PREHEATING OF INCINERATOR COMBUSTION AIR UTILIZING LOW PRESSURE STEAM AND HEAT PIPES

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ABSTRACT

Sour gas plant operations require tail gas incineration in order to meet strict environmental compliance. Typically, tail gas is supplemented with fuel gas for proper combustion and thus reduction of H_2S. To minimize fuel gas consumption, combustion air is normally preheated via a standard steam coil. However, in colder climates conventional steam coils are faced with the potential of freeze damage. During periods of extreme cold, low steam flow, or unit shut down, air along with condensate can collect, allowing the condensate to freeze and rupture tubes. Steam coil manufacturers have attempted to alleviate the problem by instituting strict and somewhat complicated operating guidelines that may or may not prevent freeze damage.

Hudson Products has developed an advanced freeze-resistant air preheater/steam condenser. This unique design offers gas plants a high performance freeze-resistant combination of combustion air preheat and steam condensate collection. This is accomplished by utilizing the thermosyphon capability of heat pipes to configure the combustion air preheater/steam condenser combination. A thermosyphon tube bundle condenses steam on the outside tube surfaces instead of inside tubes as is done in conventional designs. While for certain operating condition's condensate can still freeze and thaw on the outside of the thermosyphon tubes, it will not result in any damage to the unit. Additionally, thermosyphons are self regulating. This allows plant operators to turn on the fan, open the steam valve and walk away without having to set or monitor a specialized freeze prevention system. Heat from the condensed steam is rejected to the combustion air via the finned thermosyphon bundle and fan. This results in increased combustion efficiency (fuel gas savings) and plant steam condensing capacity. Additionally, during incinerator trips, the thermal inertia or capacitance inherent in the unit maintains incinerator stack temperature for longer periods of time. Thus, plant operators have an additional time to bring the incinerator back on line before a low temperature emission excursion can occur.

Following an extensive laboratory development program consisting of thermosyphon performance, bundle icing, and long-term materials compatibility testing, a prototype unit was installed in the Shell Canada Limited Jumping Pound Complex gas processing plant located west of Calgary, Alberta, Canada. Testing was performed over a seven-day period in March 1998 to evaluate the performance of the unit under steady-state and transient operating conditions. Overall, actual unit performance exceeded the predicted performance by as much as 10% under steady-state conditions. Transient testing showed that the unit is thermally very responsive and mechanically impervious to severe upsets. Control room data before and after start up indicates an average yearly reduction in incinerator fuel gas consumption of nearly 18%.

INTRODUCTION

Sour gas plant operations require tail gas incineration in order to meet strict environmental compliance. Typically, tail gas is supplemented with fuel gas for proper combustion and thus reduction of H_2S. To minimize fuel gas consumption, combustion air is normally preheated via a standard steam coil. However, in colder climates conventional steam coils are faced with the potential of freeze damage. During periods of extreme cold, low steam flow or unit shut down, air along with condensate can collect, allowing the condensate to freeze and rupture tubes.

Shell Canada Limited contacted Hudson Products in early 1997 to design a heat pipe based air heater to replace a traditional steam coil heating system that failed and was removed a few years earlier. The objective was to design a heat exchanger that would preheat the incinerator combustion air by condensing low pressure plant steam. The heat exchanger was to be freeze-proof, easy to operate and require minimal or no maintenance. During the same period, Hudson Products was engaged in the development of a unique freeze proof air-cooled steam condenser for use in large co-generation power plants. An extensive laboratory development program consisting of full-scale heat pipe testing, bundle freeze testing, and long-term materials compatibility testing was nearing completion and the search for a small prototype application was underway. Based on Shell Canada's requirements, a small prototype heat exchanger employing this freeze-proof technology was designed, constructed and installed in Shell's Jumping Pound gas processing plant located West of Calgary, Alberta. The prototype started operation in January 1998.

HEAT EXCHANGER DESCRIPTION

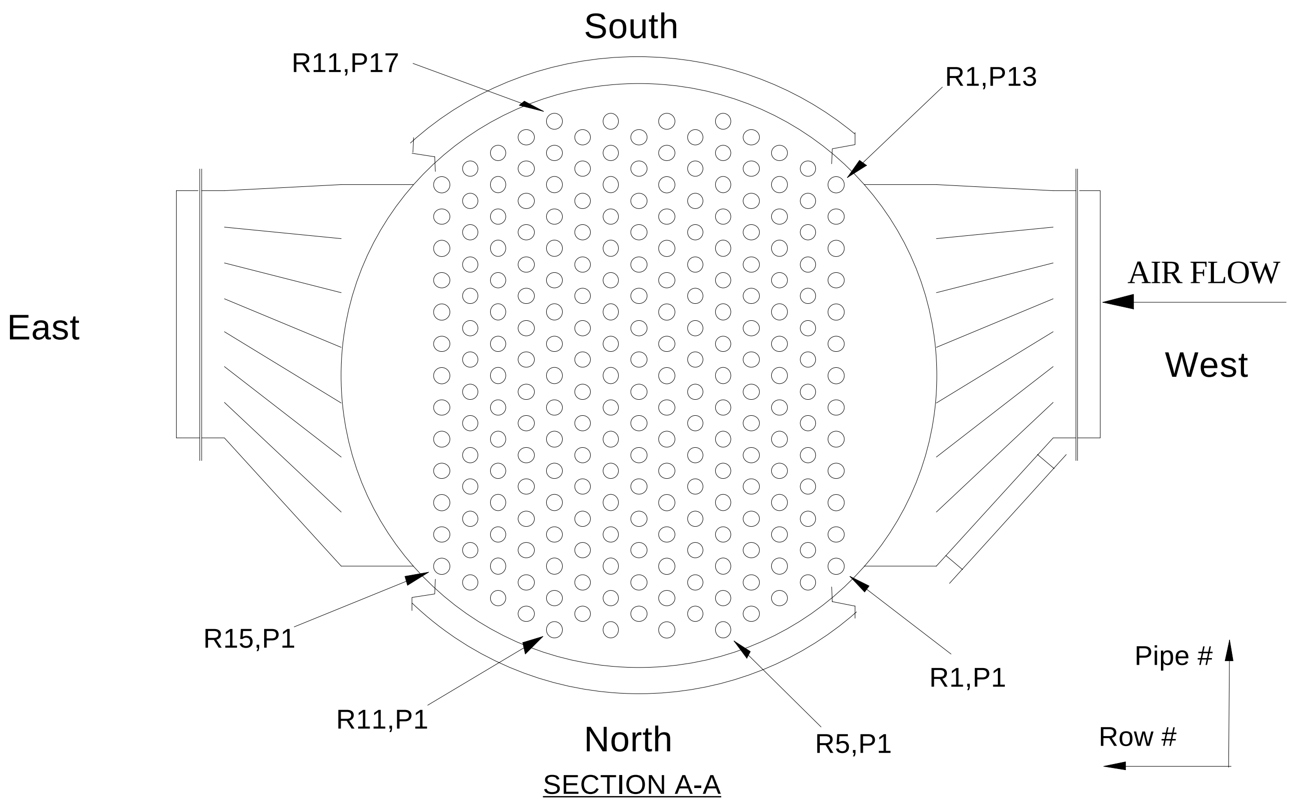
As shown in Figure 1, the heat exchanger consists of 232 vertical ammonia thermosyphons arranged in a staggered array. The pipes have 2-ft long evaporators and 4-ft long condensers. The shorter evaporator end tubing is bare for condensing low pressure steam. The condenser ends have 3/4-in high embedded aluminum fins at 11 fins per inch for enhanced convection to forced air. A 3-in. thick tubesheet separates the evaporator and condenser ends. Steam flows into the lower elliptical head through a nozzle with an erosion plate to protect the bare tubes from direct droplet impingement. The steam-side operating pressure set the design of the lower head and resulting cylindrical shape of the heat exchanger. The tubes are arranged on a staggered pitch to form a semi-circular bundle surrounded by a close-fitting shroud to prevent air bypass. Inlet and outlet air ducts have flow splitters to provide uniform inlet and outlet flow distributions. The air heater performance design data for the cold and hot days, respectively, are listed below in Table 1. Additional heat exchanger information can be found in Reference 1.

