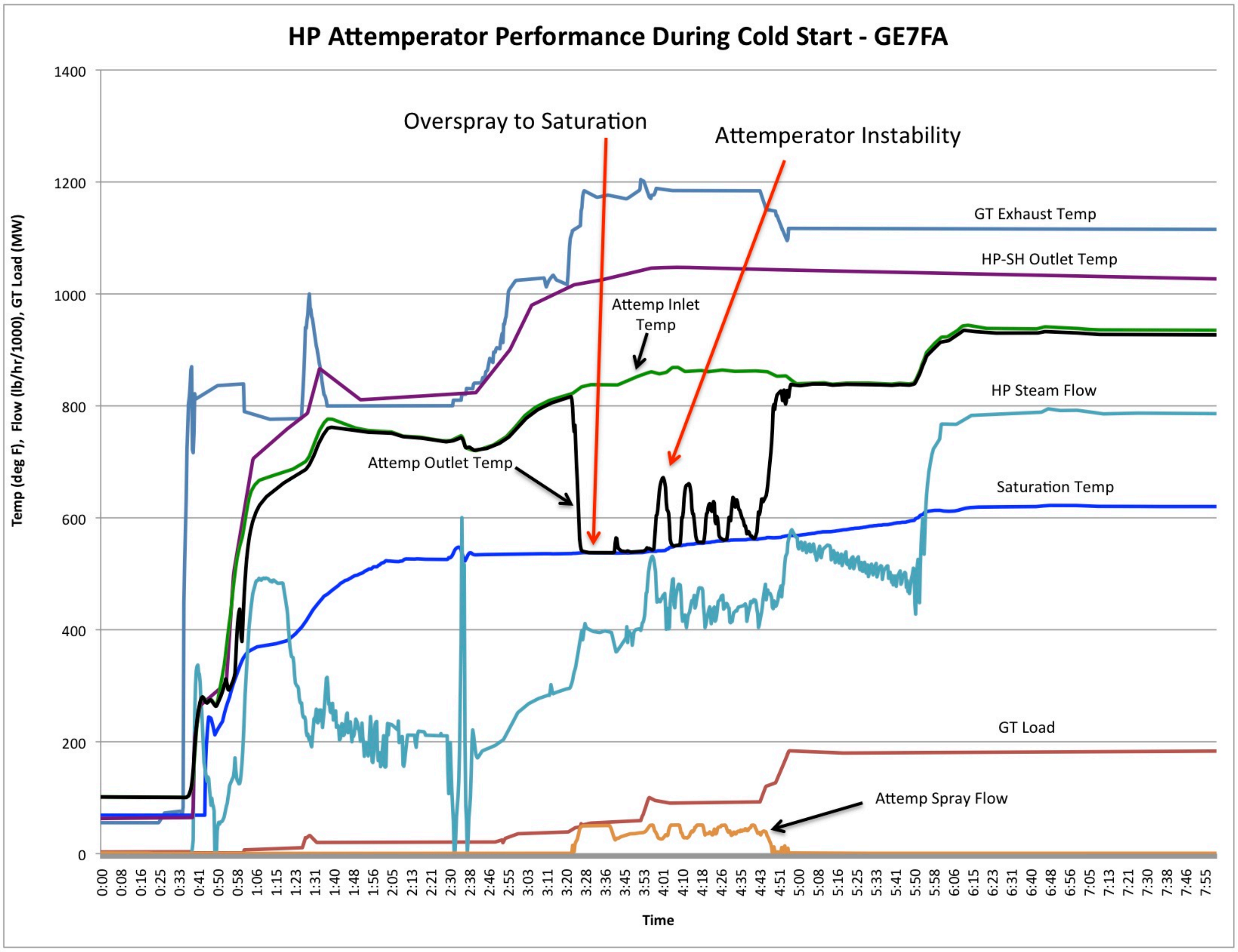

Figure 14 [3]

Overspray to saturation temperature and poor SH attemperator control stability is apparent during startup of this GE 7FA GT

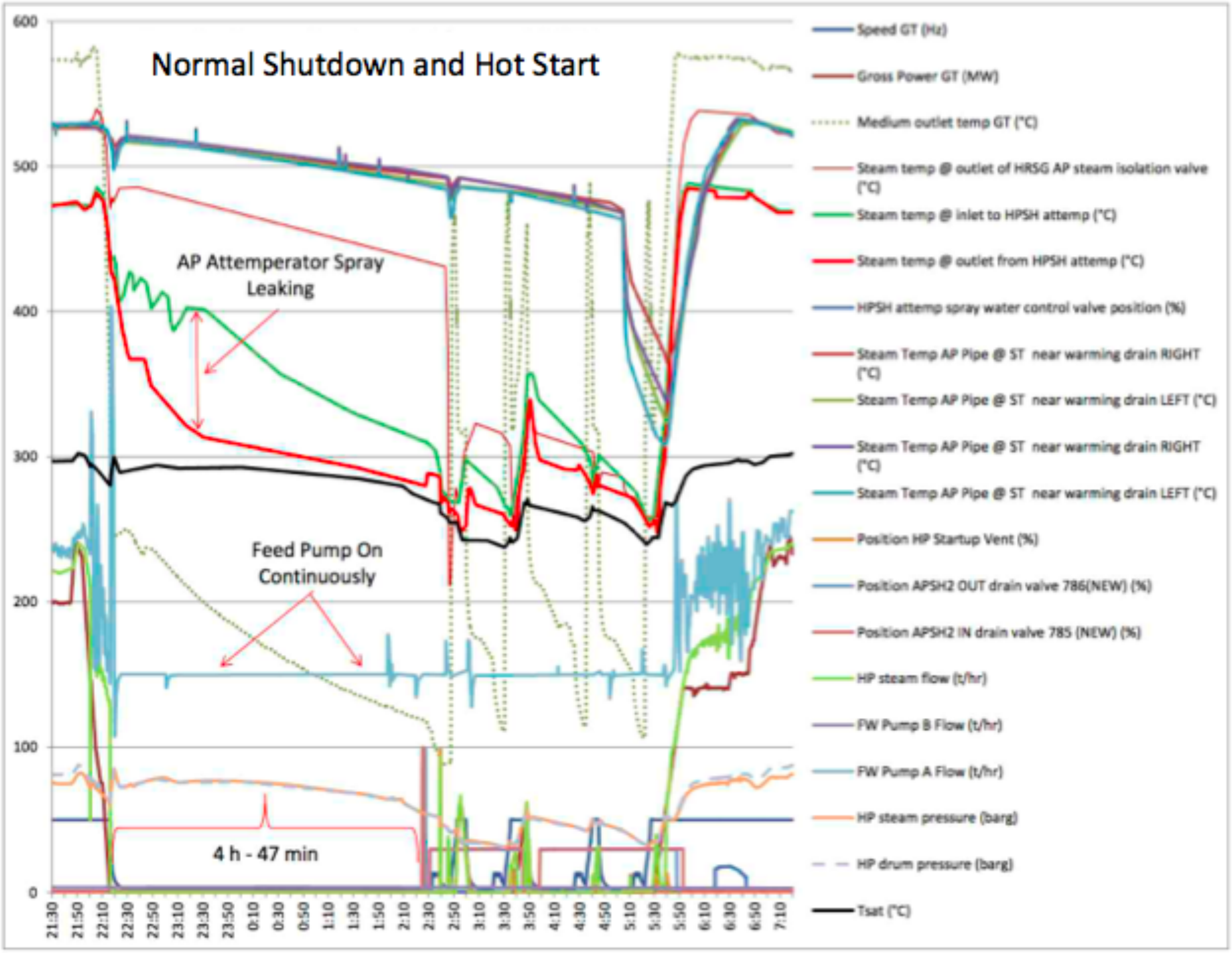

Figure 15

The large temperature difference between attemperator inlet and outlet temperature during layup indicates significant leakage of spray water into the steam pipe between the primary and secondary SH in which the attemperator is installed. The boiler feed pump was operated continuously during this layup period – providing a source of pressurized spray water to the leaking attemperator valves.

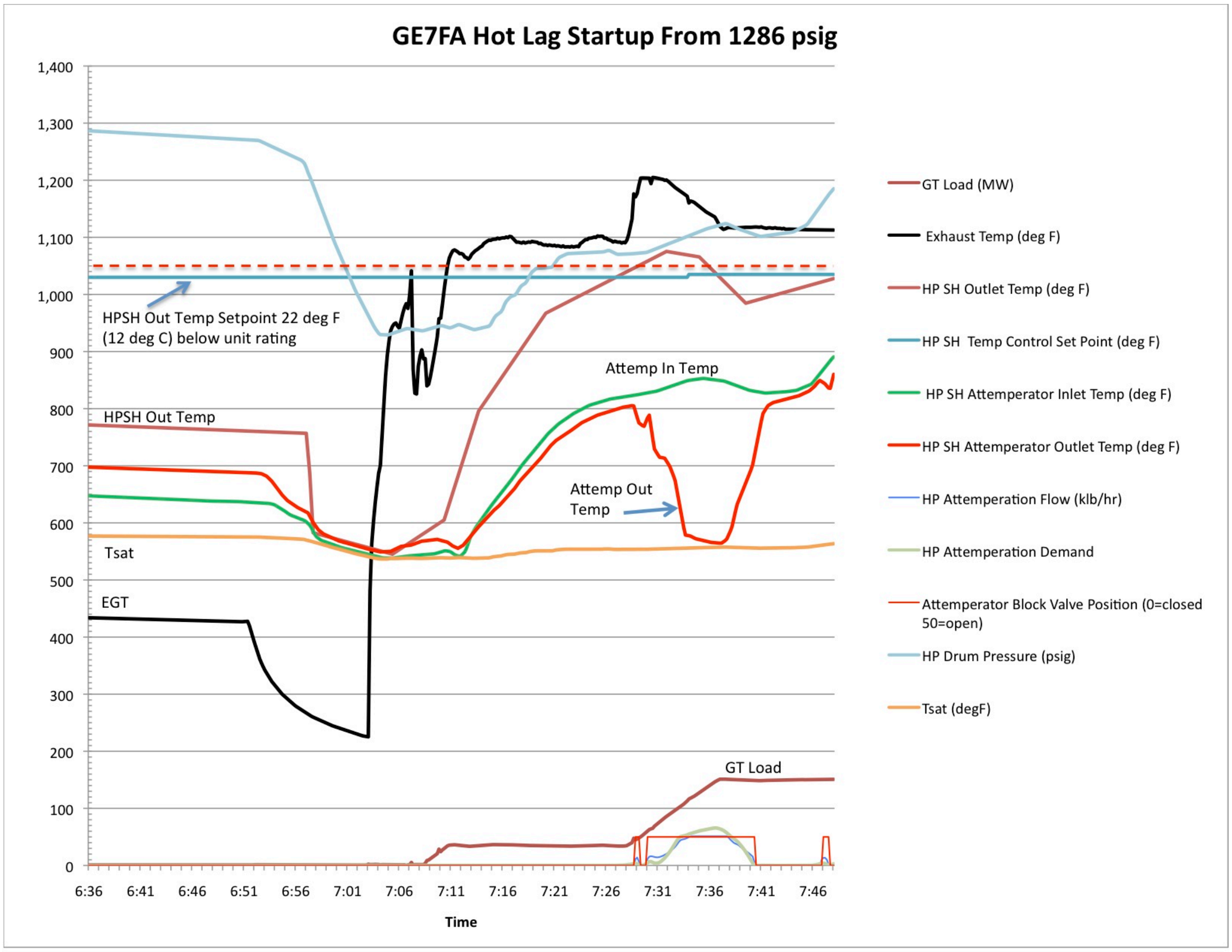

Figure 16 [3]

