ROOT CAUSES OF TRANSIENT TUBE TEMPERATURE ANOMALIES MEASURED IN HORIZONTAL GAS PATH HRSGS

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Abstract

Thermally-induced forces developed by large transient temperature differences between vertical tubes attached to the same inlet and outlet headers have caused low cycle fatigue failures in tubes in proximity to the weld attaching tube to header in economizers and superheaters of several designs of horizontal gas path (HGP) HRSG, in some cases after of the order of 10^2 thermal cycles. Examples from tube temperature measurements recorded by thermocouples attached to many superheater, reheater and economizer tubes of HGP HRSGs of differing designs are provided to illustrate the large differences in temperature that can occur transiently during startups and shutdowns, and in the case of economizers also during normal operation. Generic root causes of large differences in temperature of tubes intended to remain at closely similar temperatures at all times are discussed, with suggestions how the resultant cyclic thermally-induced stresses at tube attachment to header can be mitigated on existing HRSG installations by minor modifications to equipment and by better-informed operating procedures during startups and shutdowns, and eliminated on new installations by more enlightened design of HRSG and also of balance of plant systems that influence the transient conditions imposed on the HRSG during startup and shutdowns.

Introduction

The susceptibility of HRSGs to high thermally induced stresses during shutdowns and startups has been identified as a major threat to their reliability and durability, especially if subjected to some cycling operation [1,2]. Two mechanisms are the principal cause of thermal fatigue-related damage in HRSGs, the second of which is the principal subject addressed by this paper.

Vertical gas-path (VGP) and horizontal gas-path (HGP) HRSGs are equally vulnerable to thermally-induced fatigue cracks at the inner surfaces of tube and pipe penetrations through header wall where these intersect the bore of higher temperature headers of HP superheaters and reheaters [1]. Extensive cycling operation without cracking at inner surface of tubeholes in these headers is feasible, but only if the headers that operate at higher temperatures are appropriately designed and, - crucially important even for relatively thin superheater headers, - the operating procedures during shutdowns and startups pay attention to prevention of large, rapid changes in the temperature of the fluid in contact with the inner surfaces of header tubeholes. Many large HRSGs installed with relatively thick superheater headers of P22 material have limited cyclic life, even with correct operating procedures for shutdowns and startups. In addition the operating procedures widely used for shutdown and start up of F class combined cycle units are extremely damaging to HP superheater headers. Although to date there have been no reports of internal cracking in superheater headers of large HRSG, few, if any, have been inspected internally due to difficult access. It is quite likely that some HRSGs in “F-class” combined cycle plants with the thicker P22 HP superheater headers that have performed a few hundred shutdown-startup cycles with widely used shutdown and startup procedures [1] have already initiated and are growing cracks from inner surfaces of some higher temperature superheater headers.

The root cause of the second failure mechanism in HRSGs involving fatigue is thermally induced forces developed in individual tubes prevented from thermal expansion relative to the other tubes attached to the same pair of headers and at different temperature. VGP HRSGs usually have predominantly horizontal tubes in serpentine arrangements that are flexible and can accommodate even large differential thermal expansions between adjacent tubes without developing significant forces in tubes when at different temperatures. However, the predominantly straight vertical tubes of HGP HRSGs are mostly arranged in rows between flexurally stiff upper and lower headers which inhibit thermal expansion or contraction of tubes if they, even fleetingly, deviate in temperature from the average temperature of the other tubes attached to the same pair of headers. The tensile and compressive forces induced in individual straight tubes develop thermal stresses in the tubes that are usually highest at the tube attachment weld to each header due to stress concentrating features at the weld discontinuity. The highest thermally induced stresses caused by tube-to-tube temperature differences usually occur in arrangements that offset the connection of some tubes from the vertical centerline of one or both headers. The thermally-induced forces along the axis of offset tube connections develop forces and bending moments at the tube attachment to the header, which significantly intensifies the maximum stress at the tube extrados or intrados of the outer surface of the attachment weld to header or sometimes at the extrados or intrados of a tight radius bend connecting tube to header[1].

Fatigue failures in tubes in proximity to the weld attachment to headers attributable to transient tube-to-tube temperature differences have been most prevalent in superheater and economizer tubes [3]. In some cases, the tube failures have occurred as early as of the order of 10^2 thermal cycles in offset tubes. The failures in superheater tubes are most likely to be low cycle fatigue (LCF) exacerbated significantly by the interaction of creep and fatigue [1]. Failures at welds attaching economizer tubes to headers may be either LCF cracks that initiated in proximity to the weld on the outer surface of the tube and then propagated by LCF through the tube wall, or corrosion-fatigue initiated on the inner surface of the tube at a weld root defect or the notch of a partial penetration type of weld, and then propagated outwards through the tube wall. The incidence of similar failures is likely to become more widespread and frequent as larger HGP HRSGs accumulate more shutdown-startup cycles.

The primary root cause of fatigue-related failures in proximity to tube attachments to headers of superheaters, reheaters and economizers of HGP HRSGs is large transient temperature differences between tubes which the design concept has presumed will at all times remain at closely similar temperatures. Examples selected from measurements of many tube temperatures on several different designs of HRSG have been selected to explain some of the mechanisms identified as primary causes of transiently large temperature differences between tubes. The paper also identifies design features that intensify thermally induced stresses and contribute to the premature fatigue-related tube failures.

Tube Temperature Anomalies in HP Superheaters and Reheaters

The most widespread cause of tube temperature anomalies in HP superheaters and reheaters is failure to completely remove all condensate from lower sections of tubes, headers, steam pipes, manifolds, attemperators, etc, during CT/HRSG startups prior to establishing any steam flow from superheater or reheater. Figure 1 highlights during a typical cold start that even though superheater and reheater drains had been opened long before the startup and remained open, when a small flow of HP steam was established to warm the HP steam pipes, the consequential small pressure drop through the superheater caused condensate and saturated steam mixture to be displaced forward and upwards selectively through some tubes of the final steam outlet row of tubes, chilling those tubes at their outlet by up to 135˚F relative to other tubes which continued to follow the rise in gas temperature. Even though the condensate or saturated steam has been heated during passage upwards to the thermocouple position at the top of the 65 foot long tubes, tube #5 remained at saturation temperature at tube outlet for about 15 minutes. Notwithstanding that this HRSG is nominally provided with drains at all low points of superheaters, and the blowdown vessel is installed in a pit to facilitate continuous drain pipe fall to the vessel, a substantial quantity of condensate could not be removed and was finally cleared into the hotter outlet pipe and header when the HP bypass valve opened and established a larger steam flow. A similar phenomenon was measured on tubes of all earlier sections of the superheater.

During the same typical cold start, a similar phenomenon occurred in all sections of the reheater which were also expected to have completely drained prior to startup. The steam outlet section of the reheater comprises two staggered rows of upflow tubes with both rows attached to the same upper and lower headers with each tube offset from one of the headers, which significantly intensifies the maximum thermal stresses caused by tube-to-tube temperature differences. Figure 2 highlights that during the initial part of the startup when conditions inside the reheater were stagnant the uncooled row A tubes which are exposed to inlet gas flow heated up quicker than the row B tubes. Some reheater row A tubes were up to 100˚F hotter than adjacent row B tubes, which may be large enough to cause localized inelastic stresses, thus creep-fatigue damage, at the weld of offset tube to the header. Even more disturbing, when forward flow was established through the reheater during pressurization from HP bypass steam flow to the cold reheat pipe, undrained condensate remaining in lower sections of the reheater was displaced forward and blown up selectively through some of the row A tubes. Tube #A18 was chilled at outlet 320˚F below most of the nearby tubes, and severe inelastic stresses occurred at the weld of this offset tube to the upper outlet header which are likely to initiate creep-fatigue cracks in the weld after a small number of similar cycles. Some other row A tubes were also chilled at outlet by condensate or saturated steam. More condensate was displaced forward for several minutes through a few tubes after hot reheat steam flow was initiated. The phenomenon was never observed in any row B tubes, which at inlet are offset towards the side of the header, and tended to be more prevalent on row A tubes with inlets positioned in proximity and opposite one of the header inlet pipe branches and in tubes near to one or other blind end of the header, suggesting that the headers tilt from horizontal during the shutdown and startup. Similar behavior was observed at tubes in earlier stages of the reheater.

The phenomenon of selective chilling of some tubes by condensate immediately after establishing forward flow and pressure gradient through the HP superheater has been observed in tube temperature measurements on several different designs of HGP HRSGs during cold, warm and hot starts. Tube temperatures recorded during a typical hot start on a different design of large, non-reheat HRSG are illustrated in Figures 3 and 4. The final outlet section of this superheater has three rows of tubes with 18 tubes in each row, arranged in-line parallel to the direction of gas flow, Fig 3. Row B tubes are perfectly straight and connected at bottom- and top-dead centers of upper and lower headers, respectively. Row A and row C tubes are each offset and connected by small radius bends to upper and lower headers, which intensifies the maximum thermal stress developed by any thermally-induced forces in these tubes. Prior to establishing any flow in the superheater, row A tubes which are most directly exposed to the inlet gas flow, were heated up to 120˚F above the row B tubes. At the steam inlet to the second stage HP superheater, positioned in the steam path after the desuperheater, the row A tubes of this panel were heated more than 200F above the row B tubes, Fig, 4, and developed high thermally induced stresses at the attachment of the offset row A tubes to the headers. When steam flow was initiated, a substantial quantity of condensate was blown forward through many tubes in all sections of the superheater, Figs. 3 & 4, chilling tubes and quench cooling the tube holes through the outlet header wall. This HRSG was installed without any means for removing condensate when the unit is warm or hot.

Remedies For HP Superheaters and Reheaters

Management of Condensation.