TABLE 1
AIR HEATER OPERATING PERFORMANCE

Parameter	Cold Day Design	Hot Day Design
Inlet Air Temperature	-40 ^{o}F	70 ^{o}F
Outlet Air Temperature	180 ^{o}F	232 ^{o}F
Steam Temperature	290 ^{o}F	272 ^{o}F
Heat Duty	5,570,000 Btu/Hr	4,090,000 Btu/Hr

TEST DESIGN & INSTRUMENTATION

The heat exchanger was instrumented with 80 type K thermocouples to measure air, steam, condensate, and heat pipe metal temperatures. Pitot tubes were installed in the inlet and outlet ducts to measure static and velocity pressures and bundle pressure drop. A pressure gauge was installed in the lower head to measure steam pressure in addition to an existing plant pressure gauge located upstream in the inlet steam pipe.



Shell Canada Heat Pipe Air Heater

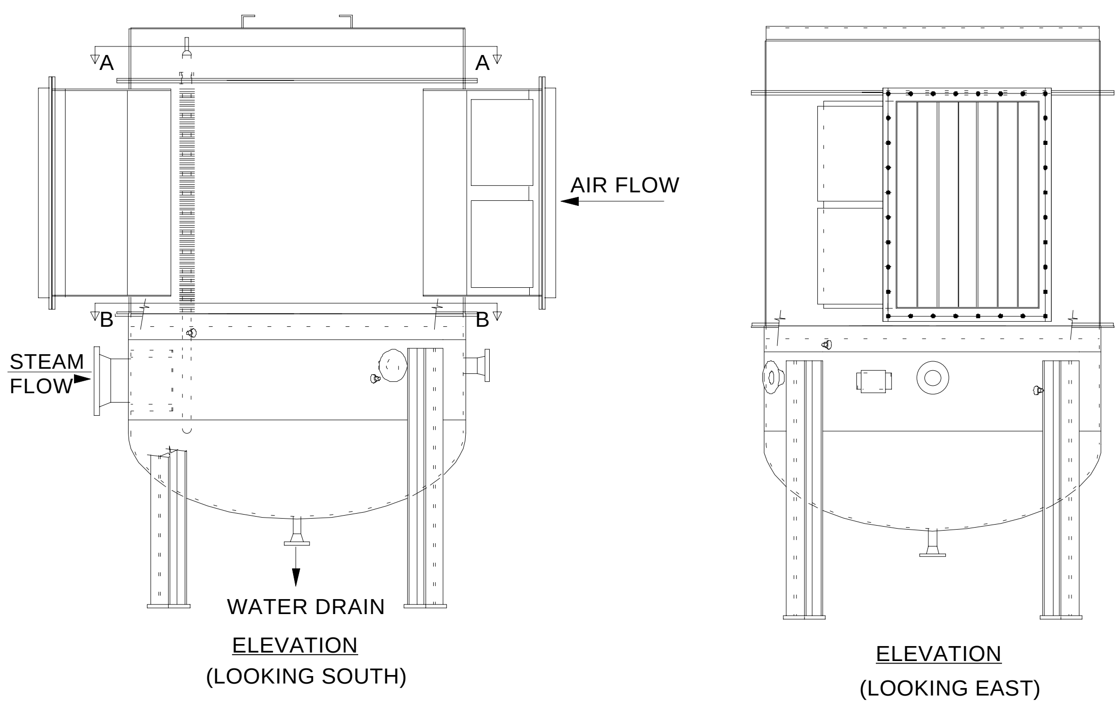


Figure 1. Air heater views.

Thermocouple data was recorded with a Fluke 2286A programmable data logger with 80 thermocouple channels. Data was recorded at about 2 channels per second and stored on floppy disk and written to paper tape. To provide some assurance that the Fluke data logger was reading accurately in the field, a certified dc voltage source was used before and after testing to confirm the Fluke's accuracy over the temperature range from 70 F to 300 F. It recorded temperatures within 0.1 F over the tested temperature range. After the field test, the data logger was returned to the Alliance Research Center where it was rechecked for accuracy in the McDermott Technology Incorporated (MTI) Standards Laboratory. The data logger had maintained its thermocouple accuracy to within 0.1 F. Other data was recorded by hand on data sheets and a test logbook stored in the project file. Hand-recorded data included manometers for air inlet and outlet static and velocity pressures and bundle pressure drop, fan speed, fan inlet air temperature, steam pressure gauges, barometric pressure, and condensate level gauge.