Overspray to saturation temperature and excursion of SH outlet steam temperature above the unit's rating both occur during lag-unit hot startup of this GE 7FA GT

The SH attemperator setpoint is lower than the SH steam outlet temperature rating throughout the hot startup in Figure 16. This contributes to the overspray condition that occurs while GT load is increased from minimum load to about 100 MW. During this load range EGT quickly increase to above 1200°F (649°C). Lowering the setpoint is a typical, but misguided, technique used by many operators in an attempt avoid SH outlet steam temperature excursion.

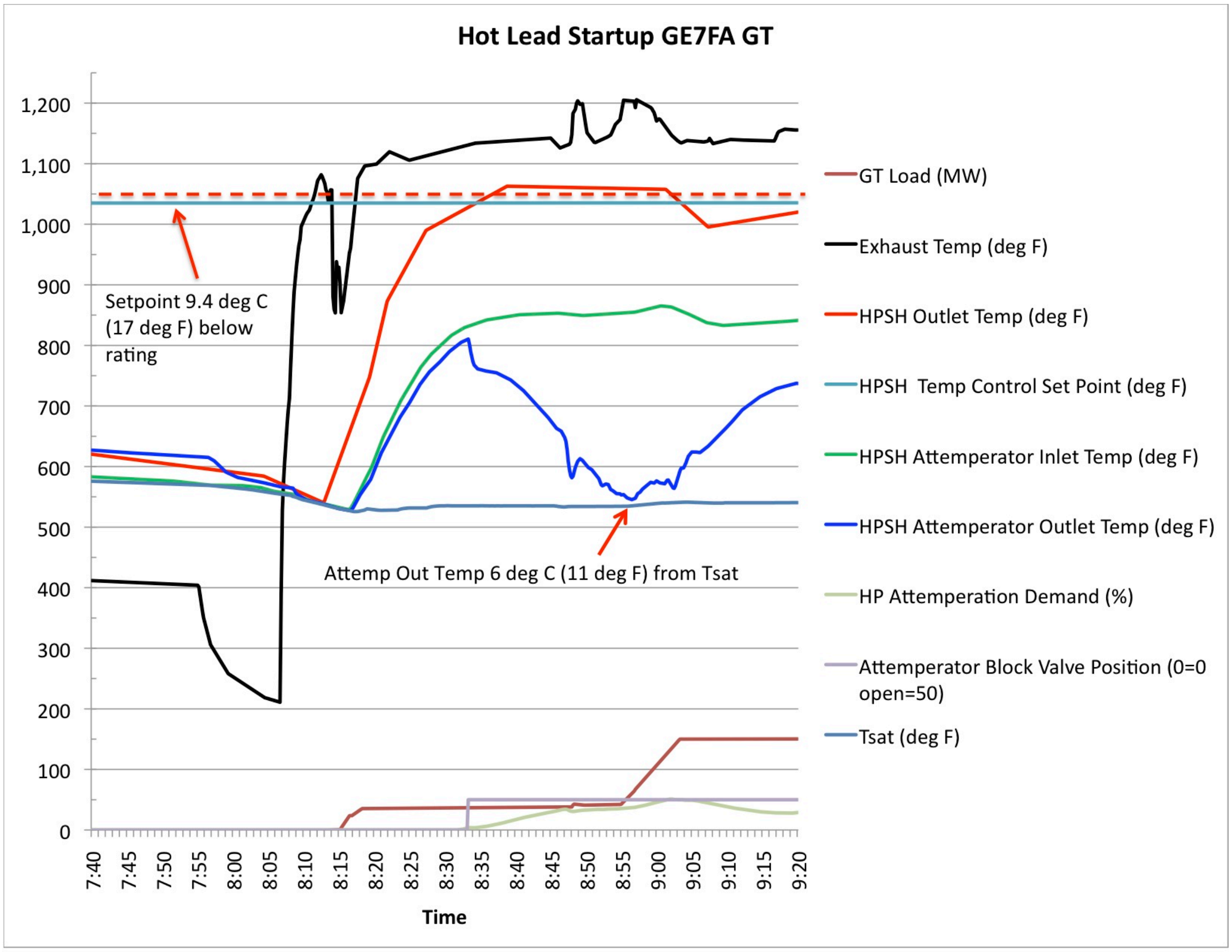

Figure 17 [3]

Overspray to within 17°F (9.4°C) of saturation temperature and excursion of SH outlet steam temperature above the unit’s rating both occur during lead-unit hot startup of this GE 7FA GT

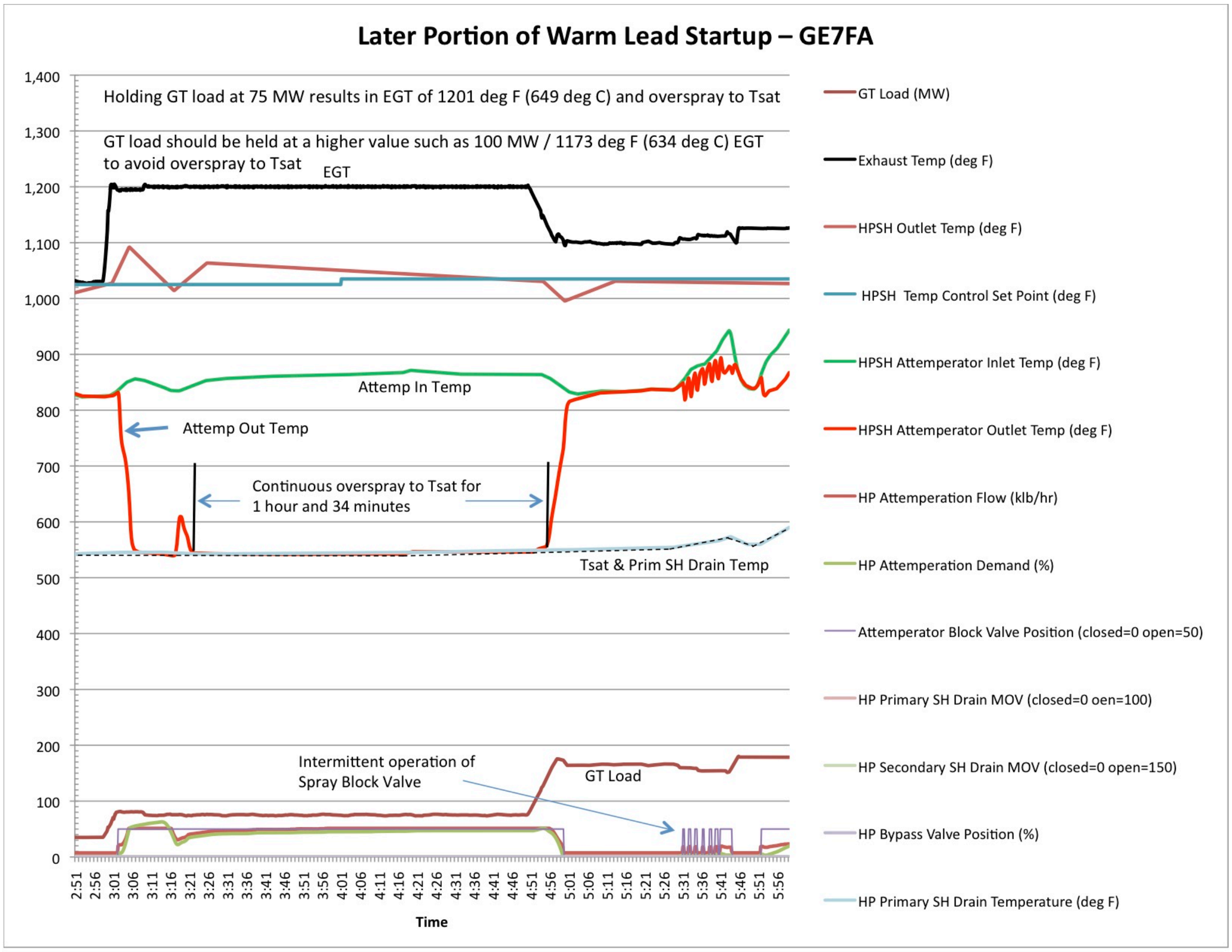

Figure 18 [3]

Continuous overspray to saturation temperature for 1 hour and 34 minutes results from holding this GE 7FA GT's load at 74 MW

The GE 7FA GT has a very aggressive exhaust temperature characteristic. EGT quickly increases to, and remains above, 1200°F (649°C) whenever GT load is between about 16 MW and 100 MW. The GE 9FA GT has a similar exhaust temperature characteristic. Attemperators installed in HRSGs behind these engines have an exceptionally difficult job.