Substantial quantities of condensate that cannot be completely drained from superheaters and reheaters caused high tube-to-tube temperature differences and also thermal shock at the tubeholes and inner surface of superheater and reheater outlet headers. Thus a primary root cause of damaging thermal stresses in superheater and reheater tubes at their attachment weld to the header and also of damaging thermal stresses at the inner surface of tubeholes is deficiencies in the design and installation of superheater and reheater drainage arrangements. Superheater and reheater drains are not merely for maintenance as sometimes assumed. It is essential that they are utilized correctly during every CT startup and shutdown.

Deficiencies in installed arrangements for drainage and disposal of condensate from superheaters and reheaters can usually be corrected by design modifications. Modifications are prudent for all HRSGs with deficient installed arrangements. They are essential for units expected to perform some periods of cycling operation to delay thermally induced failures in HRSG pressure parts and HP steam pipes and valves. Common deficiencies in the design of drains removal and disposal systems that must be avoided include: -

- Drains from different sections of the superheater are often incorrectly interconnected into a single common drain pipe to the blowdown vessel, Fig. 5, because that is the cheapest installation and the importance of reliable condensate drainage to cyclic life of superheater and reheater components is not generally appreciated. Drains from different sections of superheater, (likewise for reheater), which operate at any time at different pressures should never be interconnected. The correct arrangement and sizing for the superheater drains is illustrated in Fig. 6. The drains from each different section must be connected directly to a manifold on the blowdown vessel and have separate, duplicate, motor-operated isolation valves to ensure tight shutoff is maintained. Much of the damaging condensate blown out of the final superheater tubes in Figure 1 came from earlier, separately-drained sections of the superheater through the superheater drain pipes which had been interconnected into a single small bore pipe which was supposed to drain all condensate to the blowdown vessel, as illustrated in Fig. 5. As a result of interconnecting, in this example, ten nominally 1" diameter drain pipes taken from 3 different superheater sections into a single 1" diameter longer pipe of identical flow area, then when steam flow and thus pressure drop was established through the superheater the path of least resistance route for condensate from the higher pressure sections of the superheater was through the six 1" drain connections intended to remove condensate from the final section of the superheater, rather than through the single 1" pipe to the blowdown vessel.

- Increase the drain pipe bore after each interconnection from locations always operated at the same pressure to maintain uniform mixed-phase fluid velocities along the pipe.

- Drain connections on lower headers and pipes and valves to the blowdown vessel must be generously sized to pass condensate at the maximum rate of formation.

- Drains are often undersized on the false assumption they pass only liquid. Condensate close to saturation temperature flashes after it enters the drain pipes and the pressure reduces. Pipes and isolation valves must be sized to pass the much higher specific volume of mixed phase flow.

- There should be a minimum of two drains on all lower headers of superheater and reheater that are notionally horizontal located in reasonable proximity to the blind ends of the header to ensure that condensate can be completely removed when the header deviates from its intended horizontal position during cooling or heating of the panels. Additional intermediate drain points are recommended for longer headers to minimize the quantity of condensate that can pool along the bottom of the header.

- The elevation between all drain connections on lower headers, pipes and manifolds of superheater and reheater sections and the inlet manifold on the blowdown vessel must be sufficient to ensure adequate fall along the entire length of each drain pipe between drain point and blowdown vessel when the HRSG is hot and the drain connections on headers and pipes have expanded down to their lowest operating position.

- Operation of isolation valves on drain lines from superheaters and reheaters should be automated to ensure they are always opened when conditions are conducive to condensation in the tubes of superheater or reheater and kept shut at all other times to conserve superheater and reheater pressures.

Remedies for Other Weaknesses.

Figures 2, 3 and 4 all highlight that significant temperature differences of between 100˚F and 200˚F developed in different designs of HRSG between the first and second rows of tubes attached to the same pair of headers during the initial stages of CT startup before a cooling flow of steam could be established through the tubes. Inevitably, when more than one row of tubes is attached to the same upper and lower header, at least one of the tubes will be offset which significantly increases the maximum thermally-induced stress at the attachment weld of the offset tube to the header.

For new installations intended for some periods of cycling operation, provided also that superheater and reheater drains are designed to assure rapid, complete removal of all condensate produced during the shutdown and startup, adoption of a single row of straight tubes connected to bottom- and top-dead center of upper and lower headers, respectively, for superheaters and reheaters eliminates the second source of transient differences in tube temperature highlighted by extensive measurement of tube temperatures during shutdowns and starups. Avoidance of offset tubes and the potentially large amplification of maximum stress from the thermally-induced bending moment applied at the connection of offset tube to header, gives additional security against tube failures in proximity to headers of superheaters and reheaters.

For HRSG designs proposed for installations specified for some cycling service that have tube and header arrangements other than straight vertical tubes with no offset, then design verification that the proposed design is capable of performing the projected lifetime number of shutdown-startup cycles without risk of tube failure in proximity to the point of attachment to the headers is strongly recommended. Critical to the reliability of the tube cyclic life prediction are: -

1. The conservatism of the assumptions made for the worst case tube-to-tube temperature distribution.

2. The realism of the two FE models necessary, the first a global model of tubes, headers and supports to establish the forces and moments at the point of attachment of tubes to header,

and the second to calculate the maximum stress developed in the tube in proximity to the attachment weld.

3. Allowance in full for the interaction of creep and fatigue during the dwell period at high temperature while on load. This substantially reduces the low-temperature endurance of the material [1].

Tube Temperature Anomalies in Economizers

Anomalies Attributable to Deficient Venting Arrangements

Many HRSGs have high points in the economizers that have no provision for venting of air or steam trapped at these high points. The practice of not providing vents at all high points developed from small HRSGs, which had much shorter tubes. Large HRSGs now have tube heights up to 70 feet designed with very low pressure drops through each pass of tubes and these combine to make it much more difficult to, with absolute certainty, completely purge air from every un-vented high point each time flow is first established after filling the economizer. Steaming occurs extensively during every startup in economizer tubes while stagnant for a long period until drum swell subsides. The vapor that collects at un-vented highpoints is also difficult, in some cases impossible, to remove from every tube.

Figure 7 illustrates the conceptual features of a typical hairpin tube type of module for an HP, IP or LP economizer. This type of economizer has highpoints at the top of numerous hairpin tube bends that are practically impossible to vent. The design intent is for economizer inlet feedwater to pass from upper inlet header down a row of 12 straight tubes to a lower return header. 12 inverted hairpin tubes, each pitched between a pair of inlet row tubes on the lower return header, pass the feedwater to the next lower return header. The feedwater passes through a series of hairpin tubes to the final lower return header. Finally, 12 straight outlet row tubes convey the feedwater up to an upper economizer outlet header. The rise in feedwater temperature through each hairpin loop is typically between 10 to 20˚F. Velocities and pressure drop through each hairpin loop are usually small, so that if one or more hairpin tubes in a row remains air-locked or vapor-locked, then flow increases in the unblocked tubes and pressure drop rises by a very small amount. Provided that the flowing tubes can transfer sufficient heat for the HRSG to produce its rated evaporation, there is no means of detecting that some hairpin tubes are obstructed by an air-lock or vapor-lock and only partially filled with stagnant feedwater, except by installing thermocouples on hundreds of individual tubes.

Figure 8 provides temperatures measured at the outlet of seven of the twelve outlet- row tubes of one of three identical modules, which comprise the final HP economizer section of a large HRSG. The temperatures of those tubes, as measured by thermocouple, remain within a spread of 20˚F and range from about 10 to 30˚F below saturation temperature. There is nothing in Figure 8 to indicate whether or not there is steaming in the earlier tube rows of this economizer. However, temperatures measured below the hairpin bend but above the roof seal on seven of twelve final row hairpin tubes, Fig. 9, indicate that during a typical startup and throughout several hours of operation at baseload and 85% load, four of the seven tubes fitted with a thermocouple remain 15 to 45˚F above the saturation temperature of feedwater in the tubes and close to the gas inlet temperature to the section, indicating that the upper parts of these tubes are dry, and whatever water is in the lower parts of these tubes is stagnant. The other three tubes with thermocouples, and probably the majority of the hairpin tubes in the same row without thermocouples, operate with feedwater temperature about 25 to 45˚F below saturation temperature. In the first hairpin tube row following the economizer inlet tubes one tube is vapor-locked or air-locked continuously and the adjacent hairpin tube is also obstructed by vapor for several hours, Fig. 10. The hairpin tubes with stagnated flow are those closest to the end of headers, which have a smaller pressure drop than those closer to the inlet and outlet branches, and are also most affected by leakage of any hot gas along duct sidewall and in the gap between adjacent modules. The stagnation of feed flow through the hairpin tubes at each end of the header also caused flow to stagnate or recirculate upwards in the end tubes of the inlet row of the economizer, Fig 11. Four of seven inlet row tubes installed with a thermocouple at tube inlet record temperatures of the order of 100˚F above the temperature of the incoming feedwater.

The phenomenon, of large differences in temperatures of tubes in the same row attributed to air- or vapor-locks obstructing flow in some hairpin tubes, occurred consistently during many startups over several months. The phenomenon was equally evident and more pronounced in IP and LP economizers. For example, during the same event recorded for the final HP economizer in Figures 8 to 11, four of seven LP economizer outlet row tubes fitted with a thermocouple at their outlet end operated consistently between 35 and 70˚F above saturation temperature and close to the gas temperature at the inlet to the LP economizer, Fig 12. Furthermore, after feedwater flow commenced the temperature of some tubes fluctuated numerous times during a single startup of the CT as they cleared and re-stagnated in synchronism with sharp changes in feedwater flowrate probably caused by the continually fluctuating system resistance as vapor obstructions were cleared from tubes and then returned, Fig 13.

The consequences of air- or vapor-locking and flow stagnation or reversal in tubes of hairpin-type economizers raises several concerns regarding their impact on durability of the economizers, especially for HRSGs intended for some periods of cycling service. These include:

- Potentially significant thermally-induced stresses in inlet and outlet row tubes at tube attachments to upper and lower headers, which can accumulate many cycles per startup.