Obtaining an energy balance required measuring the steady flow rates on both the air and steam sides of the air heater. The air-side required pitot traverses to calibrate the outlet duct pitot tube for three flows covering the test range. The steam-side required measuring steam flow by observing condensate level changes in the air heater lower elliptical head during steady operation.

Air Flow Measurement

A single exit pitot tube was calibrated by conducting a pitot traverse at the air heater inlet through four ports installed in the top of the duct. Twenty-four total and static pressure measurements were made at this inlet plane at nominal fan speeds of 600, 1200, and 1600 rpm. This was done with the air heater isolated from the incinerator, steam off, and downstream duct hatch open. By simultaneously recording the total and static pressures from the single fixed downstream pitot tube, a flow coefficient could be calculated for that pitot tube. The flow profiles entering the heat exchanger were relatively uniform for cold isothermal operation, and the calibration coefficient was almost constant over the tested flow range. During hot operation, the inlet volumetric and mass flows are calculated with this flow coefficient and corrected for density.

$$Q_1 = K * (\Delta P_2 / \rho_2)^{½}$$

where: Q_1 = volumetric airflow at air heater inlet, ft^3/hr
K = outlet pitot tube calibration constant
ΔP_2 = velocity pressure measured by outlet pitot tube, in. water
ρ_2 = air density at outlet pitot tube, lbm/ft^3

and the mass flow is:

$$m_a = Q_1 * \rho_1$$

where: m_a = air mass flow, lbm/hr
ρ_1 = air density at air heater inlet, lbm/ft^3

Air-side heat duty was obtained from the airflow rate and temperature rise of the air.

$$Q_a = m_a\, c_p\, (T_o - T_i)$$

where: Q_a = Air-side heat duty, Btu/hr
c_p = Air specific heat (0.24 Btu/lbm-F)
T_o = Average outlet air temperature, F
T_i = Inlet air temperature, F

Steam Flow Measurement

The steam-side required measurements of timed condensate level changes during steady operation to obtain the condensate, and thus steam flow. The air heater has a 2:1 elliptical lower head with condensate level monitored by a sight gauge. Before a steady performance test, the lower head was almost drained. Then the condensate level was recorded at time intervals until the condensate filled the lower head and approached the bottom of the heat pipes. The capacity of the elliptical portion of the lower head was 35.95 ft^3 (268.9 gal). The level in the elliptical head was corrected for water density differences between the cooler level gauge and hot condensate in the head. Figure 2 shows the measured condensate level during the steady test at 600 rpm fan speed.

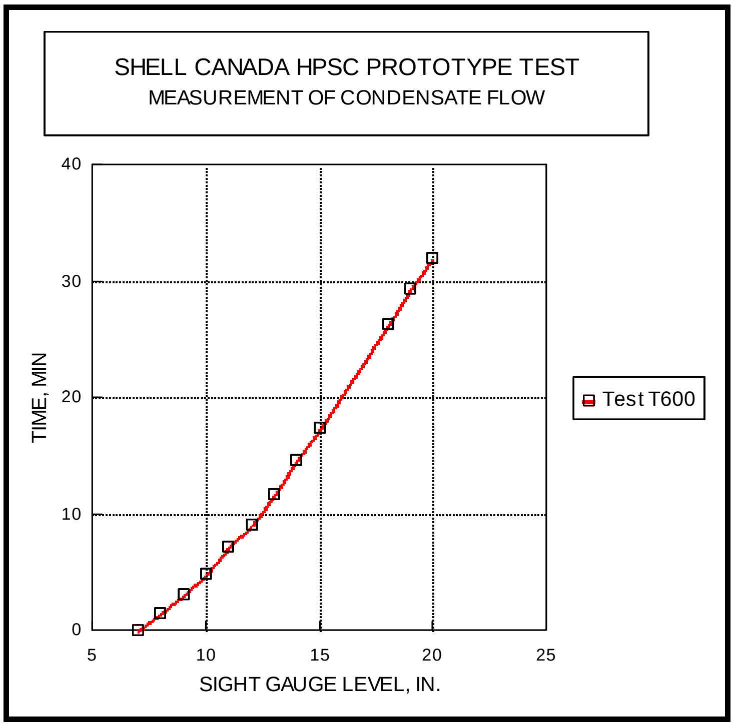


Figure 2. Condensate level change.

Condensate flow, and thus steam flow rate, was calculated from the ellipse equation that describes volume versus liquid level.

$$V = a^2 \pi [L - b/3 + (b - L)^3/3b^2]$$

where: V = Condensate volume in elliptical head, in^3
a = Head inside diameter/2 (79/2 = 39 in.)
b = Head inside height (39/2 = 19.5 in.)
L = Condensate level, in.

and $$m_s = \rho_c(V_2 - V_1) / dt$$

where: m_s = Condensate (and steam) flow rate, lbm/hr
ρ_c = Condensate density, lbm/ft^3
V_2 = Lower head water volume at end of test, ft^3
V_1 = Lower head water volume at start of test, ft^3
dt = Time interval between level measurements, hr

Steam-side duty was obtained from the enthalpy of superheated inlet steam and draining condensate. Enthalpies were based on measured steam and condensate temperatures and elliptical head pressure.

$$Qs = m_s (h_s - h_c)$$

where: Qs = Steam-side heat duty, Btu/hr
h_s = Inlet steam enthalpy, Btu/lbm
h_c = Condensate enthalpy, Btu/lbm

TEST RESULTS

Steady-State Performance

Steady-state data was recorded over a range of airflows for nominal fan speeds of 600, 1200, and 1600 rpm. This data was used to calculate air heater performance for comparison to the design code. Most of these performance tests were conducted with the air heater isolated from the incinerator by a slide gate and with the outlet duct hatch open. Having the air heater isolated allowed for manual control of the fan speed without interfering with plant operation. One performance test was conducted during normal air heater operation supplying heated air to the incinerator. A low pressure steam test was also recorded to provide additional validation data for the code. Steady-state temperatures were also recorded continuously overnight during normal air heater operation as it supplied hot air to the incinerator. Other steady data were recorded to calibrate pitot tubes and check thermocouples. The following table summarizes the steady-state tests conducted during the period from March 2, 1998, through March 6, 1998.