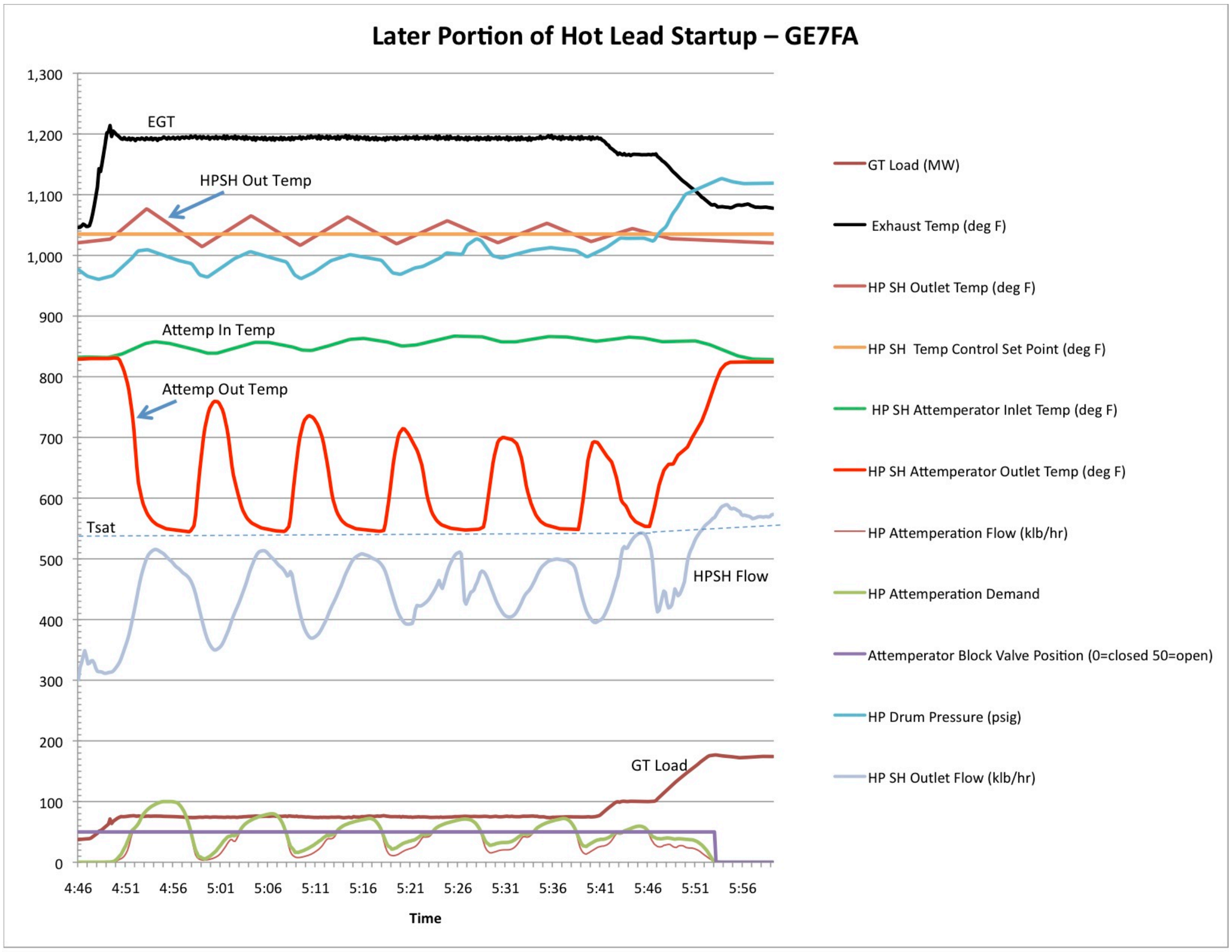

Figure 19 [3]

Repeated overspray to saturation temperature occurs along with large oscillations in spray valve demand, valve position, spray water flow, attemperator outlet temperature, and SH outlet steam temperature during the later portion of this GE 7FA GT hot startup. The oscillations in drum pressure and steam flow are a result of the large changes in spray water flow.

One, or more, of the following, could cause the spray water valve demand oscillations in Figure 19:

- Poor control loop tuning
- Damage to the spray valve trim and/or seat
- Sticky action of the spray valve and/or positioner

Attemperator Protective Logic

Introduction of spray water into the attemperator during periods other than when SH outlet steam temperature requires it results in a high probability of pressure part damage. Protective logic should prevent the spray block valve from opening during startup before SH outlet steam temperature increases close to the unit's temperature rating. Likewise, the spray block valve should be forced closed when SH outlet temperature decreases a moderate margin below the unit's rating. A moderate time delay should be included to prevent short cycling of the block valve during startup and shutdown should the SH outlet temperature remain for a period near the "trigger" value. Some units use an EGT value of around 950°F (510°C) to release and lockout the attemperator during startup and shutdown, respectively.

Cooling the steam at the attemperator outlet below 50°F (28°C) superheat significantly increases the probability of damage to downstream pressure parts. An "overspray protection" feature should be included in the protective logic to require a control valve demand runback to ensure the foregoing margin to superheat is always maintained. The spray block valve should not be included in the overspray protection feature. Linking the block valve action directly to control valve demand will result in intermittent spray water flow and severe wear of the block valve's trim and/or seat.

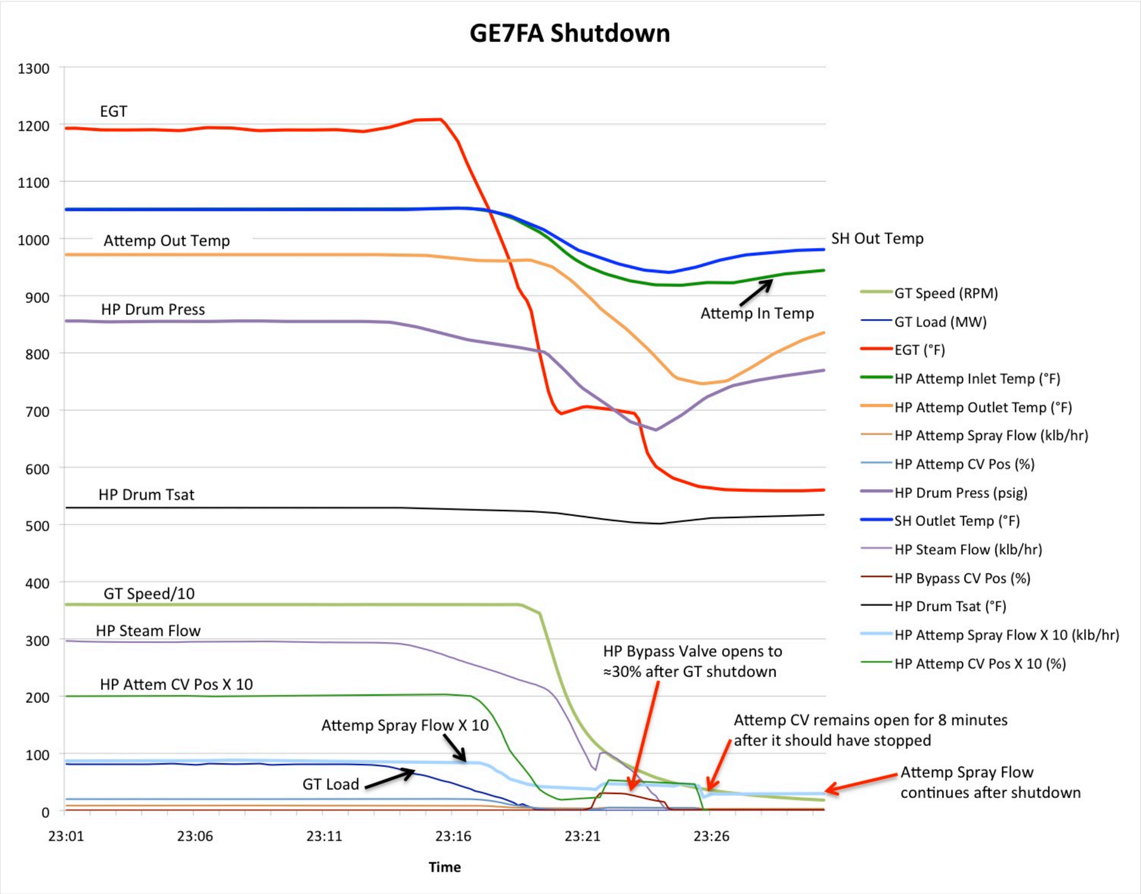

Figure 20 [3]

Inappropriate protective logic permits this SH attemperator to remain in service after GT shutdown.

The attemperator in Figure 20 should have ceased operation when EGT decreased below 950°F (510°C). Instead it continues to operate for another 8 minutes, most of which is after the GT has ceased firing. This inappropriate attemperator behavior is certain to cause pressure part failures. Indication of spray flow continues after the spray control valve closes. It is not known if this indication is correct or an instrument zero error.

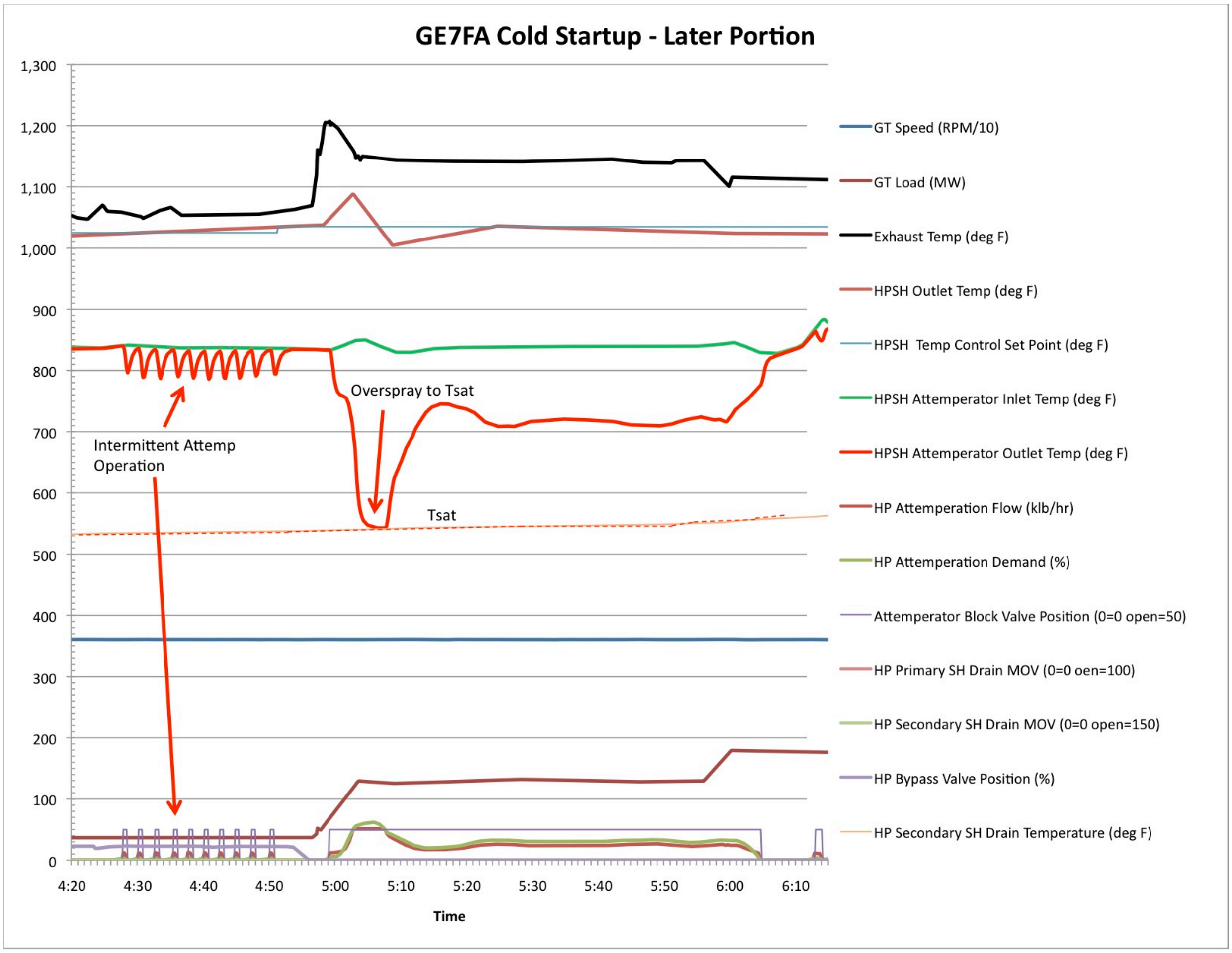

Figure 21 [3]

Intermittent on-off operations of the spray block valve and spray control valve causes unnecessary wear of the valves and thermal fatigue damage to attemperator components and pressure parts.

