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On the 13 000 kg/hr Supercritical Pressure Test Boiler*

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This paper presents the features of a 13 000 kg/hr supercritical test boiler which was erected at the third factory of the Ishikawajima-Harima Heavy Industries Company and also describes the experimental results of its steady and dynamic test. Especially the analysis of dynamic test is made in comparison with the simulation by the analogue computer. Test results can be applied to development of the central supercritical pressure steam generator.

1. Description of test boiler

In order to solve the problems encountered to develop a large supercritical pressure boiler, an epochal large size test boiler designed to generate 13 000 kg of steam per hour at pressures and temperatures up to 350 kg/cm²g and 600°C was erected in December, 1960 and is being operated at the third factory of the Ishikawajima-Harima Heavy Industries Company.

This paper deals specifically with the features of this boiler and the experimental results of its steady and dynamic test.

There exist a considerable number of investigations concerning the behavior of the supercritical pressure steam. Some of their interesting problems are as follows.

(1) Heat transfer in this transition region differs from that of sub-critical steam⁽¹⁾⁽²⁾.

(2) The deposits are not always accumulated in the transition region, but in all of tube length from inlet to outlet of boiler⁽³⁾⁽⁴⁾.

(3) Heat transfer rate in the transition region is relatively high⁽⁵⁾.

(4) The local deviation of the heat absorbed causes a distribution of fluid temperature, and accordingly the shifting of the transition region along the tubes⁽⁴⁾.

The following features of the test boiler were taken into consideration to confirm actually the above problems.

Principal specifications are listed in Table 1. The boiler is outdoor type except the auxiliary

Table 1 Specification of test boiler

Type of boiler	Ishikawajima-Harima once through boiler
Evaporation at MCR	13 000 kg/hr
Design pressure at superheater outlet	380 kg/cm ² g
Normal steam pressure	350 kg/cm ² g
Normal steam temperature	600°C
Feed water temperature at economizer inlet	130°C
Method of firing	Heavy oil firing (pulverized coal firing in future)
Draft system	Balanced draft
Feed water pump	
Type	Ishikawajima-Harima motor driven six stage plunger type
Capacity	16 000 kg/hr
Delivery pressure	420 kg/cm ² g
Method of flow control	Speed control
Fans	
Type of forced draft fan	Ishikawajima-Harima motor driven single suction turbo-vane type
Type of induced draft fan	Ishikawajima-Harima motor driven double suction turbo-vane type
Oil burning equipment	
Type of oil burner	Ishikawajima-Harima P-BI type oil pressure atomizing type
No. of burners	Two
Capacity	Each 800 kg/hr
Oil pressure	Max. 21 kg/cm ² g
Type of oil pump	Ishikawajima-Harima motor driven gear pump
Deaerator	
Type	Ishikawajima-Harima tray spray type
Feed water demineralizer	
Type	Isuzu two mixed-bed ion exchangers
Maker of valve	Okano valve Co.
Automatic control equipment	
Type	Bailey air operated type
Maker of boiler tubings, pipings	Sumitomo Kinzoku Kogyo Co.

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machine room at the front of the boiler as shown in Fig. 1 and Fig. 2. The boiler proper consists of a primary economizer, a secondary economizer, a rear wall, a left wall, a right wall, a front and roof radiant superheater of furnace and a pendant superheater which are connected with pipings respectively. The tube has a 32 mm outside diameter with 6 to 7.5 mm wall except the economizer supporting tubes having a 50.8 mm outside diameter. The furnace rear wall and front and roof wall comprise up and down flow panels with nine parallel tangent tubes.

Both side walls are horizontal meander type with nine parallel tangent tubes in which the fluid passes through upward. The transition region was planned to have such position at the furnace wall as receiving less radiant heat between left and right wall. The convection superheater is pendant type with fourteen rows installed at the rear of the slag screen. For the radiant superheater tubes of front and roof wall and the convection superheater tubes and outlet header and piping, ASTM type 321

stainless steel was adopted so that it would be possible to do testing at over 600°C.

The boiler is fired with a heavy oil from two pressure atomizing burners on the front wall. But the gas temperature, the tube spacings, the gas velocity were so selected as to be suitable for the pulverized coal firing in future and also the dry hopper was adopted.

The gases pass through the slag screen, the convection superheater and downward through the economizer and upward through the tubular air heater and then are exhausted through a stack with an induced draft fan on the top of structure.

The feed water is demineralized from the city water in two mixed bed ion exchangers in series with a preflock equipment, a carbon filter and a cellulose filter. After being deaerated, the feed water is fed to the economizer by a six stage plunger pump. The feed water is injected with ammonia for PH control and with hydrazine for scavenging

Fig. 1 13 t/hr supercritical pressure test boiler

Fig. 2 13 t/hr supercritical pressure test boiler

Fig. 3 An example of steady test results
●—Test No.1 12.1 t/hr × 353 kg/cm²g × 591°C
○—Test No.2 9.97 t/hr × 343.5 kg/cm²g × 589°C

of dissolved oxygen.