Table 2
STEADY STATE TESTS

TEST ID	DATE	TEST DESCRIPTION	NOMINAL INLET STEAM CONDITIONS
PITOT01	3/2/98	Isothermal Pitot Traverses, Cold Steady State, Hatch Open & Fan at 640, 1200, & 1600 RPM	Steam Off
T600	3/3/98	Low Load Steady State, Hatch Open & Fan at 600 RPM	23.5 psia, 344 F, 107 F Superheated
T1200	3/3/98	Medium Load Steady State, Hatch Open & Fan at 1200 RPM	22.8 psia, 341 F, 106 F Superheated
T1200R	3/3/98	Repeat of Previous Test T1200	22.8 psia, 341 F, 106F Superheated
T1600	3/3/98	High Load Steady State, Hatch Open & Fan at 1600 RPM	24.9 psia, 331 F, 91F Superheated
T0000S1	3/4/98	Isothermal Hot, Hatch Open, Fan Off, Steam On & Fan at 1600 RPM	25.8 psia, 237 F, 5 F Superheated
T1600P	3/4/98	Low Pressure Steam Steady State, Hatch Open & Fan at 1600 RPM	14.6 psia, 341 F, 130F Superheated
T1200PT	3/5/98	Hot Pitot Traverse at Normal Load Steady Operation, Hatch Closed & Fan at 1200 RPM	26.3 psia, 251 F, 8 F Superheated
T1200PM	3/5-6/98	Normal Overnight Steady-State Operation, Hatch Closed & Fan at 1200 RPM	23.8 psia, 240 F, 3 F Superheated

Hot Performance Tests

Six hot, steady-state performance tests were conducted, five with the air heater isolated from the incinerator and one during normal operation while supplying hot air to the incinerator. The air heater was isolated from the incinerator by closing a slide gate and removing the top hatch cover in the outlet duct. Fan speed was held relatively constant for each test at nominal values of 600, 1200, or 1600 rpm.

The latest version of the steam condenser design code (Reference 2) was used to predict the steady-state performance of the heat exchanger for the measured air and steam inlet conditions. Predictions included the heat duty, outlet air temperature, air-side pressure drop, and steam flow. Heat duty on the air-side is summarized for the six steady performance tests in Table 3, while the steam-side is summarized in Table 4. The air- and steam-sides are presented separately since heat flows were measured by completely independent methods. Air mass flow and temperature increase were measured on the air-side, while steam condensate flow and enthalpy change (via temperature and pressure) were measured on the steam-side. Since the air heater had seen about a month of service, predictions are presented for a clean heat exchanger and for a slightly fouled condition with 10% of the surface area unavailable due to deposits.

Table 3
MEASURED AND PREDICTED AIR-SIDE HEAT DUTY

Test ID (~Fan RPM)	Air-side Heat Duty -Measured- Btu/hr	Air-side Heat Duty -Predicted- [Clean] Btu/hr	Percent * Difference [Clean] %	Air-side Heat Duty -Predicted- [Fouled] Btu/hr	Percent* Difference [Fouled] %
600	4,772,575	4,522,613	5.24%	4,438,759	6.99%
1200	6,817,643	6,094,270	10.61%	5,947,264	12.77%
1200R	6,763,819	6,068,905	10.27%	5,923,655	12.42%
1200PT	3,617,899	3,717,577	-2.76%	3,660,344	-1.17%
1600	8,293,193	7,228,650	12.84%	7,036,090	15.16%
1600P	6,389,401	6,063,215	5.11%	5,907,217	7.55%

*A positive percentage indicates that measured performance exceeded the predicted value.

Table 4
MEASURED AND PREDICTED STEAM-SIDE HEAT DUTY

Test ID (~FAN RPM)	Steam-Side Heat Duty -Measured- Btu/hr	Steam-side Heat Duty -Predicted- [Clean] Btu/hr	Percent * Difference [Clean] %	Steam-side Heat Duty -Predicted- [Fouled] Btu/hr	Percent * Difference [Fouled] %
600	3,815,505	4,522,613	-18.53%	4,438,759	-16.33%
1200	5,290,831	6,094,270	-15.19%	5,947,264	-12.41%
1200R	5,628,786	6,068,905	-7.82%	5,923,655	-5.24%
1600	6,439,567	7,228,650	-12.25%	7,036,090	-9.26%

Air-side and steam-side predictions are compared to data in Figures 3 and 4. In addition to the 45^o line of perfect agreement, +10% and –10% dashed lines are also shown. Data points for the clean condition were also fit to a straight line by linear regression analysis. The air heater performed much better than predicted based on air-side measurements, which are more accurate than steam-side data based on the measurement techniques used. On the steam-side, air heater duty was below predictions. There was considerable uncertainty in the level gauge markings compared to the actual water level in the lower head, which was assumed to be a perfect ellipse. In addition, the design analysis was based on saturated steam, while the inlet steam ranged from 3 F to 130 F superheated, depending on time of day and air heater load. Air temperatures measured along the finned tube bundle are shown in Figure 5. The air heater appears to be oversized, since there is little additional increase in air temperature in the last few tube rows.