The protective logic in Figure 21 opens and closes the block valve in response to spray control valve position. The intent of this flawed logic is to protect the control valve seat from wear due to opening and closing against high differential pressure. Instead, the protective logic should be designed to "protect" the much more expensive to repair steam pipes, headers, and tubes. It is more important to protect the block valve seat than to protect the control valve seat. Due to its severe duty the control valve cannot be expected to remain leak-tight. Therefore, the block valve seat must remain tight if damaging spray water leakage is to be avoided.

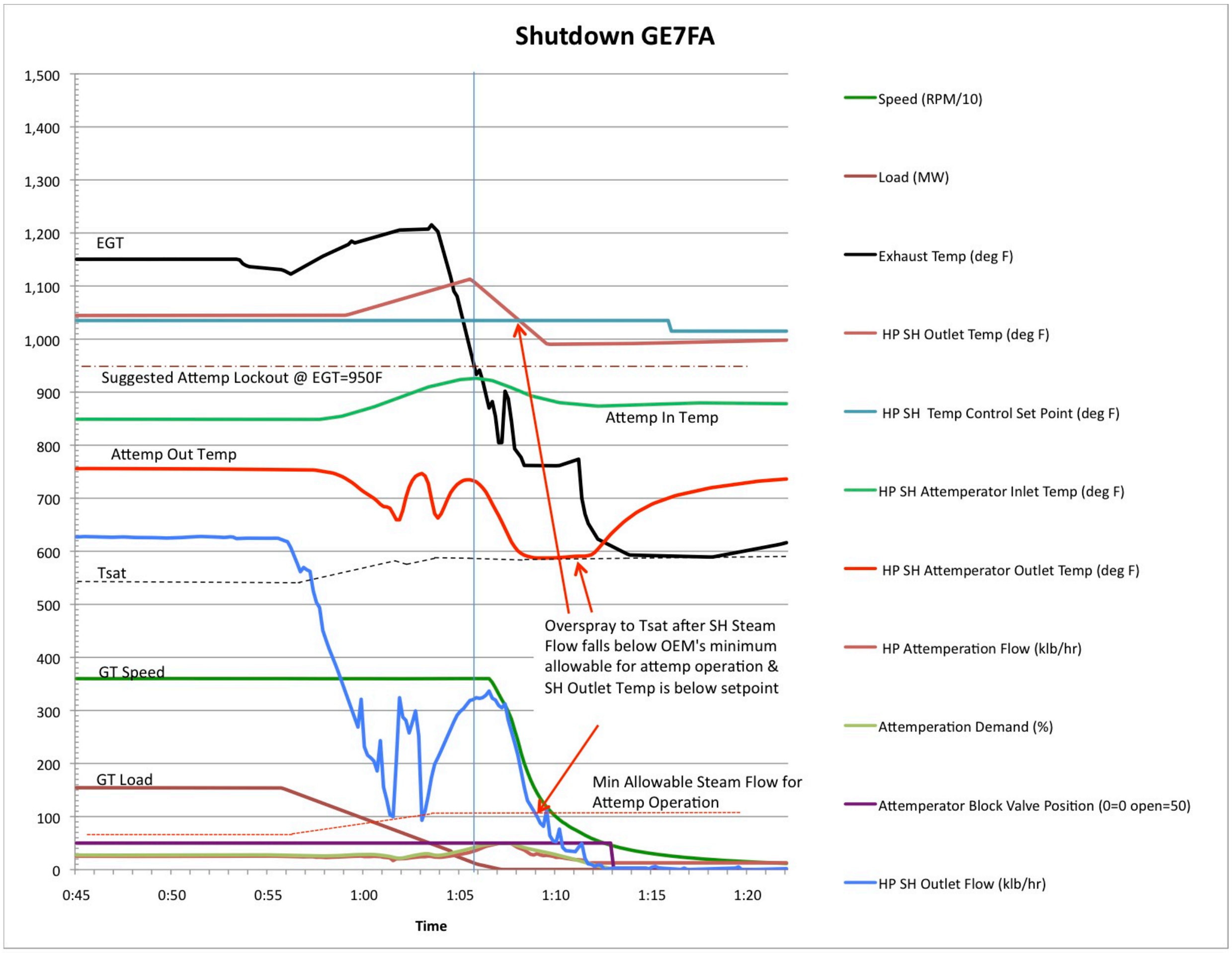

Figure 22[3]

The SH attemperator in Figure 22 continues to operate after EGT decreases below the value where attemperation is necessary. It also continues below the minimum steam flow required for proper evaporation and distribution of spray water.

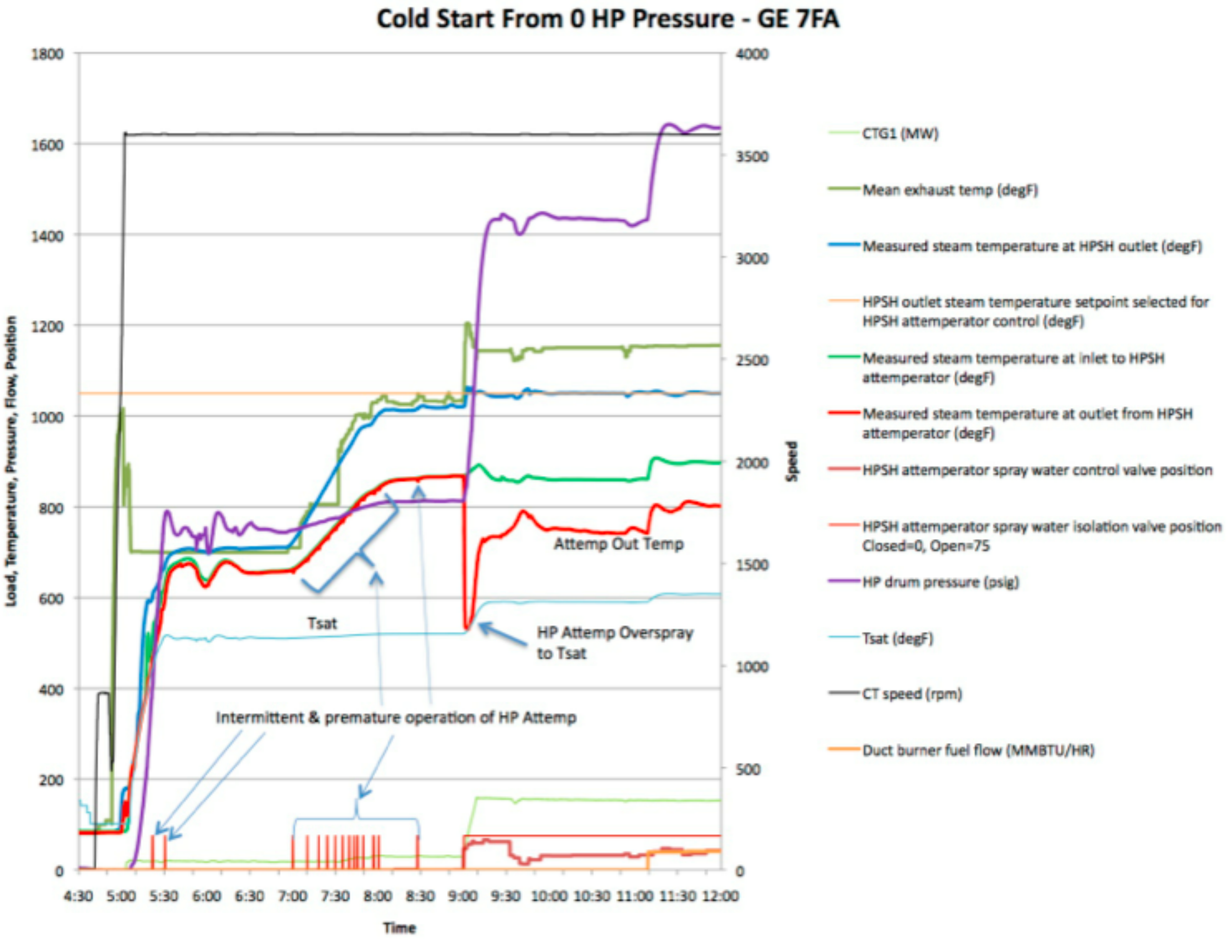

Figure 23 [3]

The protective logic in Figure 23 permits this SH attemperator to come into service before EGT increases to a value where attemperator spray is necessary. It also results in intermittent on-off action like that in Figure 21.

Influence of Operation Actions on Attemperator Performance

A properly designed and installed attemperator using appropriate control logic does not require operator intervention. Unfortunately, many attemperator systems do meet the foregoing criteria. Many HRSGs are equipped with high steam temperature trips or run-backs. If the attemperators in these units are unable to limit SH and/or RH outlet temperature below maximum limits, then the operators typically develop work-around procedures to get the unit on-line. It is not uncommon for operators to reduce the attemperator setpoint, and in some cases manually position the spray control valve, in an attempt to avoid over-temperature operation. Due to the thermodynamic complexity and the speed with which temperature changes occur, most operator manipulation of the attemperator results in significantly more thermal fatigue damage to pressure parts than the over-temperature excursions the operator is trying to avoid would have caused if left alone.