- Potentially significant thermal stresses in the upper bend and at header attachments of hairpin tubes caused by intermittent chilling of tube inlet legs each time flow re-establishes, in conjunction with fluctuating transfer of gravity loads between tubes as hotter tubes expand upwards and unload at the upper support bars.

- Unstable feedwater control influenced by fluctuations in system resistance as some tubes intermittently clear and then re-stagnate.

- Increased risk of flow assisted corrosion at tube bends, which intermittently vapor-lock and then re-establish flow.

- Deposition of dosing chemicals on tube surfaces during steaming leading to obstruction of flow.

- Potential corrosive environment in tubes in proximity to the boundary between water and steam phases.

Shared Headers for Different Economizer Sections

Many HRSGs were installed with two economizer sections, e.g., HP and LP, or HP and IP, in the same panels and sharing the same upper and lower headers with a pressure plate inside each header to separate HP from LP or IP feedwater. During every startup drum swell usually necessitates the dumping of water from all steam drums to avoid high drum levels. During the initial part of startup, there is no feedwater flow through any economizers and the stagnant water-filled tubes heat up to gas temperature, which is significantly hotter at the low temperature economizers when there is no feedwater flow than under normal operation on load. Eventually HP drum swell subsides and feedwater flow then commences through the HP economizer sections rapidly cooling the tubes in the HP economizer sections to the feedwater temperatures. In Figure 14, HP feedwater flow commenced about 30 minutes after CT startup. Commencement of LP feedwater flow had to be delayed until about 73 minutes after CT startup, by which time the LP economizer tubes had heated to 375˚F. Meanwhile, the adjacent HP economizer tubes were cooled by HP feedwater flow to about 250˚F, i.e., about 125˚F lower than the LP economizer tubes attached to the same upper and lower headers. Temperature differences between HP and LP economizer inlet panel tubes in excess of 150˚F have been observed on some starts when the start of LP feedwater flow was further delayed. This phenomenon develops high thermally-induced forces of opposite sign in the tubes of the two sections sharing the same upper and lower headers, which have caused many tube failures that typically begin to occur after about 200 startups. Tube failures are most prevalent at tubes closer to the adjacent economizer section and in tubes that are offset from the vertical center-line of the header, which develop a bending moment that intensifies the thermal stress at the point of attachment of tube to header.

To alleviate this problem, some units have installed pressure-part modifications that physically separate the lower headers so that tubes of each section can expand en-bloc independently of the other section, Fig 14.

Although physical separation of lower headers overcomes one design weakness, it does not address a related thermally-induced fatigue failure mechanism which affects many, but not all, HRSGs installed with economizers assembled from a series of panels arranged with two or more serpentine feedflow passes, Fig.16. The feedwater inlet panel of serpentine-type preheaters or of LP, IP and HP economizers that are supplied with cold feedwater water direct from the condenser without significant preheat are subjected to transiently large thermally-induced stresses in the tubes of the first two passes each time during startups that feedwater flow is introduced into stagnant economizer tubes heated to gas temperature. For example, the inlet panel of a small LP economizer section illustrated in Figure 15 comprises a downflowing inlet pass of two tubes and an upflowing return pass of three tubes. All five tubes are attached to a lower return header and to a common upper header with an internal division plate to separate the inlet from outlet feedwater. During the typical startup recorded in Figure 15, LP feedwater was not required until about 73 minutes after CT startup, by when the stagnant inlet and outlet pass tubes had all heated up to about 375˚F. The initial flow of cold feedwater took of the order of one minute to flow down to the bottom of the inlet pass tubes displacing the hot water into the outlet pass tubes. When cold water reached the return header, transiently the inlet pass tubes were chilled to about 125˚F while the outlet pass tubes were still at 375˚F. Thus for a brief period, (which is all that is needed to develop the tube forces and inelastic thermal strains that cause localized plasticity and fatigue damage), the temperature difference between the two passes was about 250˚F. This temperature difference is higher than those that have caused tube failures in economizer panels with shared upper and lower headers. On other units supplied with colder feedwater from the condenser, and when the commencement of LP feedwater flow is delayed and stagnant tubes heated above 400˚F, the brief transient difference in average temperature of inlet and second pass tubes may exceed 300˚F, which will develop very high thermal stresses at tube attachments to headers, especially in tubes that are offset from the vertical center-line of the header.

On installed HRSGs, which have serpentine-type economizer panels, it is usually impractical to modify the design of the economizer to eliminate this potential source of premature fatigue failures. The problem can be mitigated if some means can be devised to preheat the incoming feedwater, or re-circulate feedwater through the economizers during the startup period when feedwater cannot be fed to the steam drum, or by operating procedures that permit earlier feeding of water to the drums, but these options involve significant modifications for many installed units.

For new installations the weakness can be mitigated by locating the inlet to the first pass of lowest temperature economizers to the lower header, which permits the physical separation of the inlet header from the lower header of tubes in subsequent passes in the same panel, Fig. 17. This eliminates the most severe transient thermally induced tensile forces in tubes of the first pass by allowing them to contract when chilled as cold feedwater is introduced. Transient temperature differences still occur between tubes of subsequent passes after feedwater flow is initiated. However by the time the feedwater flows down the second pass its temperature has already increased by of the order of 75 to 100˚F, thereby reducing the transient difference between tubes of second and third passes to of the order of 150 to 200˚F. This temperature difference would probably still be large enough to develop thermal fatigue stresses in tubes of second and third passes large enough to cause early fatigue failures, especially in tubes offset from the header vertical center-line. Therefore, for units intended for cycling service the provision of a separate deaerator capable of supplying deaerated feedwater at about 250˚F considerably reduces the risk of thermal fatigue tube failures in the lower temperature sections of economizers and preheaters, as well as reducing the risk of pitting corrosion in economizer tubes by stagnant, poorly-deaerated water introduced during feed water flow interruptions at startups.

It should be noted that the hairpin tube type of economizer or preheater is also vulnerable to early fatigue failure in inlet row tubes and in the hairpin bends closer to the inlet portion of the module attributable to a similar transient chilling mechanism in inlet row and subsequent rows of down-flowing tubes when cold feedwater is first admitted during reloading of the CT after startup. The transient differential contraction of the colder downflowing tubes lifts the hotter downstream rows of hairpin tube bends off their supports, which transiently transfers a large portion of the weight of the module to inlet row tubes and to the earlier rows of hairpin bends.

Recirculation in Economizer Tubes

Temperature measurements at the top and bottom of tubes in the first downflow pass of serpentine-type economizers have highlighted that buoyancy effects cause some tubes to reverse flow direction and recirculate feedwater from the lower return header back to the upper inlet header. Figure 18 highlights that immediately after HP feedwater flow commenced 32 minutes after CT startup the tube at the end of the header adjacent to the duct sidewall started to recirculate heated feedwater back to the inlet header. The same behavior is evident to a slightly lesser extent in the adjacent tube and, to a progressively reducing extent, also in the third tube from the header end for about 20 minutes after HP feedflow commenced. The feedwater recirculation phenomenon disappeared after feedwater flowrate increased above 50% of baseload flow provided also that rapid fluctuations in HP feedwater flow rate remained below +/-25%. The phenomenon occurs in tubes which have lower pressure drop between tube inlet and outlet, being furthest from the second pass in the lower header and also from the inlet branch on the upper header. Thus flow rate through these tubes is lower, and temperature rise and upward buoyancy forces higher. It is suspected that an air pocket may remain trapped at the blind end of the header and increases the system resistance from inlet branch to end tubes. Tubes adjacent to the duct sidewall may also pick up more heat from leakage of hot gas between tubes and duct wall.

In this case, the difference in average temperature of recirculating tubes and tubes flowing in the intended downward direction is about 65˚F, being half the temperature difference measured at the top of the tubes. Although this is lower than tube temperature differences developed between economizer tubes by other undesirable design features, there are nevertheless concerns that the large number of thermal cycles experienced during each startup may lead to thermal fatigue failures in the affected tubes when the operating regime for the unit changes from baseload to cycling service.

Practical options to attempt to eliminate this design deficiency on installed units are limited. Improved baffles to reduce leakage of hot gas between tubes and duct sidewall and the installation of small vents on all upper headers of economizers may mitigate the problem. For new installations more attention must be given to ensuring that pressure drops through all downflowing tubes have adequate margin to be certain they overcome the upward buoyancy forces after account is taken of pressure gradients along upper and lower headers.

Improved Venting for Economizers

For new installations, most of the design weaknesses highlighted in this paper can be mitigated or entirely eliminated by relatively minor modifications, which do not require changes to established design concepts.

The new, major area of doubt highlighted by some of the tests outlined in this paper concerns the suitability of economizers with un-vented high points for large HRSGs with long vertical tubes, especially for installations anticipating periods of cycling operation. Tube temperature measurements on large HGP HRSGs employing different design concepts for economizers have highlighted that the presence of un-vented high points in the feedwater flow path through the economizers can cause obstructions by air or vapor pockets that disrupt the intended distribution of feedwater through parallel-path tube circuits, thereby developing potentially harmful thermally-induced stresses and an adverse chemical environment in the critically- affected parts of economizer tubes. There are clearly a variety of opinions on the importance of economizer venting. Many existing larger HRSGs, whether they employ economizers that have all tubes connected to upper and lower headers, or whether they employ hairpin tubes, have no operational vents on any headers. Other existing HRSGs have additional operational vents from upper headers preceding the outlet header of each economizer section. Prohibition of unvented high points in economizer tubes and interconnecting pipes would mandate a change to the established design concepts used by some OEMs, and is thus controversial. However, based on the evidence of extensive steaming found in economizers with numerous un-vented tubes, there is a strong case for adopting the conventional utility steam generator practice of venting all high points in economizers, simply to be absolutely certain that all air and vapor pockets at high points will quickly clear. For those who believe that venting of all high points in economizers is unnecessarily restrictive, prudence would indicate the need to perform the investigations, tests and analysis necessary to either prove beyond doubt that all high points completely clear, or quantify the significance of measured tube temperature anomalies and other adverse consequences of air or vapor pockets on the cyclic life of economizer tubes.