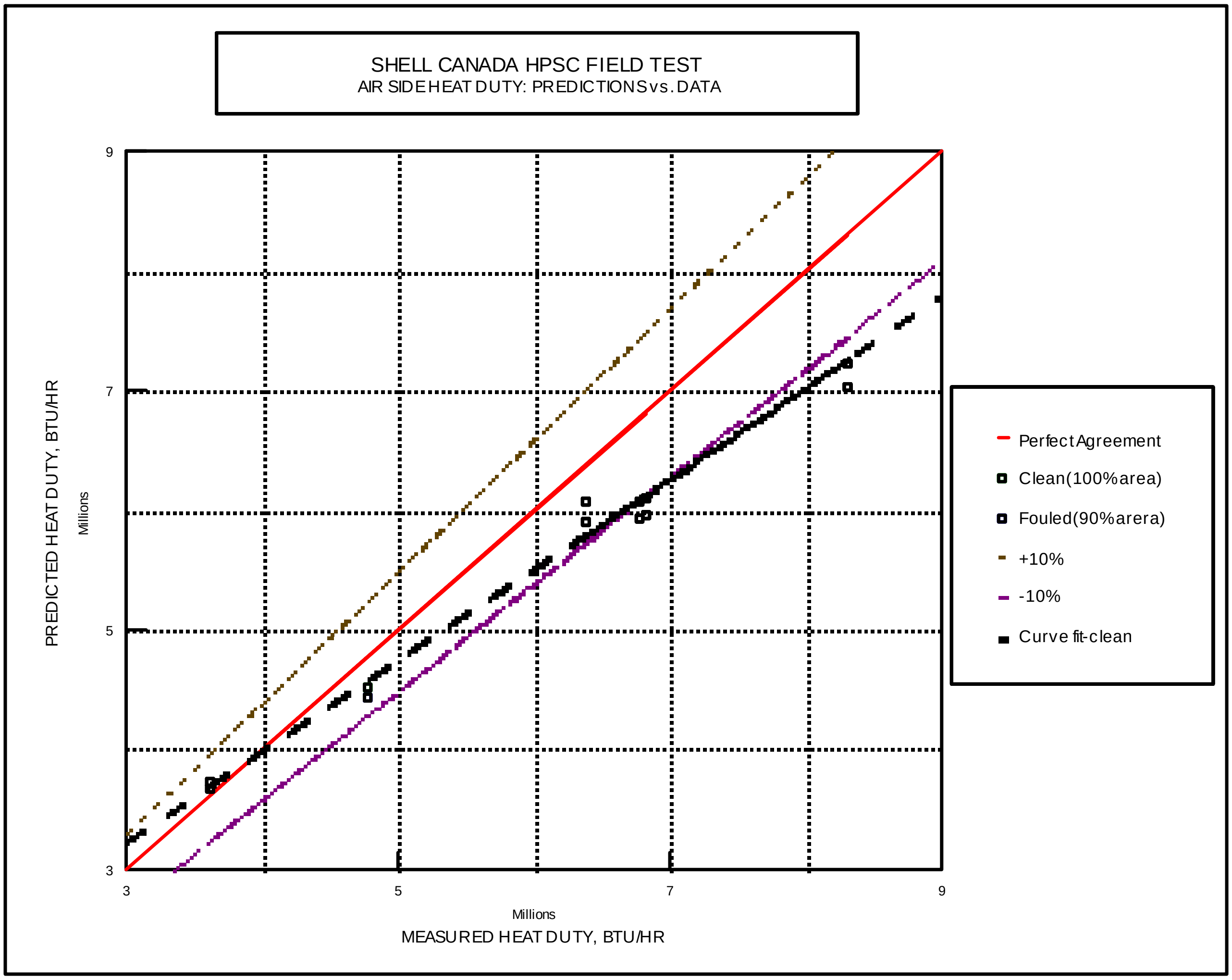


Figure 3. Air-side heat duty: predicted vs. measured.