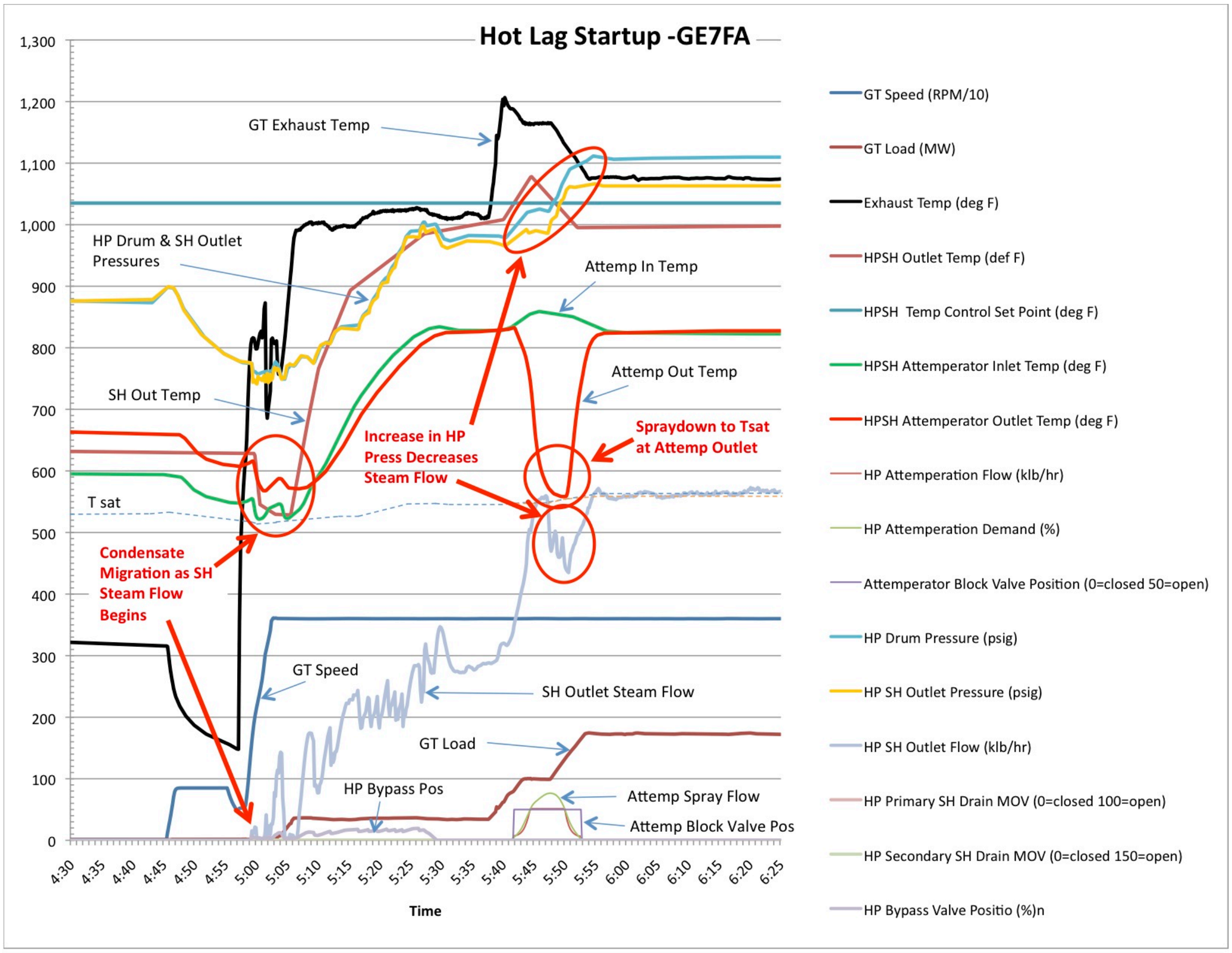

Figure 24 [3]

Rapid increase in HP pressure results in reduction in steam flow. This occurs at the worst possible time – just as the attemperator requires the highest-available steam flow to maintain proper spray water evaporation and distribution.

In units with marginal attemperator performance, such as many units equipped with the GE 7FA and 9FA GT, significant increase in steam pressure during startup should be delayed until after increasing GT load though the "hot zone". Delaying steam pressure increase also results in a lower saturation temperature while passing through the hot GT load zone.

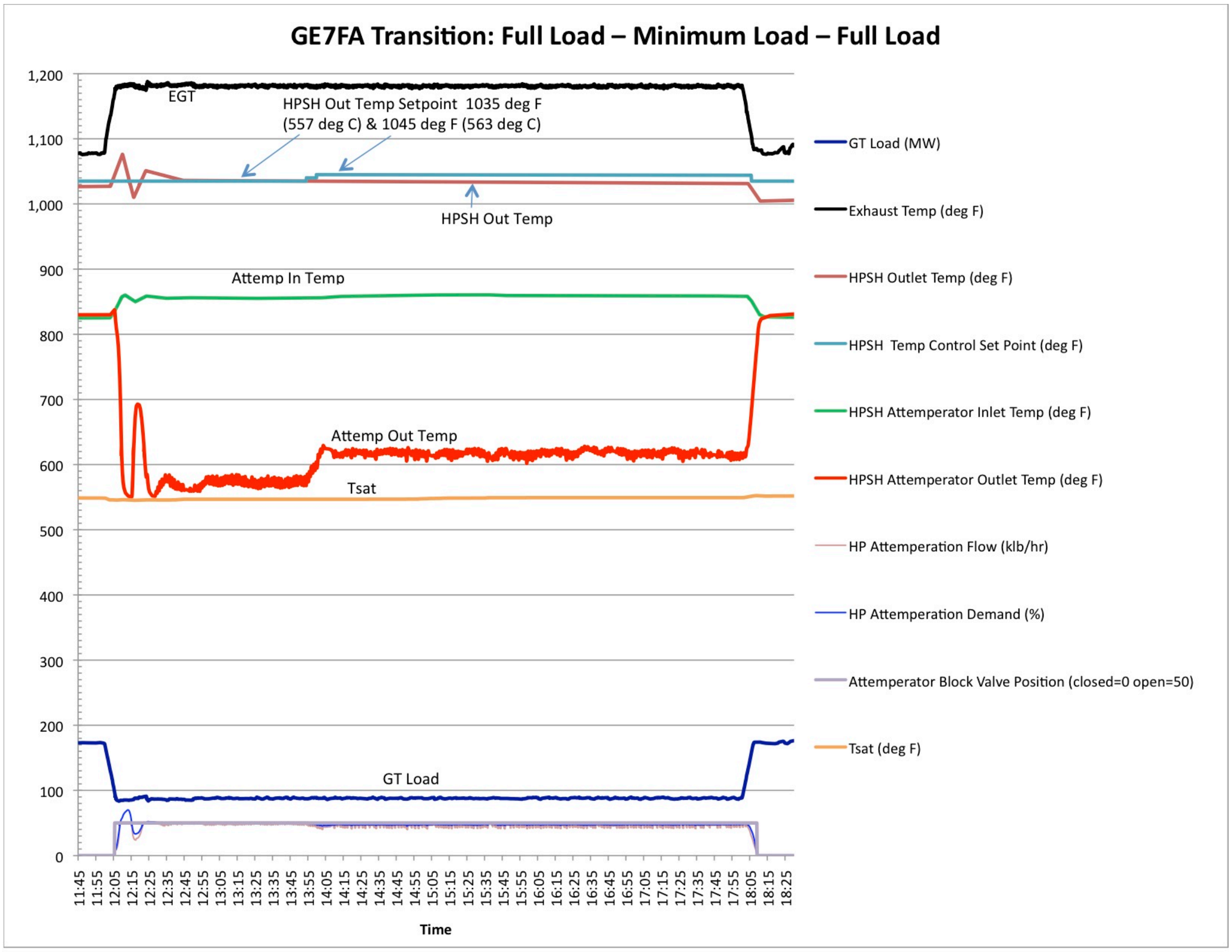

Figure 25 [3]

Transition from full to minimum GT load results in transient overspray to saturation temperature. An increase of 10°F (5.6°C) in attemperator setpoint results in an increase of about 70°F (39°C) at the attemperator outlet.

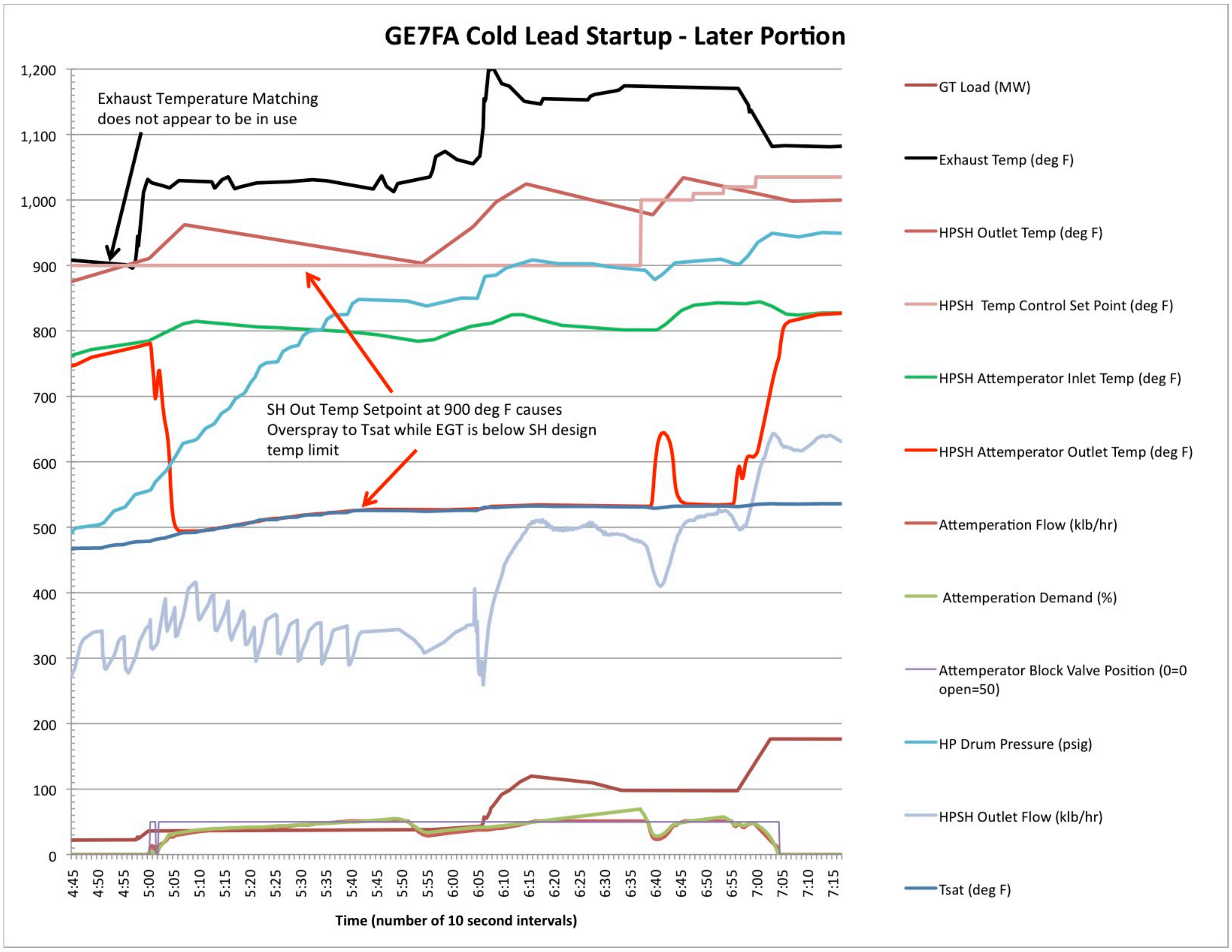

Figure 26 [3]