A common misconception was that steaming is only a short term problem during startups in the final row of economizer tubes and these were arranged to flow upwards and vent up to the steam drum. Since many units without vented economizers have operated without insuperable problems, this appears to have encouraged a presumption that, either un-vented economizer tubes do successfully blow out all air and vapor collected in high points during every startup and that there are then no obstructions by air or vapor pockets in the economizers during normal operation, or alternatively the presence of some air-locked or vapor-locked tubes was perceived to have no impact on the durability of economizer tubes. The tests reported in Figs, 8 to 13 have highlighted extensive air or vapor locking in hairpin tubed economizers that have caused operating conditions in the economizer tubes that raise major concerns for the durability of these tubes, especially when the HRSG is subjected to cycling service. Until thermocouples were attached to hundreds of economizer hairpin tubes there was no obvious indication of obstructions to flow in any tubes during operation of that HRSG. Thus it is possible that similar problems may be found in other economizers employing similar tube configurations if subjected to similar testing.

For recent HRSG designs, it has become the standard practice of some OEMs who traditionally supply economizers with all tubes connected to an upper and lower header to now provide operational vents from every high point on upper headers and interconnecting pipes in all economizers. This unilateral decision by some OEMs to incur the additional cost for adding many vents acknowledges the substantial benefits of being absolutely certain that air and vapor is quickly removed from all high points in order to eliminate the possibility of potentially harmful consequences of local pockets of trapped air or vapor.

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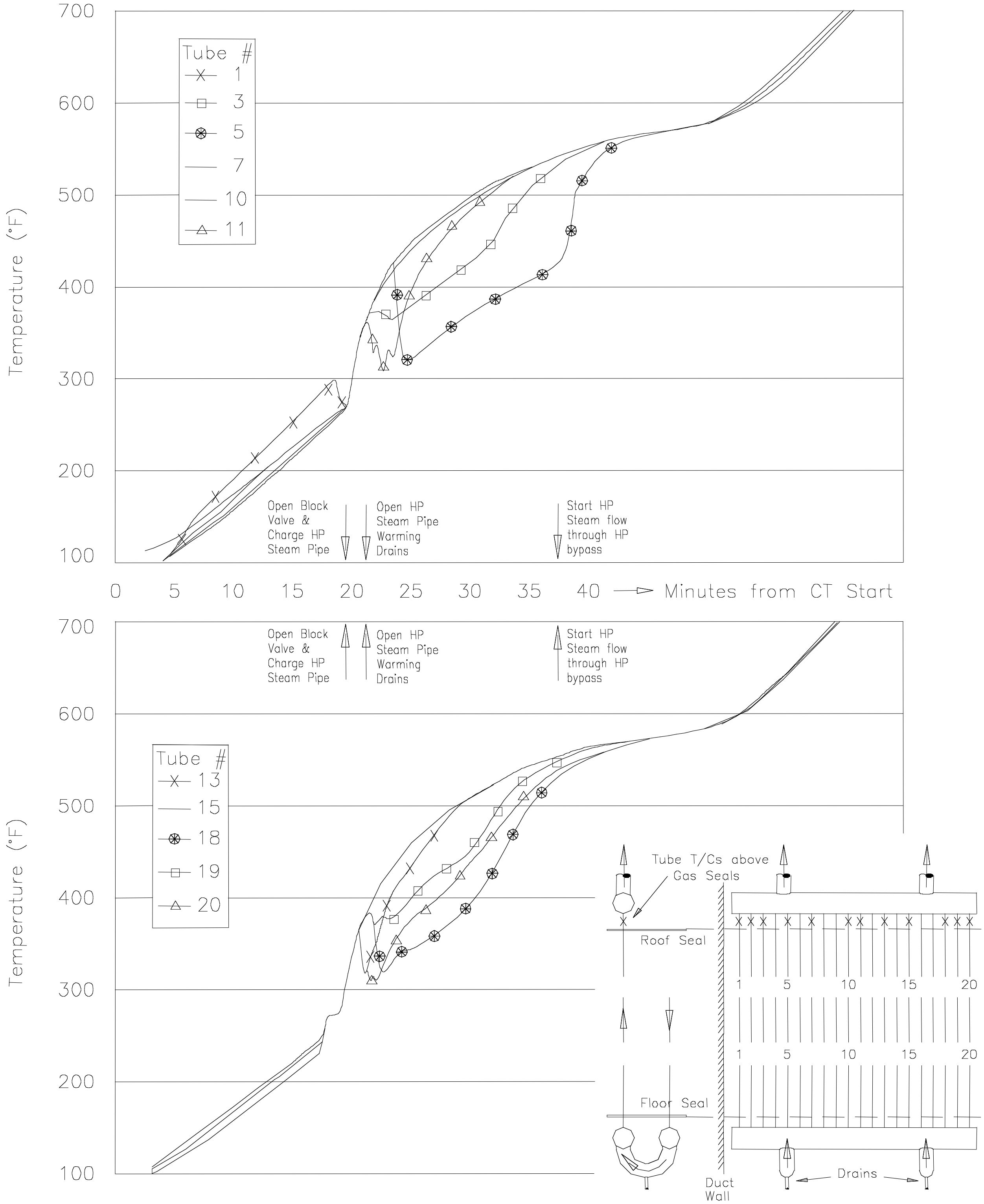

Fig. 1 – HP Superheater Outlet Tube Row Arrangement and Tube Outlet Temperature at Cold Start

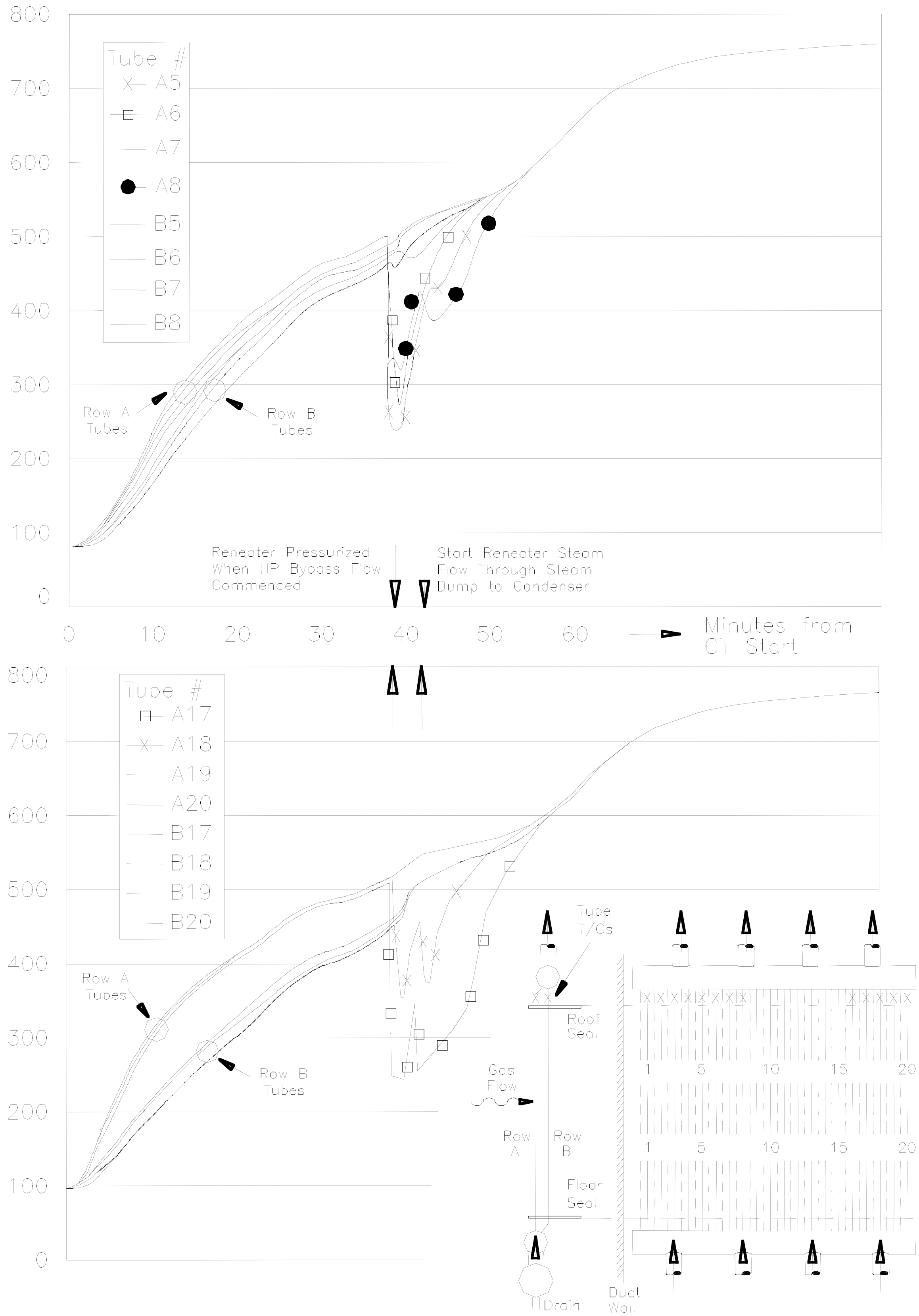

Fig. 2 – Reheater Outlet Tube Rows Arrangement and Tube Outlet Temperature at Cold Start