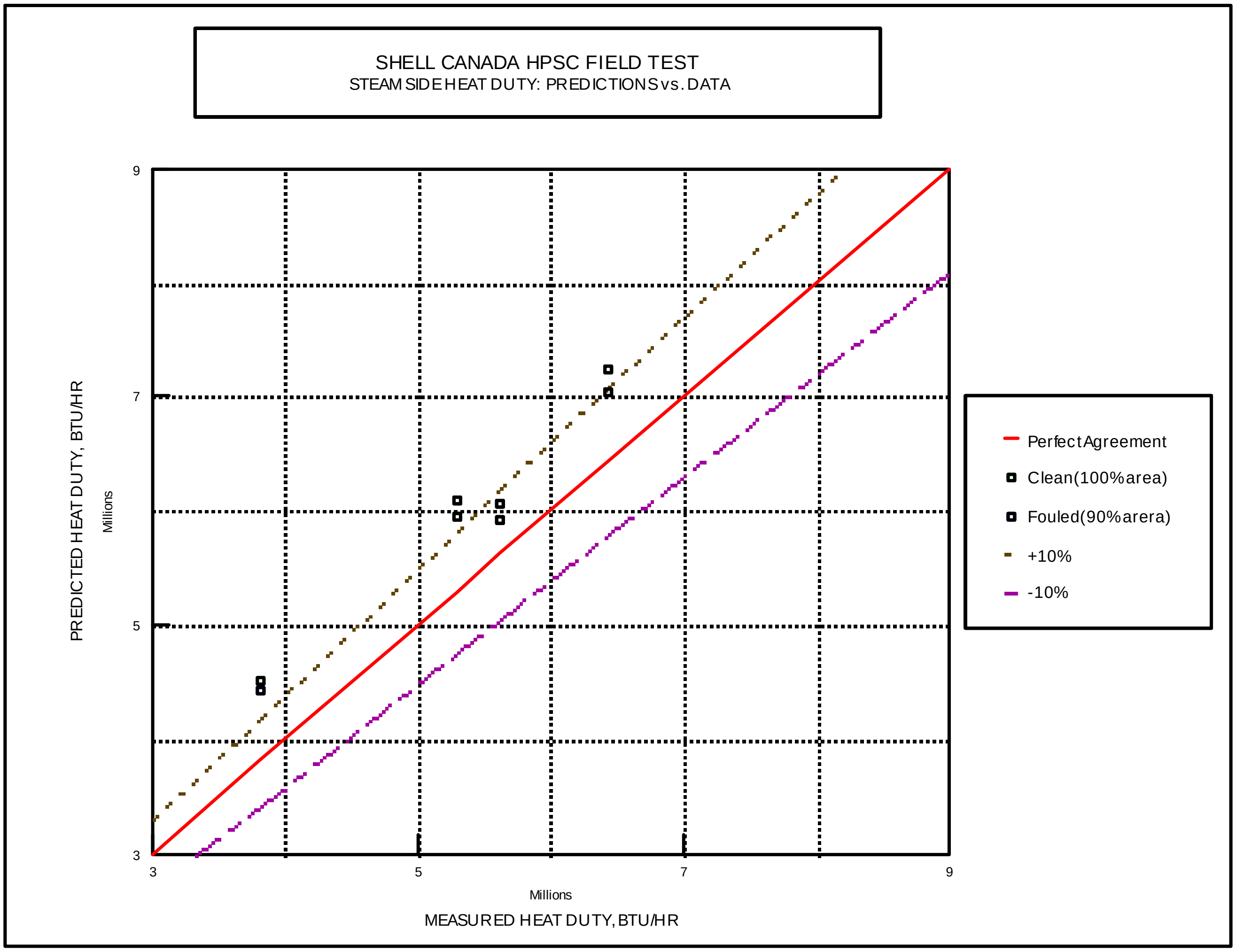


Figure 4. Steam-side heat duty: predicted vs. measured.

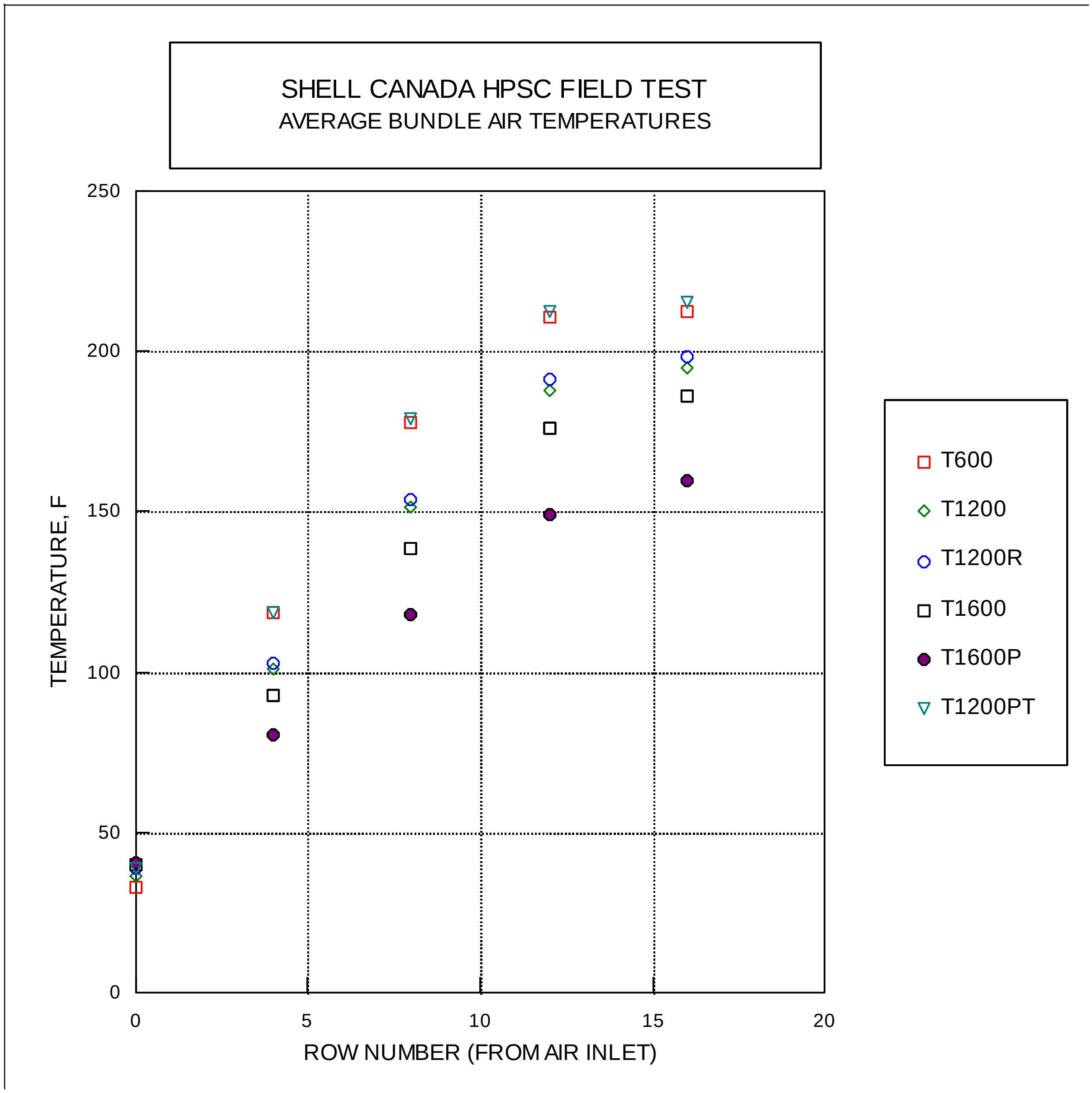


Figure 5. Measured air temperatures in tube bundle.

Dynamic Response

Several transients were run to record air heater response to upset conditions. These included a simulated steam trip, cold start up and fan trip and restart.

Steam Trip

This transient test simulated a loss of steam. The inlet steam valve was manually closed during steady operation. Figure 6 shows that the air heater responded very quickly and stabilized in about 20 minutes.

Cold Startup

This test simulated a cold start-up transient with both the fan and steam initially off. Figure 7 shows the air heater response when the steam and fan are turned on.

Fan Trip and Restart

This test simulated a fan trip transient during normal operation, followed by a fan restart. Figure 8 shows the air heater response to a fan trip and fan ramp-up. The fan restart was relatively slow since it was hand controlled in increments. In all cases, the air heater reached equilibrium within a relatively short period of time with no special operating procedures required.