Attemperator setpoint at 900°F (482°C) results in continuous overspray to saturation temperature. Increasing the setpoint to the unit's rating of 1050°F (566°C) restores adequate superheat at the attemperator outlet.

Attemperator Controls

Many units are equipped with simple feedback loop controllers. See Figure 1. This type of attemperator control is adequate for conventional boilers and many aero derivative and E-Class HRSGs. However, they are not well suited for use on F-Class HRSGs – particularly those equipped with a GT having an aggressive exhaust gas temperature profile.

If the attemperator hardware is of reasonable quality and installed within a steam pipe of adequate length, conversion to a well-tuned cascade control scheme will often resolve attemperator performance problems. Figure 27 shows the schematic for a typical cascade controller.

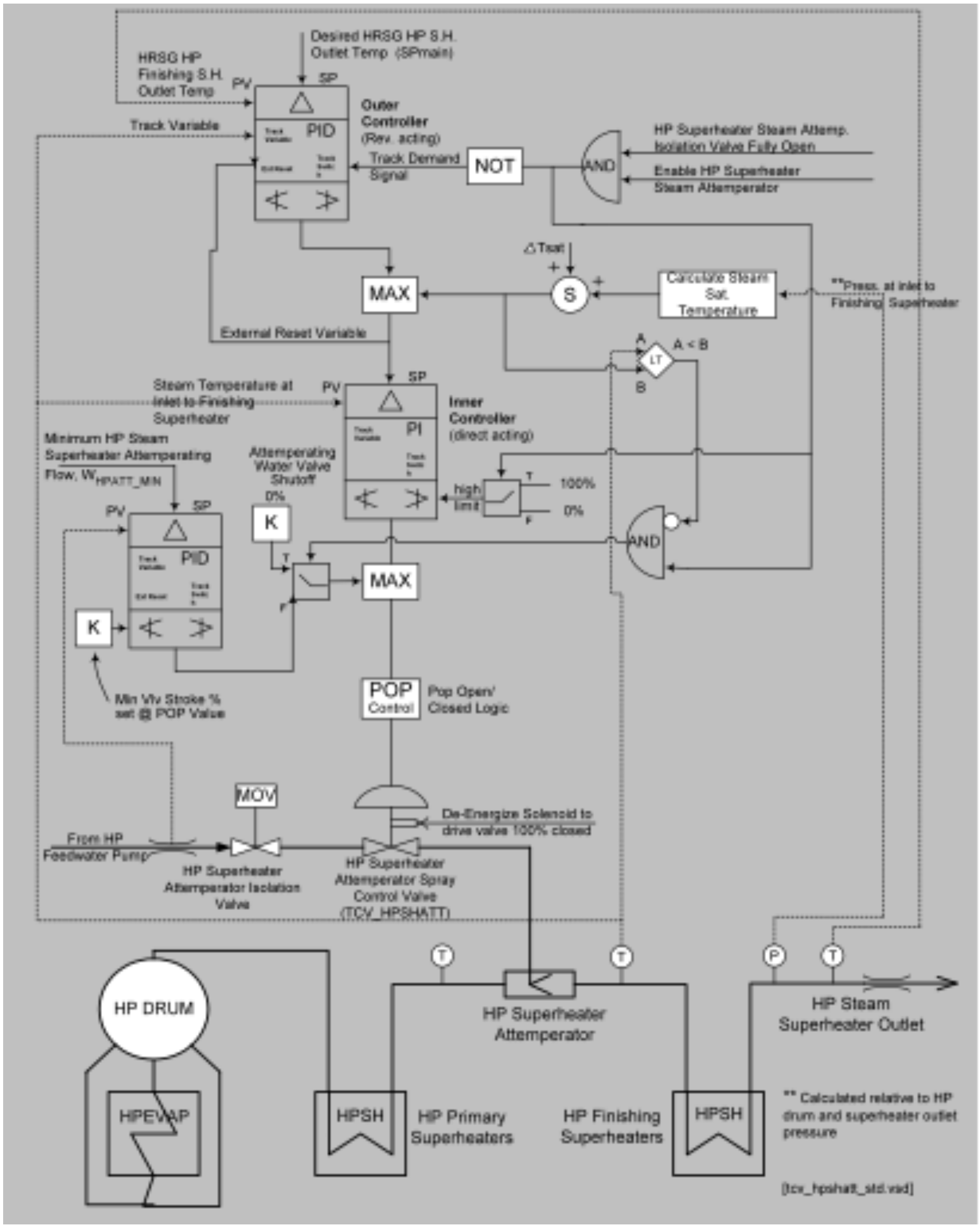

Figure 27 [4]

Schematic diagram of a cascade attemperator controller used on a F-Class HRSG

The cascade controller in Figure 27 is comprised of two control loops. The “outer” (or master) controller uses the following key inputs and outputs:

- Setpoint = desired SH outlet steam temperature
- Measured Process Variable = SH outlet steam temperature

- Controller Output = inner (or slave) controller setpoint

The inner controller responds faster than the outer controller and uses the following key inputs and outputs:

- Setpoint = desired attemperator outlet temperature
- Measured Process Variable = attemperator outlet temperature
- Controller Output = Spray water control valve position demand

In unusually demanding conditions, such as those resulting from the GE 7FA and 9FA exhaust temperature characteristic, it is often necessary to send GT fuel demand or inlet guide vane position to the outer controller so it can anticipate imminent changes in EGT.

Attemperator Design Considerations

Adequate attemperator performance during challenging operating conditions requires proper installation of high quality attemperator hardware. The following design requirements are critical:

- Sufficient straight steam pipe length upstream of the spray nozzle to permit eddies and turbulence induced by tees and elbows to dissipate prior to reaching the nozzle
- Atomization of spray water into very small droplets, uniformly distributed within the steam pipe
- Sufficient straight steam pipe length downstream of the spray nozzle to permit complete evaporation of spray water
- Thermal liner to protect steam pipe from large thermal transients
- Spray water turndown ratio up to 50:1
- Spray water control valve located external to spray nozzle

Obtaining the high turndown ratio required typically requires the use of variable area spray nozzle orifices or sequentially operated multiple orifices. Multi-orifice spray nozzle attemperators with the control valve trim located inside the nozzle mast have proven unreliable in high temperature HRSGs.

A rough determination if an attemperator is installed with sufficient straight pipe length to permit compete evaporation of spray water can be made my as follows. The distance between the attemperator spray nozzle and the upstream pipe elbow or tee should be a minimum of 5 times the steam pipe diameter. The distance between the attemperator spray nozzle and the first downstream pipe elbow or tee should be a minimum of 20 times the steam pipe diameter. Figure 28 shows a more quantitative method of determining minimum acceptable straight pipe length.

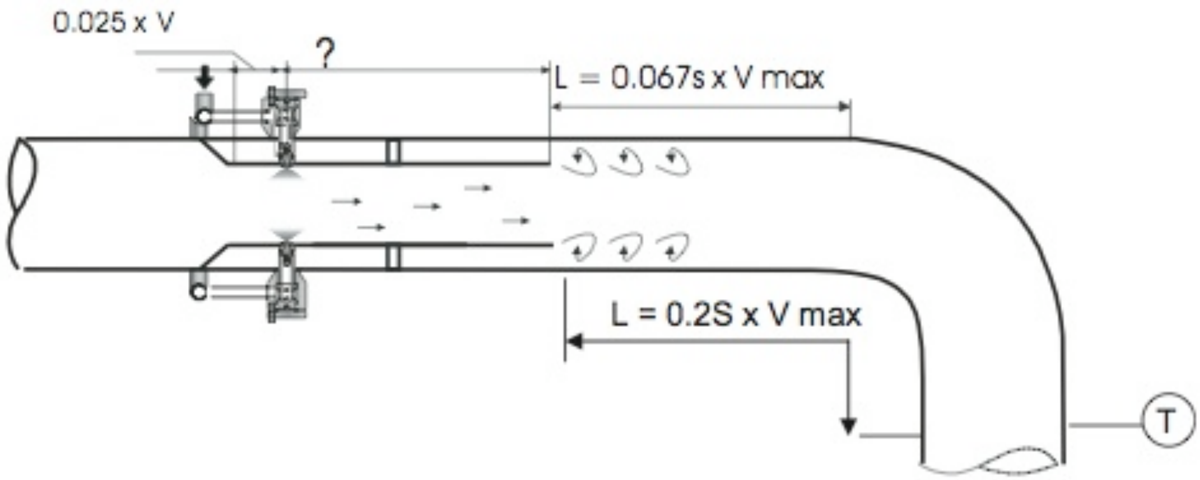

Figure 28 [4]

Minimum straight steam pipe length required downstream of the spray nozzle to ensure complete evaporation is a factor of resident time and steam velocity. (courtesy of CCI)

Figures 29 through 31 show some common attemperator hardware.