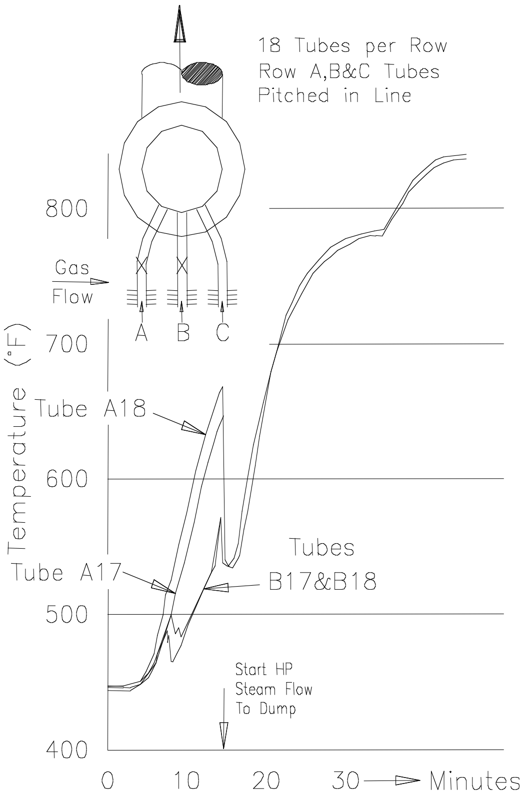

Fig. 3 – Tube Temperature at Outlet from Final HP SH Panel – Hot Start

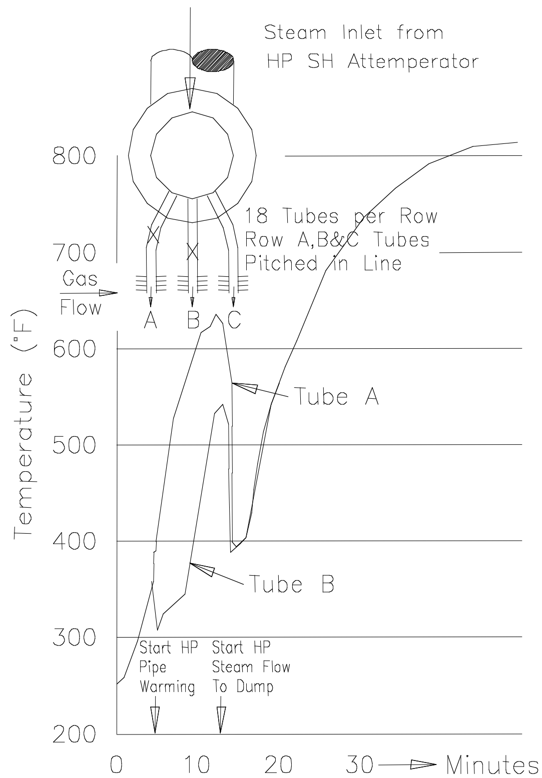

Fig. 4 – Tube Temperatures at Inlet to Second Stage HP SH – Cool Start

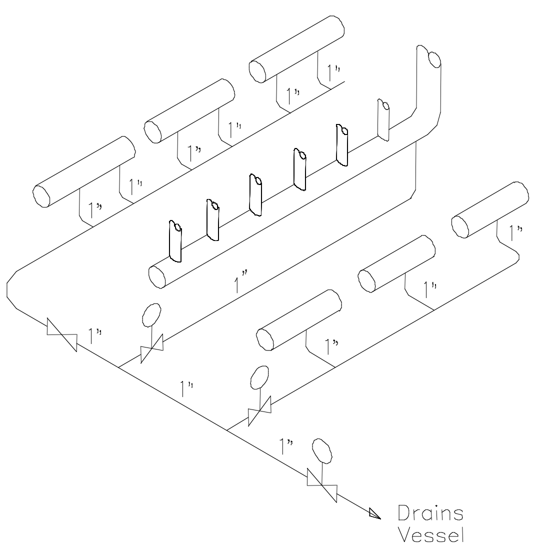

Fig. 5 – INCORRECT Interconnection of Superheater Drains to Disposal

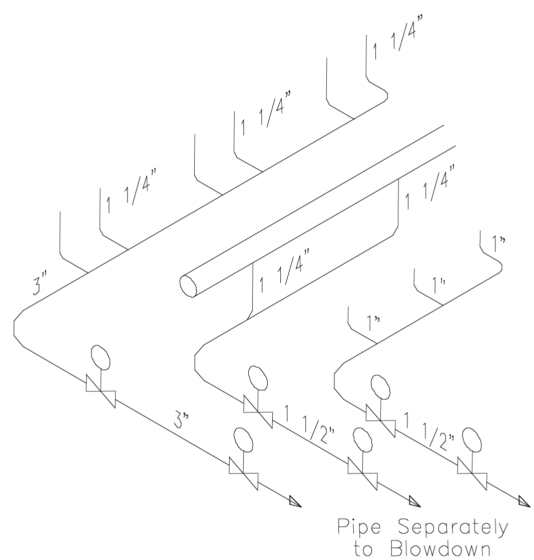

Fig. 6 – CORRECT Drains Disposal Arrangement for Superheater