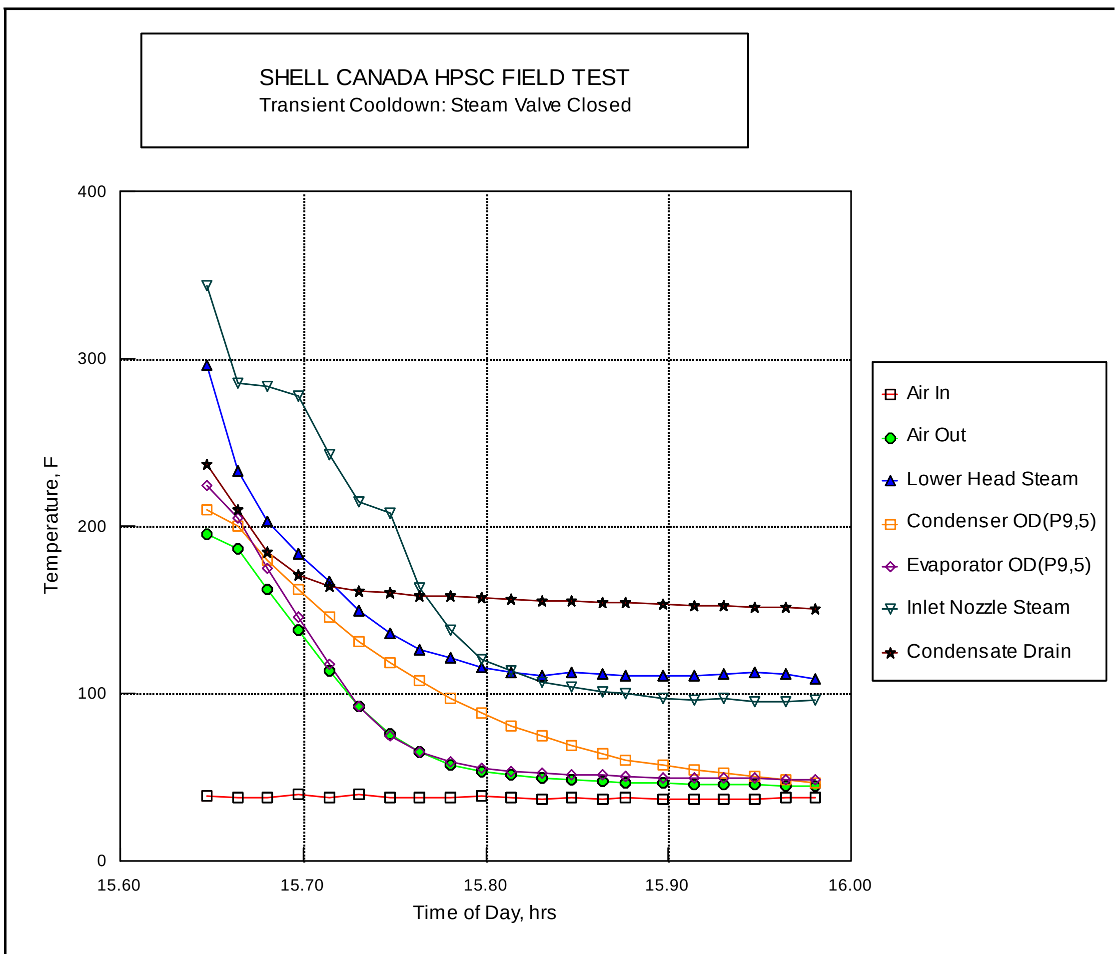


Figure 6. Steam trip transient.

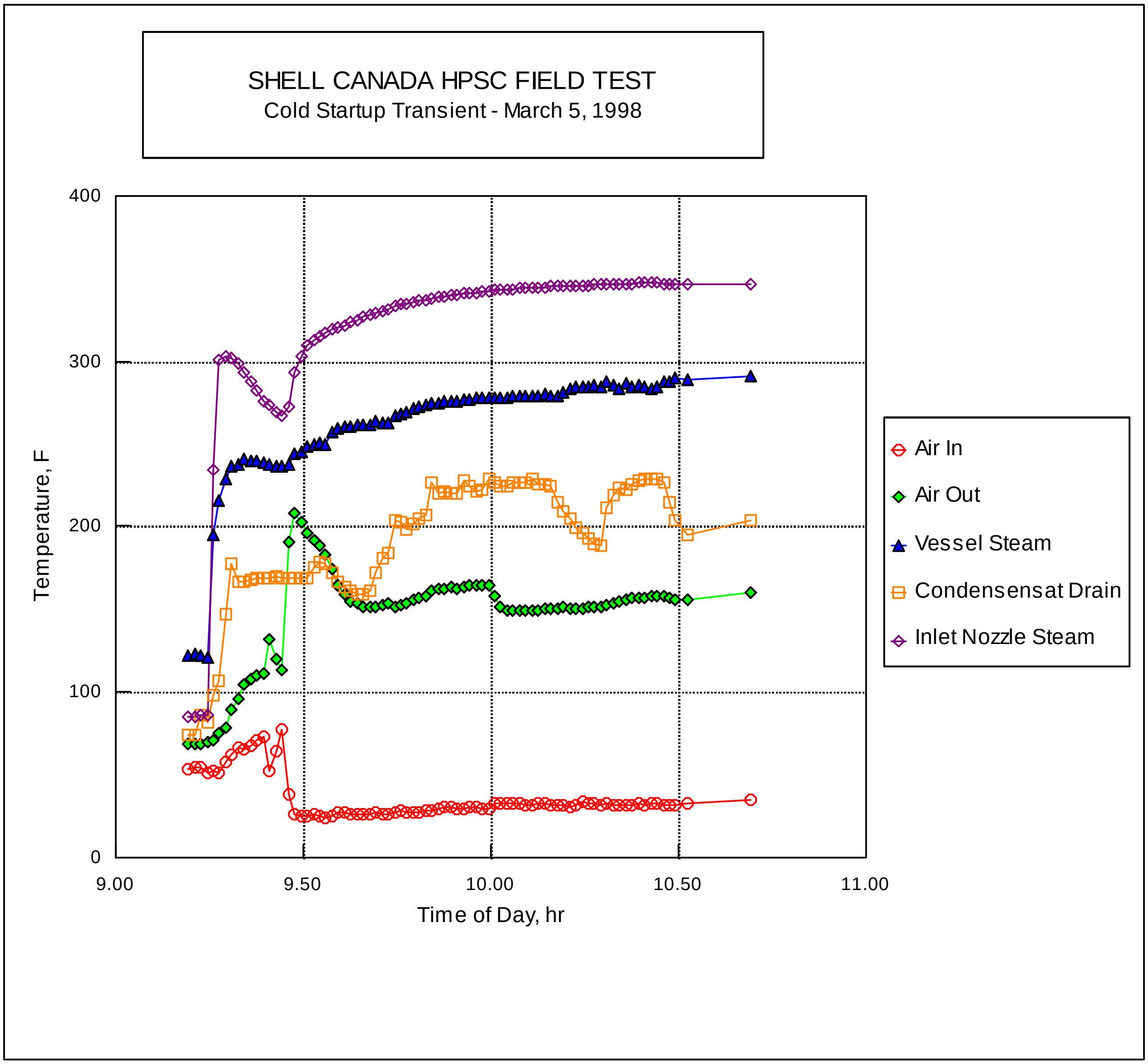


Figure 7. Cold start-up transient.