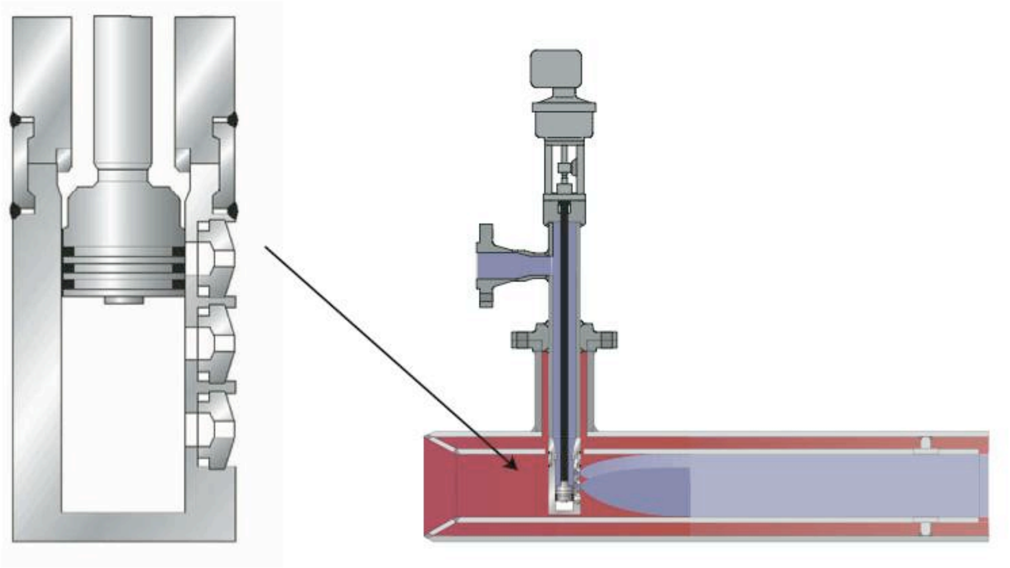

Figure 29 [4]

This attemperator achieves a high turndown ratio by using the control valve trim to sequentially open multiple fixed orifices. Because of its location inside the nozzle mast, the control valve trim is exposed to large thermal transients that result in spray water leakage and sticking of the valve trim. The multi orifice nozzle assembly is also prone to cracking. This style of attemperator has proven to be unreliable in high temperature applications.

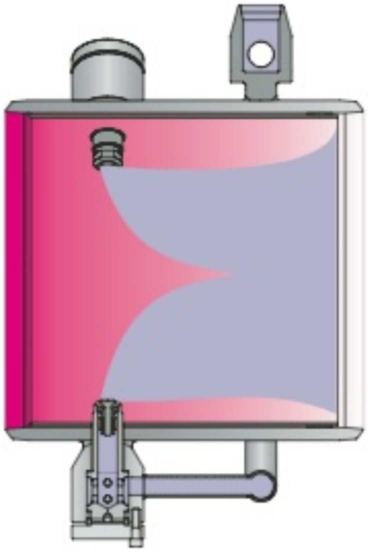

Figure 30 [3]

This attemperator injects spray water perpendicular to steam flow to maximize secondary atomization. It is often capable of achieving complete evaporation of spray water in a shorter pipe-length than nozzles that inject water in-line with steam flow. (courtesy of CCI)

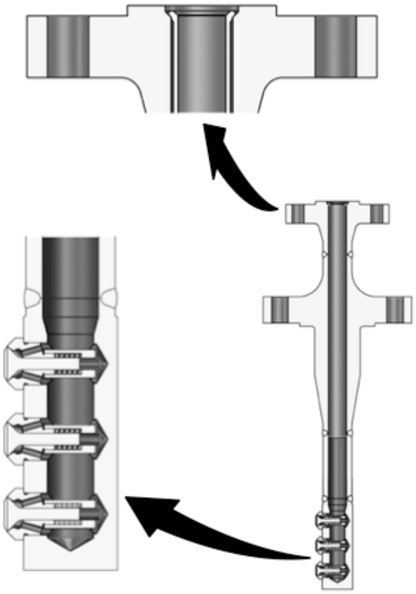

Figure 31 [4]

This mast style nozzle uses variable area orifices to achieve fine primary atomization over a wider range of spray flow than fixed nozzles could achieve.

Heating Surface Distribution

The distribution of total SH heating surface area between the primary and secondary SH has a significant impact on attemperator performance. In fact, some HRSGs are installed with a SH surface area distribution that makes startup without exceeding the maximum SH outlet steam temperature limit thermodynamically impossible. The very important priority of avoiding overspray conditions limits the amount of spray water that can be injected by the attemperator. If the secondary SH surface area is too large the outlet steam temperature cannot be limited within the unit's rating.

The temperature rise split (TRS) between the primary and secondary SH is useful in determining how demanding the attemperator's job will be. TRS is expressed as a ratio and calculated using the following equation:

TRS = (Pri SH Out Temp – Tsat) / (Sec SH Out Temp – Tsat) X 100 : (Sec SH Out Temp – Pri SH Out Temp) / (Sec SH Out Temp – Tsat) X 100

or

$$\mathrm{TRS} = \Delta T_{\mathrm{primary}} \div \Delta T_{\mathrm{total}} : \Delta T_{\mathrm{secondary}} \div \Delta T_{\mathrm{total}}$$

The temperature values used to calculate TRS are measured with the GT at full load, zero duct burner firing, and zero attemperator spray flow. The TRS for the RH is calculated in the same way, but is rarely a problem due to the very low saturation temperature in the RH system. Figures 32 and 33 show SH TRS calculations for two different HRSGs.

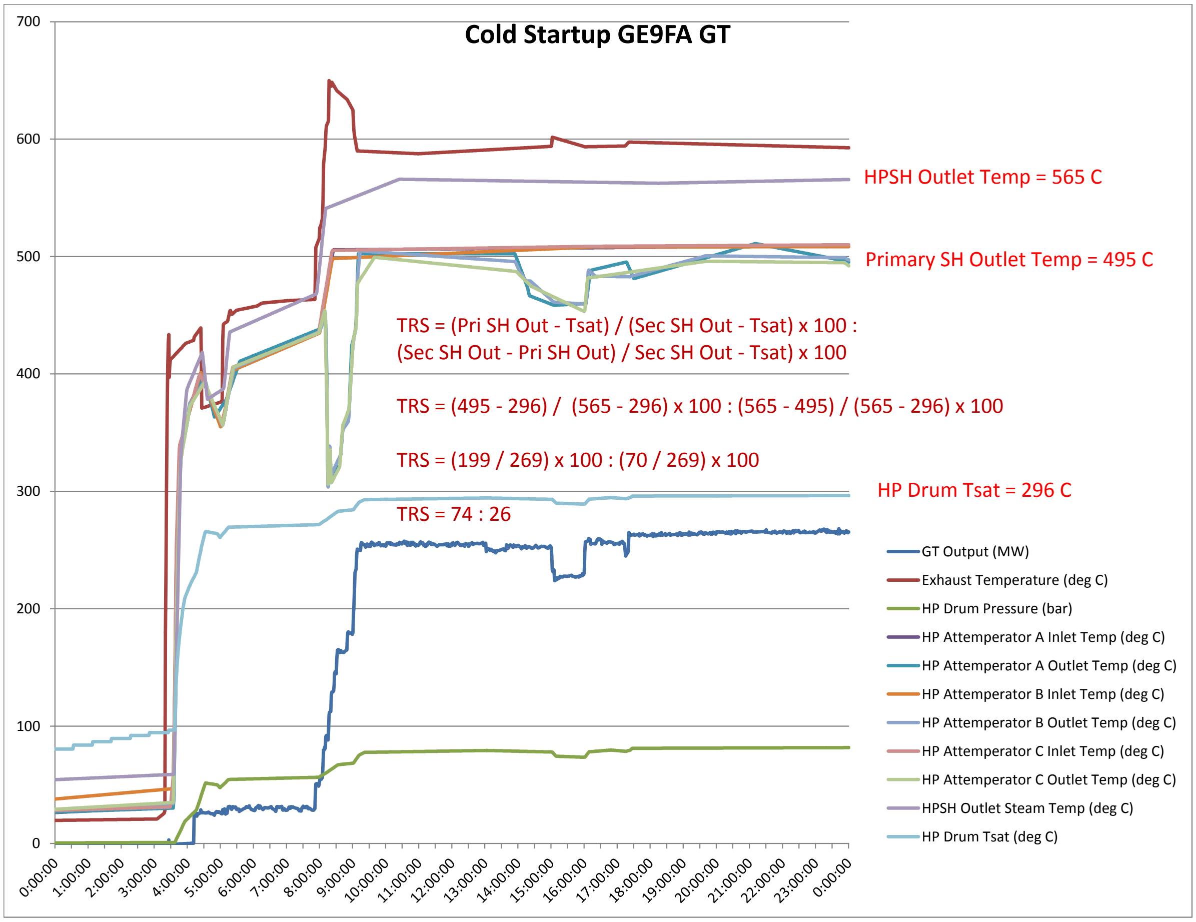

Figure 32[3]

This HRSG has a SH TRS of 74:26. The relatively small temperature rise in the secondary SH will not make the attemperators job particularly difficult if it is properly designed, installed, controlled and maintained.