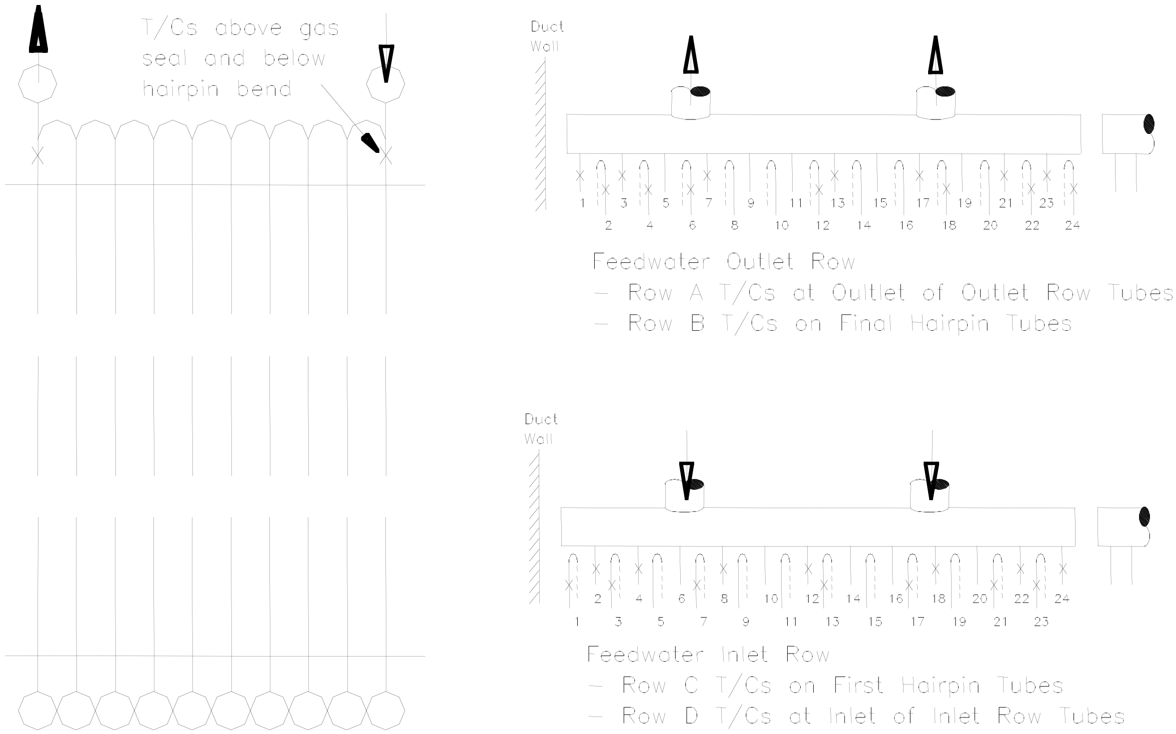

Fig. 7 – Hairpin Tube Economizer Module

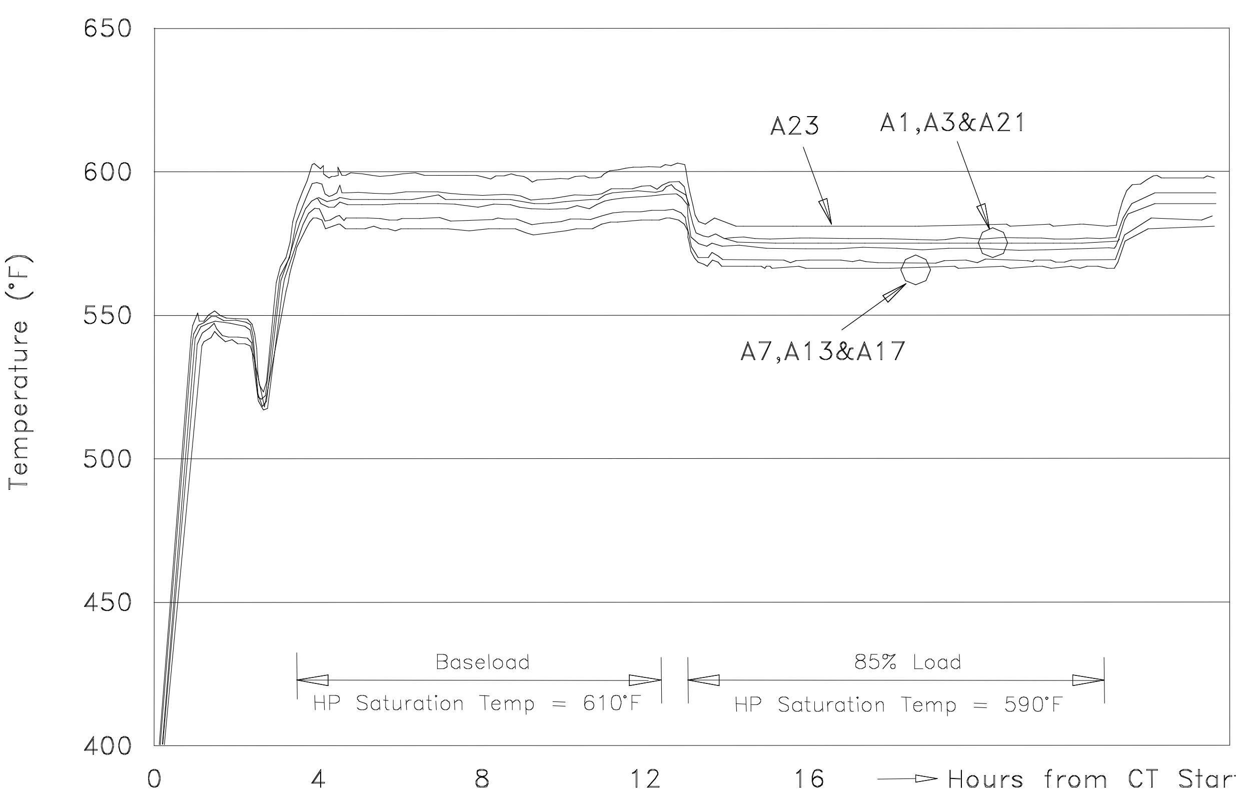

Fig. 8 – HP Final Economizer Module – Tubes Before Outlet Header

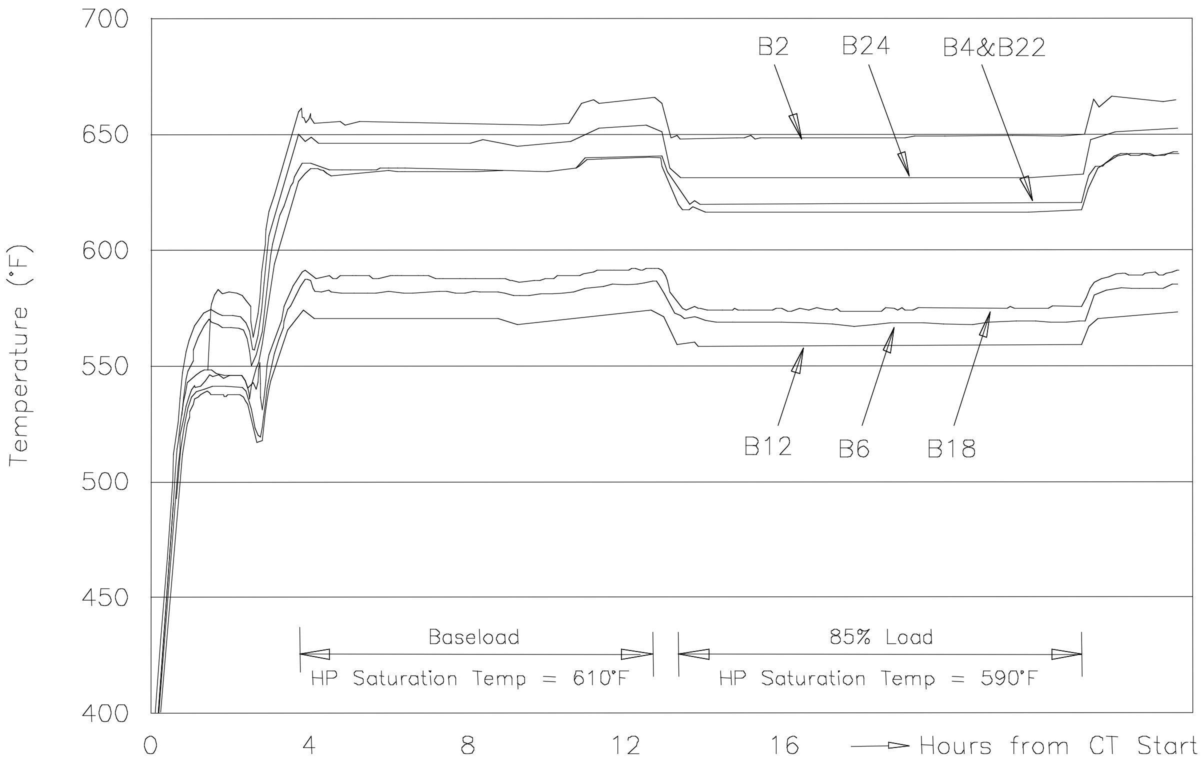

Fig. 9 – HP Final Economizer Module – Final Hairpin Tubes Below Bend

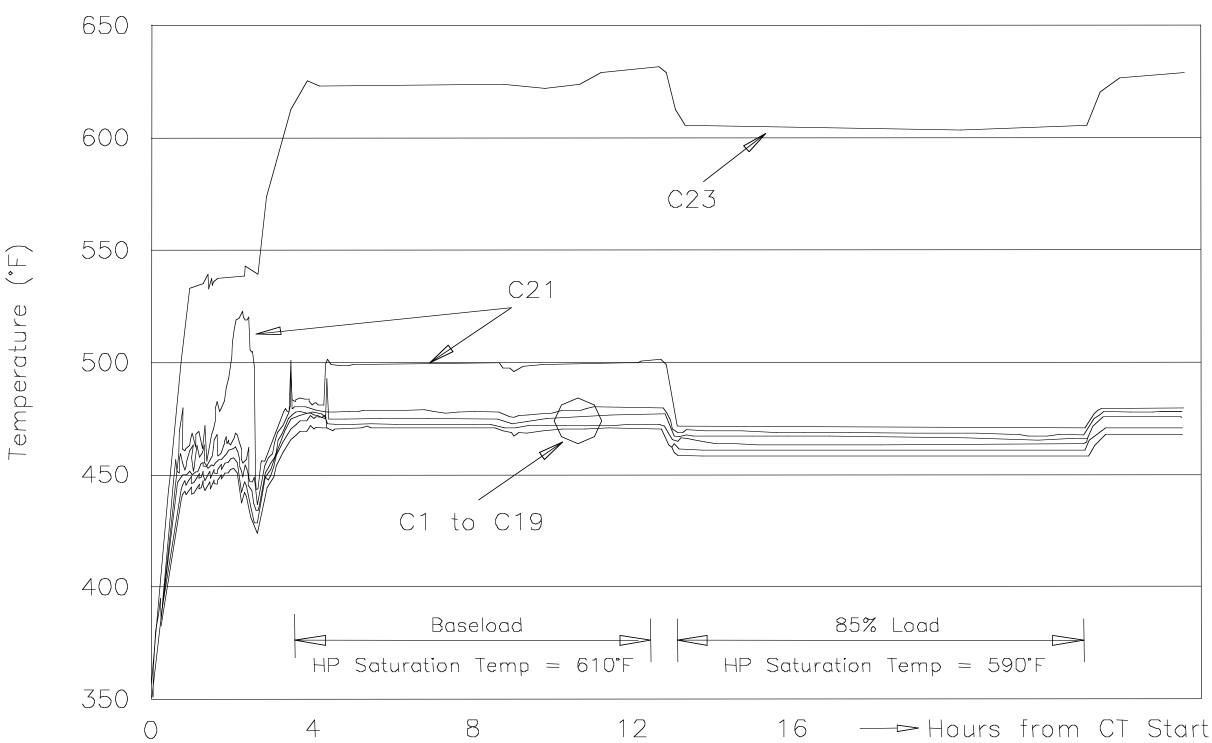

Fig. 10 – HP Final Economizer Module – First Hairpin Tube Below Bend

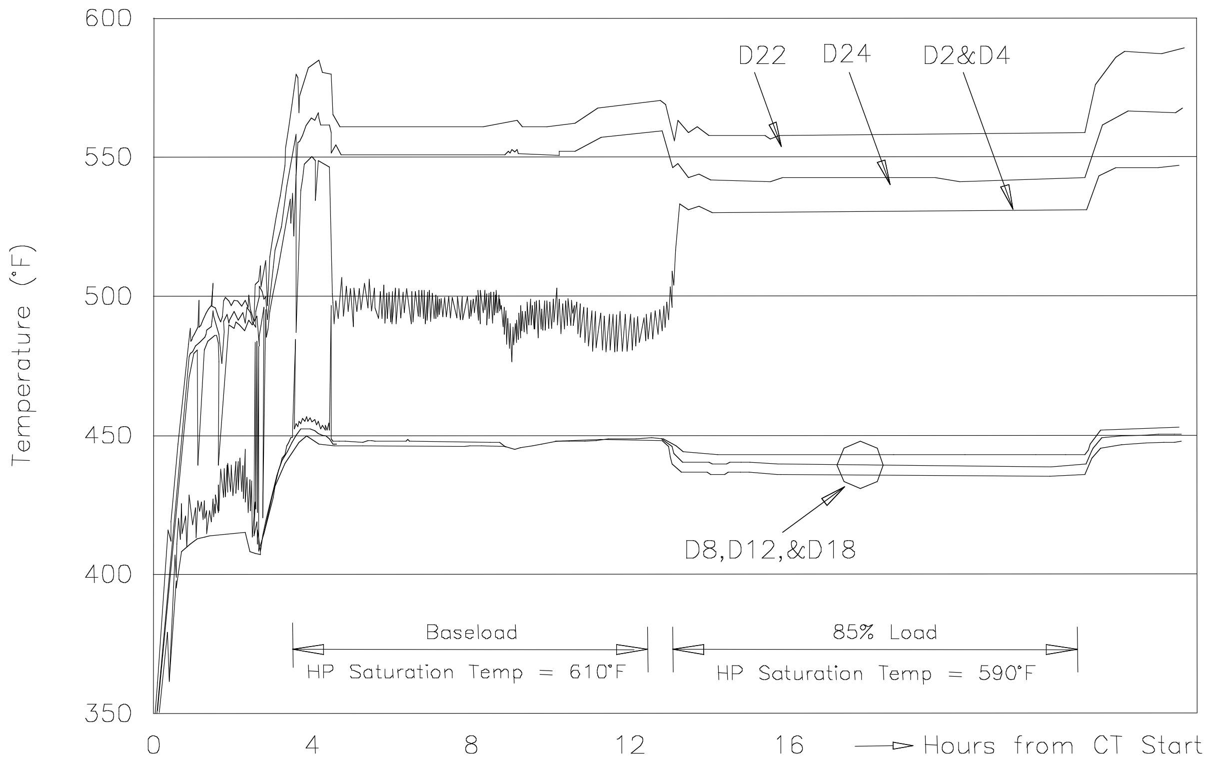