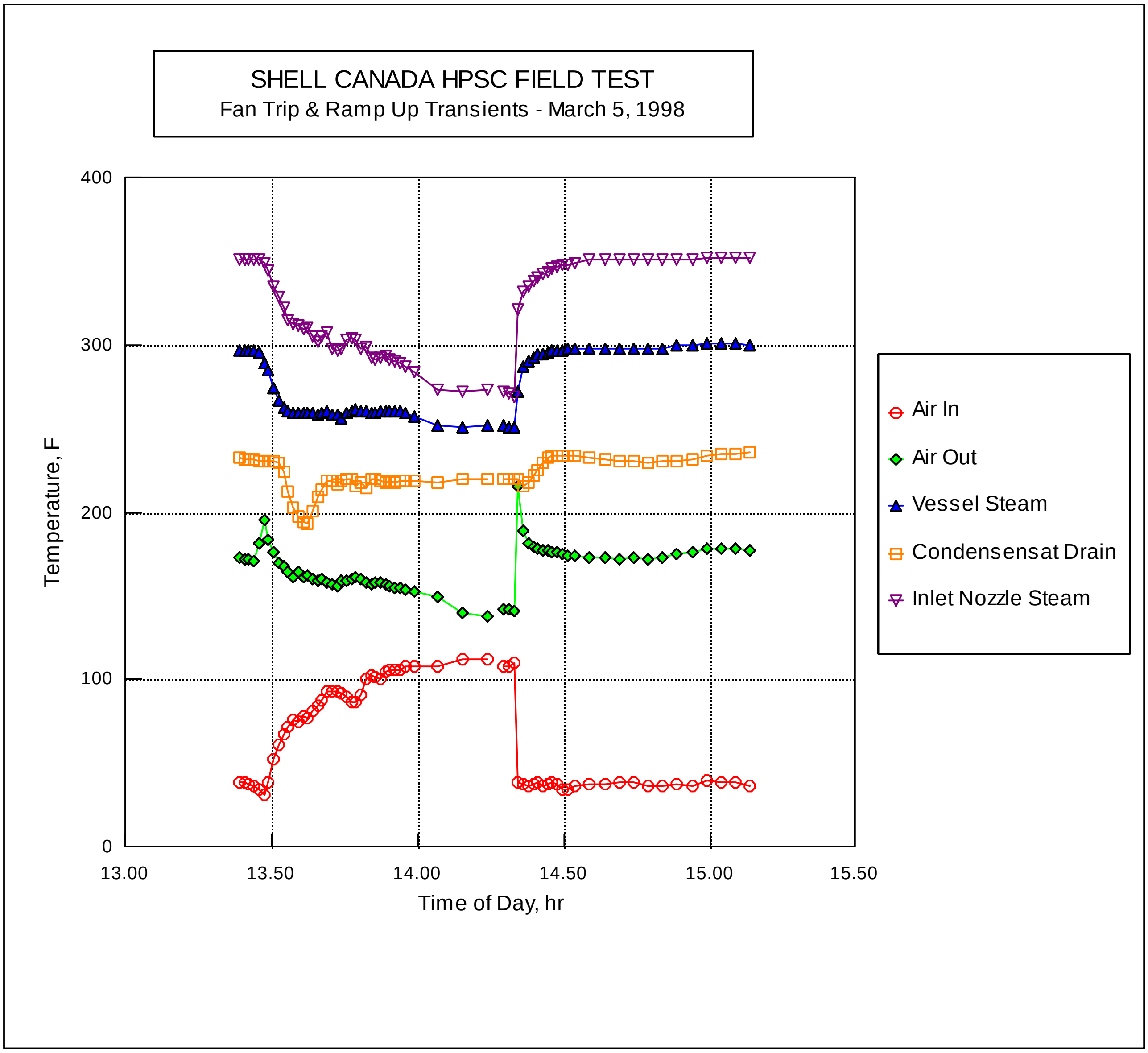


Figure 8. Fan trip and restart.

OPERATIONS IMPACT

After the heat exchanger was brought on line, plant operations were impacted the following three ways:

1. Overall incinerator fuel gas consumption has decreased an average of 17.9% due to combustion air preheating.

2. During incinerator trips, the thermal capacitance inherent in the heat exchanger helps maintain the incinerator stack temperature for a longer period of time. This gives plant operations additional time to bring the incinerator back online before a low stack temperature emission excursion can occur.

3. The plant was able to take off line one-quarter of their secondary air-cooled steam condenser, eliminating the operating and maintenance costs associated with this portion of the equipment.

CONCLUSIONS

Based upon the testing performed, the following conclusions can be made about the prototype technology employed:

- Air heater performance compared favorably to the predicted design. Measured air side performance was about 12% better than design predictions at nominal load.

- Transient testing showed that the unit responded very quickly to air or steam side upsets without any damage to the tube bundle.

- Unit operation is very simple and self regulating.

As part of the development program to improve the technology, Hudson Products will continue to monitor the heat exchanger performance over the next several years. This monitoring will allow for more accurate forecast of heat pipe life and assist in verifying long-term application of the technology.

REFERENCES

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2. H.W. Wahle, "Steam Condenser Design Program – vbHPSC Beta Version 1.2," 25737-005, HPSC-9, March 10, 1998.