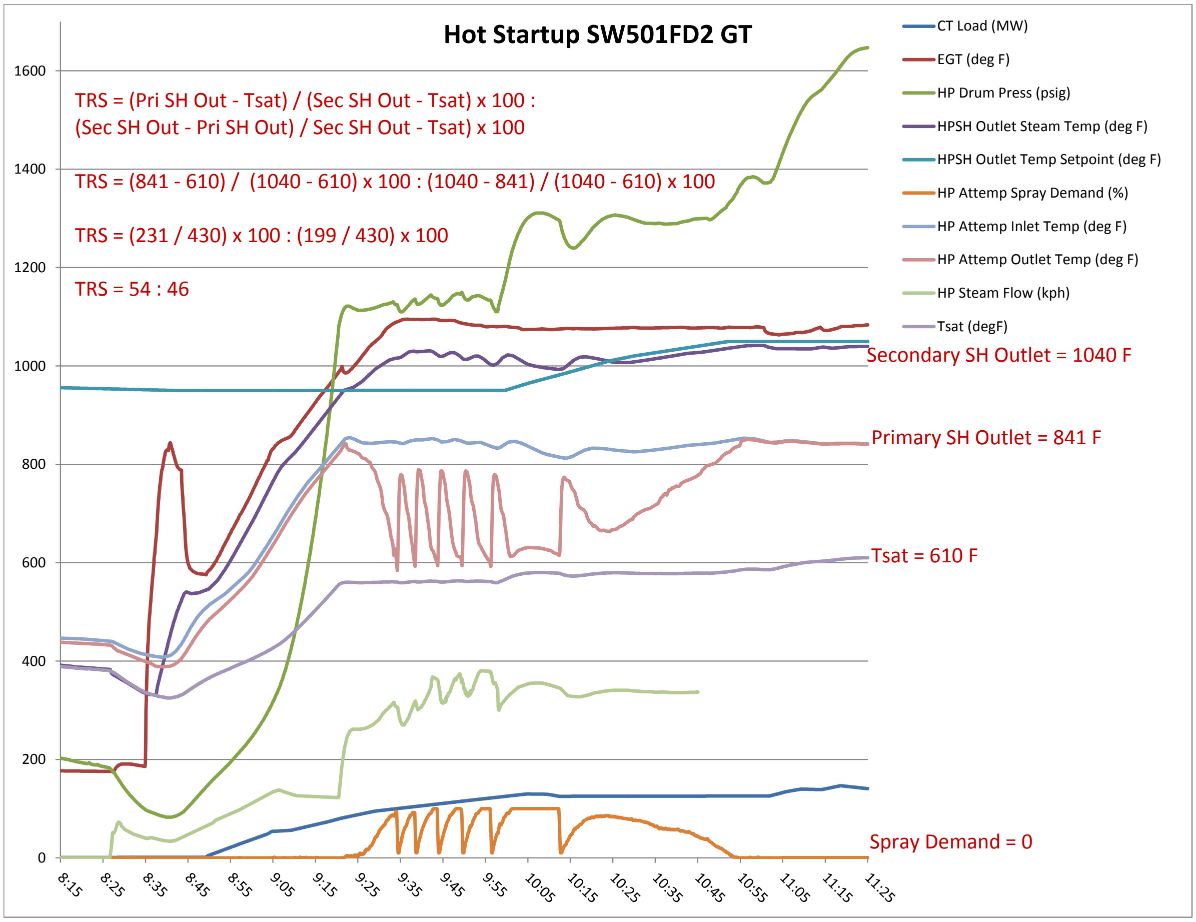

Figure 33[3]

This HRSG has a SH TRS of 54:46. The relatively high temperature rise in the secondary SH will require very good performance from the attemperator and its controls if overspray and SH outlet temperature excursions above the maximum limit are to be avoided.

Figures 33 and 34 demonstrate quantitatively how TRS impacts the thermal head available to the attemperator to evaporate spray water.

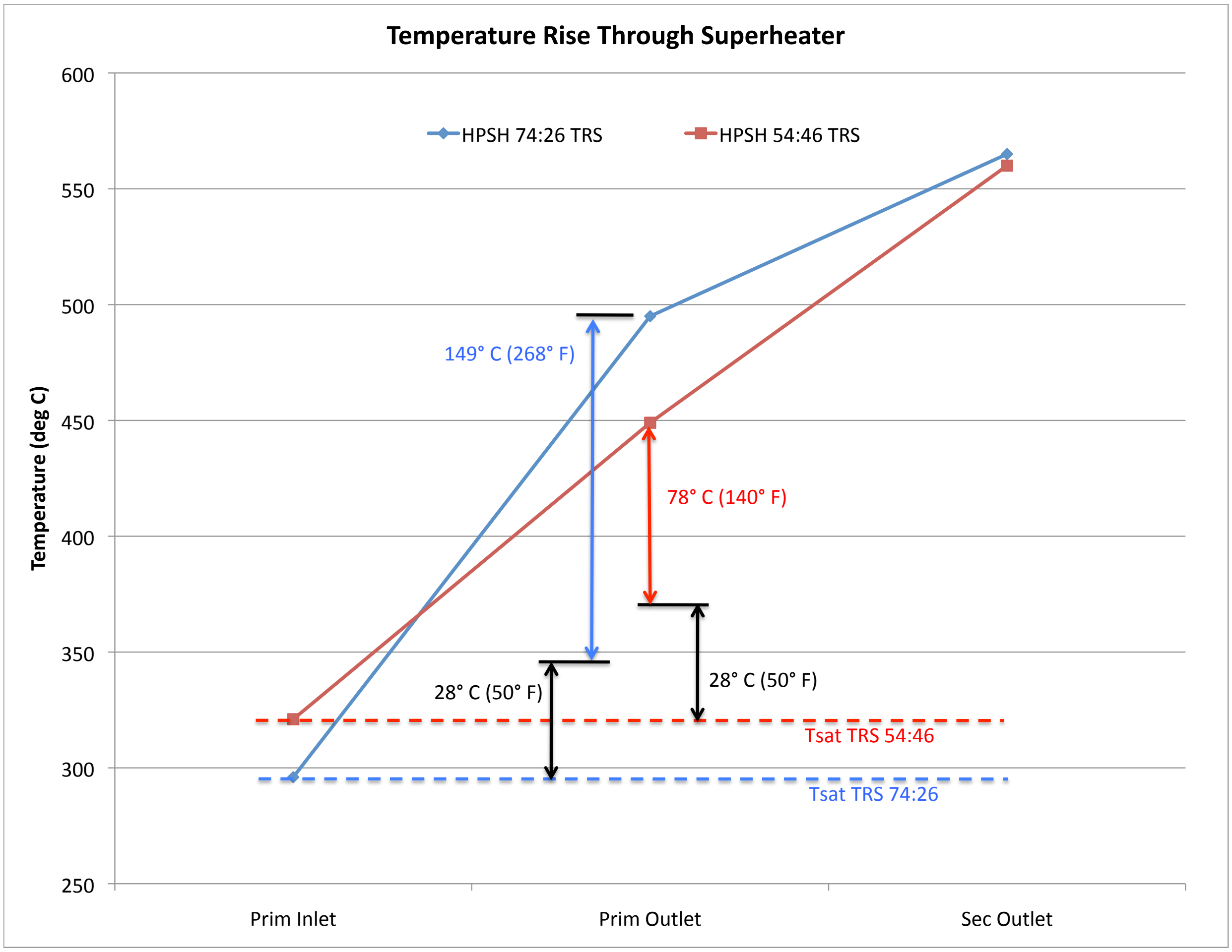

Figure 33 [3]

After subtracting the 50°F (28°C) superheat required to avoid overspray conditions the SH TRS of 74:26 provides 268°F (149°C) of thermal head that can be used to evaporate spray water. The 54:46 TRS SH only provides 140°F (78°C) of thermal head for evaporation of spray water.

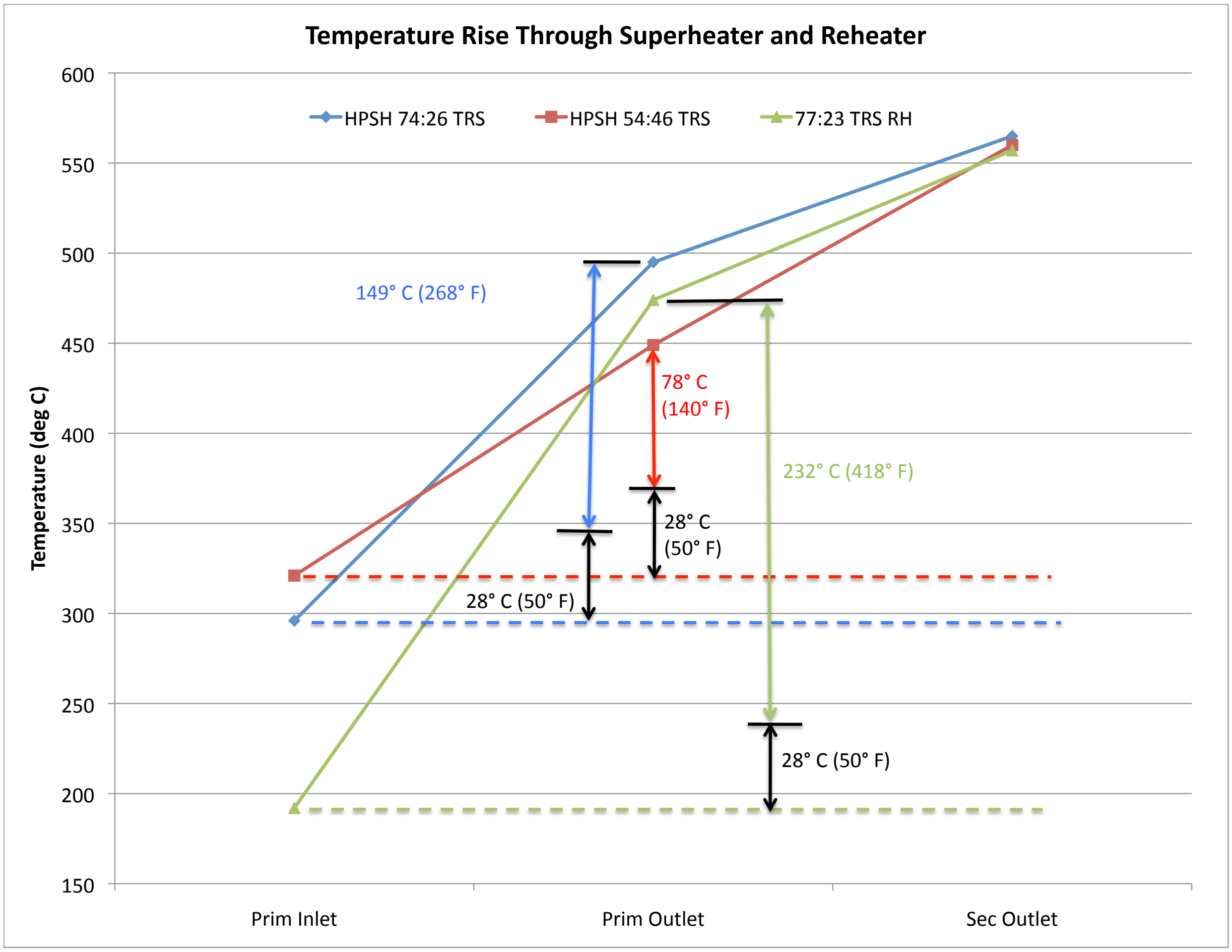

Figure 34 [3]

The relatively low saturation temperature and 77:23 TRS in this RH results in 418°F (232°C) of thermal head for evaporation of spray water.

Conclusions

Attemperators installed in F-Class HRSGs have a challenging job to perform. Their design, hardware quality, installation details, control logic, and protective logic must all be properly executed if premature pressure part failure is to be avoided. Once properly installed and tuned, the attemperator block valve, control valve, spray nozzle, and thermal liner must be regularly inspected and repaired as necessary if acceptable performance is to be maintained. Existing attemperator problems can be identified, and early detection and correction of problems in otherwise good performing attemperators can be accomplished via routine review of readily available DCS data.

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