Fig. 11 – HP Final Economizer Module – Inlet to First Tube Row

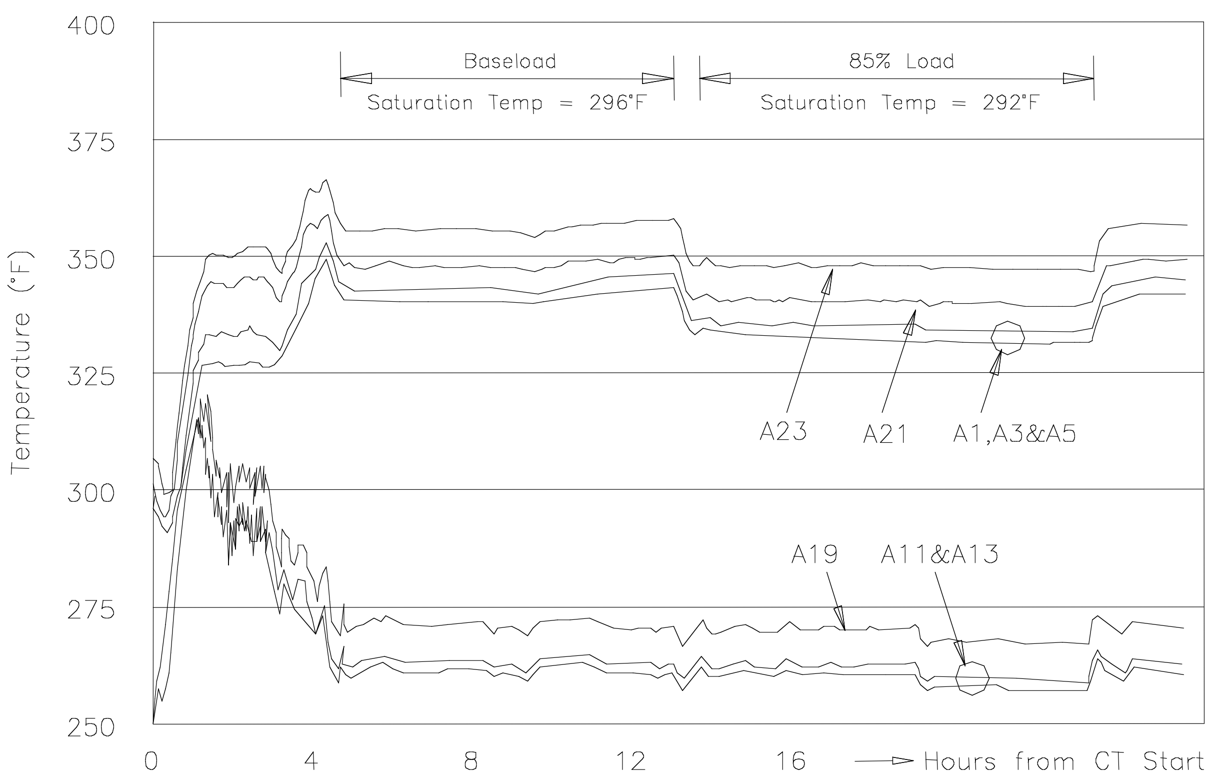

Fig. 12 – LP Economizer Module – Tubes Before Outlet to Outlet Header

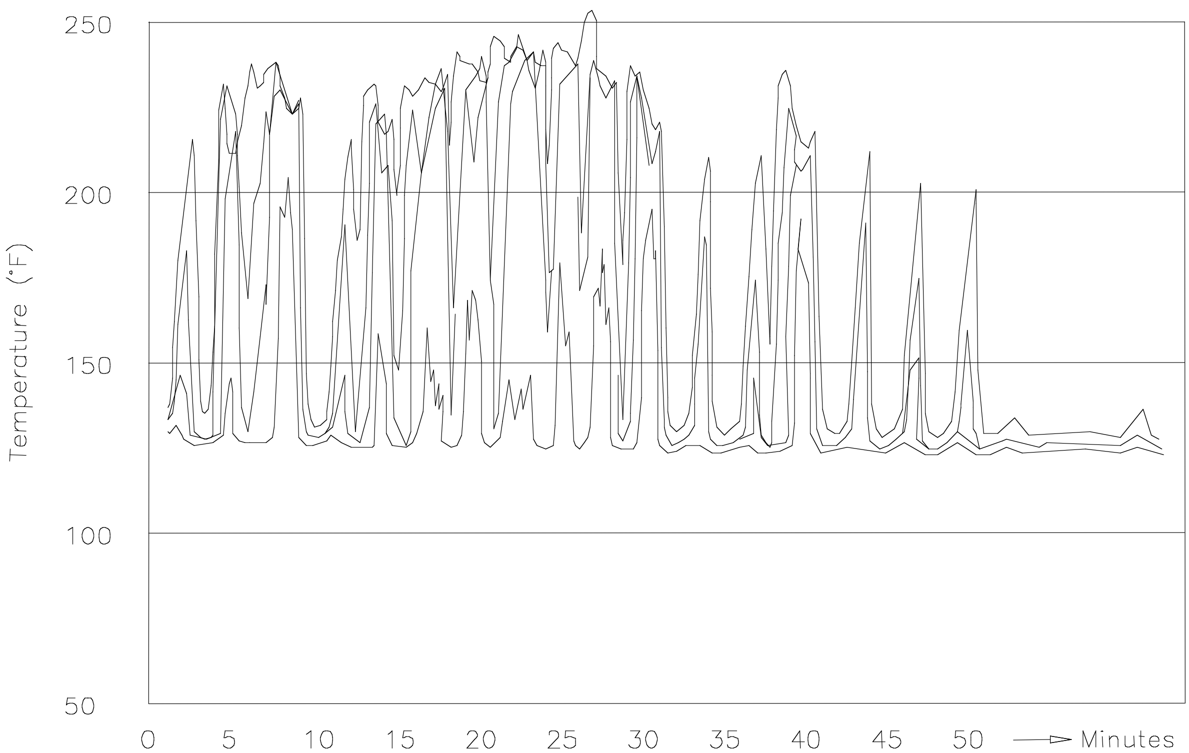

Fig. 13 – LP Economizer Module – Inlet to Inlet Row Tubes

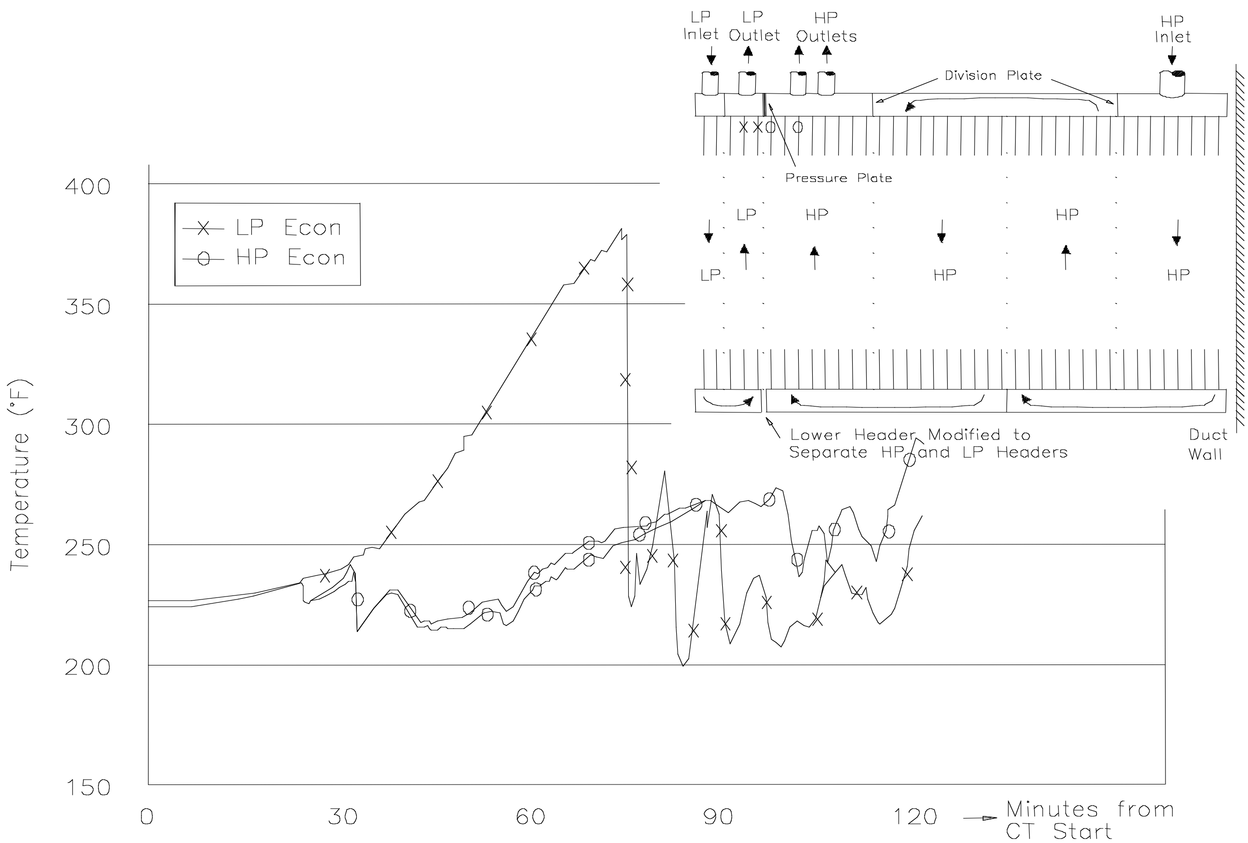

Fig. 14 – Startup for LP/HP Economizer Feedwater Inlet Panel Installed with Shared Top and Bottom Headers

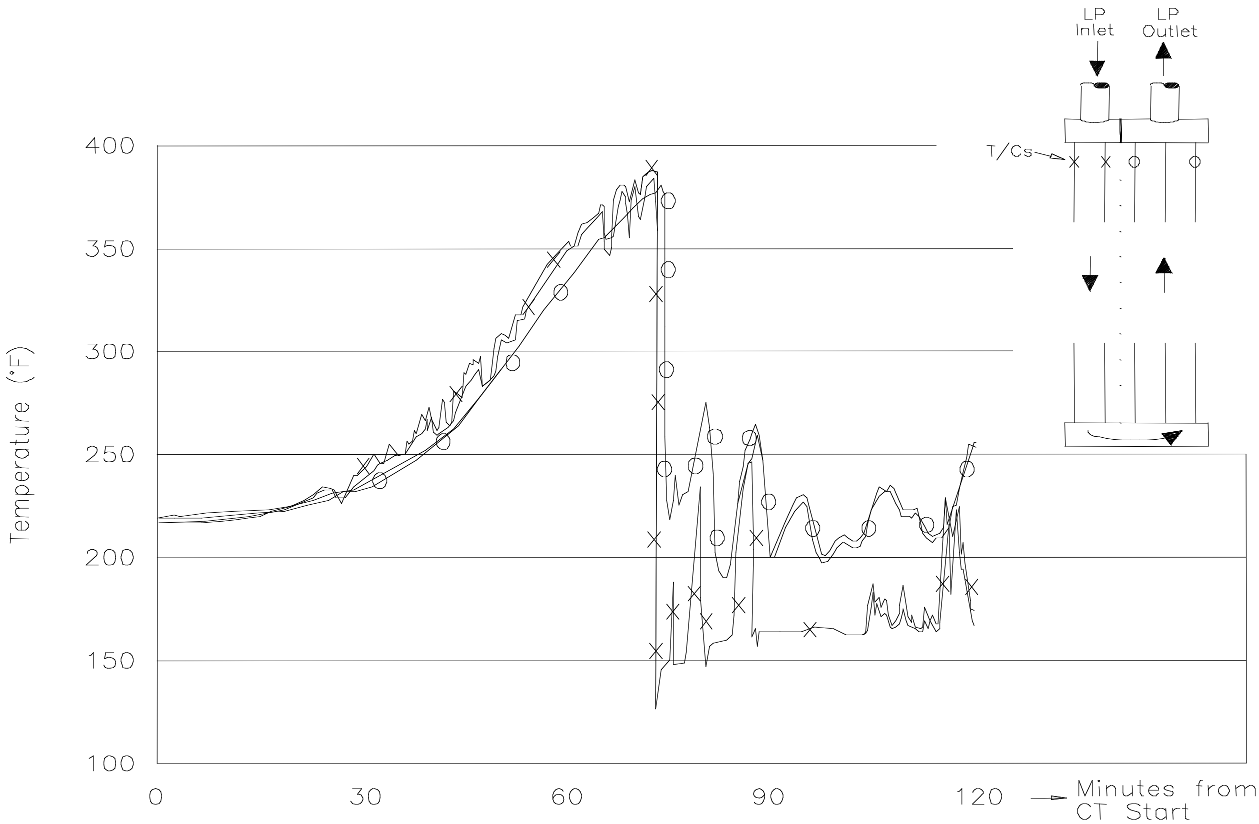

Fig. 15 – Two Pass LP Economizer Panel with Division Plate in Upper Header

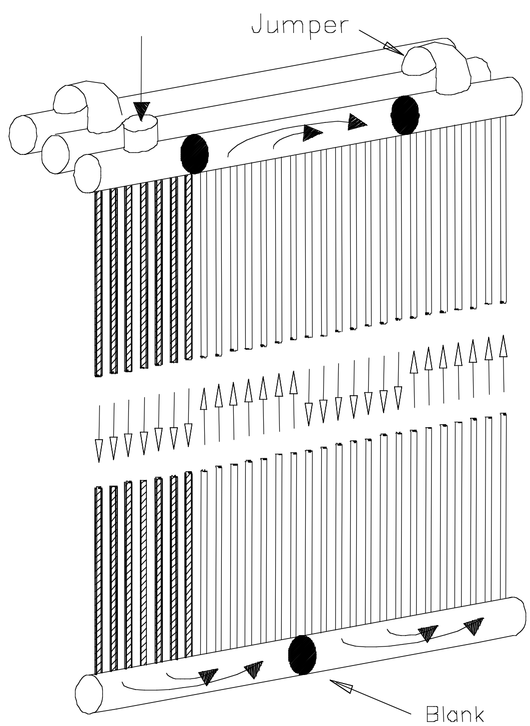

Fig. 16 – Typical Multipass Low Temp Economizer/Preheater Inlet Panel

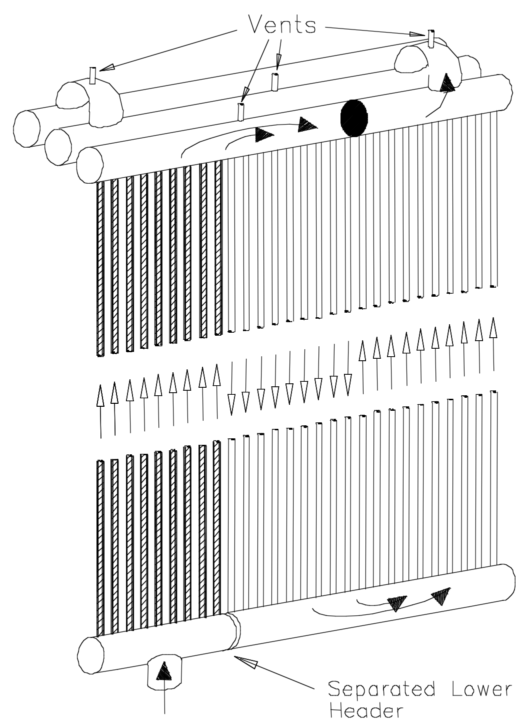

Fig. 17 – Design of Low Temperature Econ/ Preheater Inlet Panel for Cycling

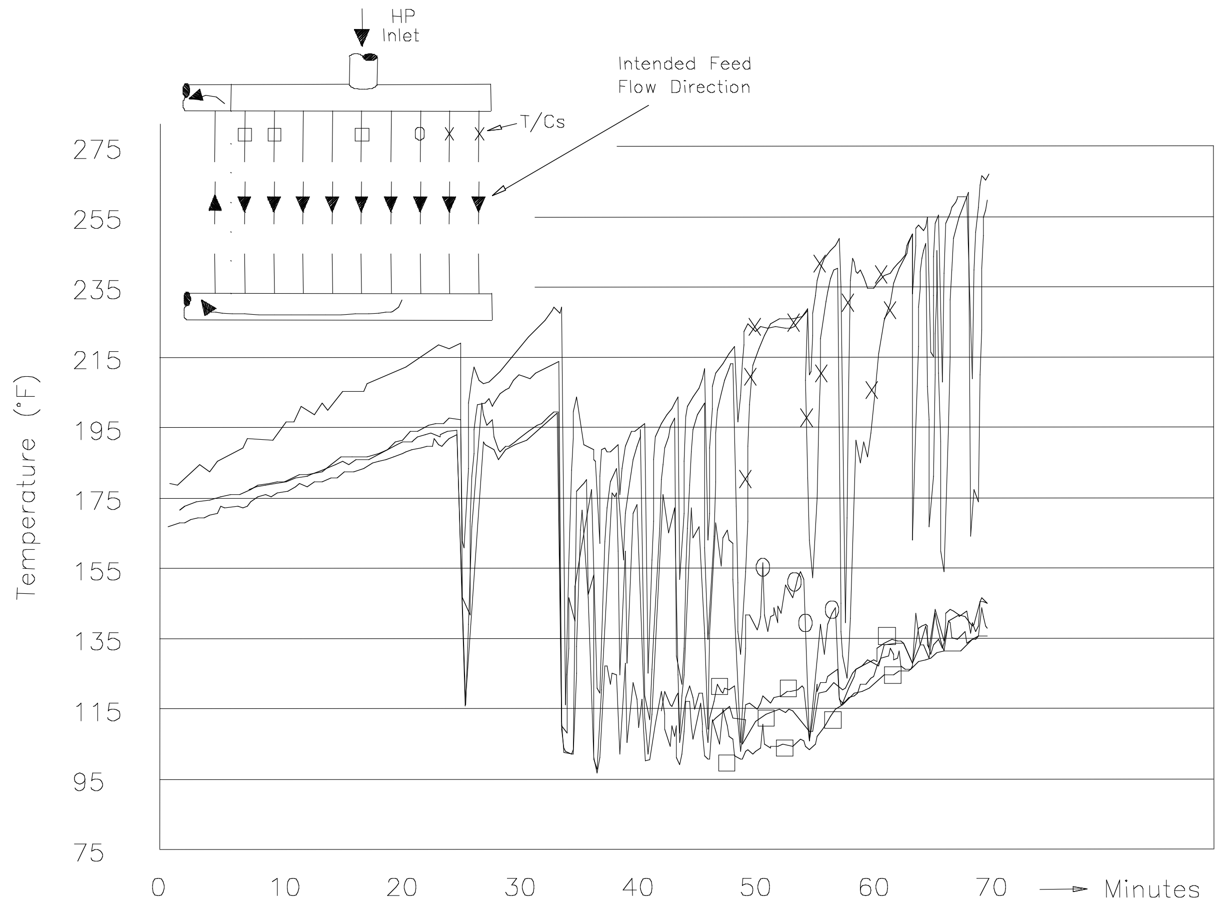

Fig. 18 – Inlet Pass Tubes of HP Economizer with Flow Recirculation in Tubes