The steam is supplied to the turbine test shops after being reduced to $120 \text{ kg/cm}^2\text{g}$ from $350 \text{ kg/cm}^2\text{g}$ by two stage reducing valves. During the starting time of unit, the mixture of water and steam passes through a bypass pipe with a motor-driven valve and a multi-stage orifice and goes into a flash chamber to be released from a silencer under the sea water level, therefore the silent operation is easily made. The boiler is normally put on the line from a cold condition to $350 \text{ kg/cm}^2\text{g}$ and 600°C in one and half hours.

2. Steady test

Steady test was carried out under the steam conditions of $350 \text{ kg/cm}^2\text{g} \times 600^\circ\text{C}$, $350 \text{ kg/cm}^2\text{g} \times 550^\circ\text{C}$, $300 \text{ kg/cm}^2\text{g} \times 600^\circ\text{C}$ and $250 \text{ kg/cm}^2\text{g} \times 550^\circ\text{C}$ and under the steam loads of 13 t/hr, 10 t/hr and 4.5 t/hr respectively. After the boiler had reached a steady state, the flow, pressure and temperature were measured.

Fig. 3 shows an example of a series of the tests. Here the heat absorption rates are based on projected area for the furnace wall and full circumferential area for the convection surface. The actual distribution of fluid temperature differs slightly from the designed condition due to the less heat absorbed in the front and roof radiant superheater, however the transition point is located closely at the anticipated

position as shown in Fig. 3.

It was confirmed that the transition region tends to move into inlet side of the boiler only slightly with the decreased load and the decreased pressure, but in a large amount with the increased excess air.

The pressure drop of fluid at a rated steam condition was 26 kg/cm^2 which was close to 28 kg/cm^2 anticipated from calculation.

Fig. 4 shows the relation between the surface temperatures of heated tubes and unheated tubes. It is noticed that the temperature difference is only about 20°C in the transition region, but approximately 35 to 45°C at the liquid phase below the transition region.

Fig. 5 shows the relative heat transfer coefficient inside the tube represented as a lateral axis of enthalpy having a transition point as an origin.

It is shown that the heat transfer decreases at liquid phase near the transition region and reaches its maximum at the transition point, then decreasing with an increase of enthalpy.

Fig. 6 shows the temperature deviation of the unheated tube surface in parallel from its average. This deviation does not exceed 5°C for the furnace wall tubes. The unbalance of fluid flow obtained from this temperature deviation is within the allowable limit due to the allowance of the tube wall thickness.

The well maintained feed water condition contributed to a great extent to the successful

Fig. 4 Distribution of surface temperature of heated tubes and fluid temperature

Fig. 5 Relative heat transfer coefficient

Fig. 6 Temperature deviation of unheated tube surface in parallel from its average

operation of the test boiler. Since all of the feed water was obtained from the city water and the transition region was installed in the furnace, special attention was paid to the feed water condition.

The water was maintained within the limits shown on Table 2 during the test operation.

3. Dynamic test

3.1 Test results

A simplified diagram of the pneumatic control system used with the test boiler is shown in Fig. 7.

To study the effect of the transient behavior of a fluid in the boiler for given input and output disturbances, at first, tests were carried out with the boiler on manual control and with step impulse by one of the three valves such as feed water flow, fuel flow, and the opening of boiler outlet valve.

The measured changes of pressure and temperature of a fluid under such conditions are plotted against time, in Figs. 9, 10 and 11.

In these tests, the particular phenomenon revealed are as follows.

- (1) Under sudden change of feed water flow, temperature θ_l near the transition point

Fig. 7 A simplified diagram of the control system

deviates in an unexpected direction first, and then recovers its normal behavior.

- (2) Under sudden change of fuel flow, steam pressure at boiler outlet shows such a behavior as cubic.
- (3) Under sudden change of the valve opening at boiler outlet, temperature θ_l shows the quickest response and θ_{ri} and θ_f in the superheater region show such a behavior as cubic.

The above-mentioned particular points coincide with the test results of the supercritical experimental boiler of Foster Wheeler Corporation⁽⁶⁾.

The next step of study is the test of over-all characteristic covering all control equipments on automatic positions. The tests were carried out with step impulse by the opening of boiler outlet valve. The results shown in Fig. 13, indicate the following facts.

- (1) The pressure deviation from the set point converges quickly to zero, which shows the propriety of pressure control by feed water flow.
- (2) The temperature deviation of θ_l from the set point converges to zero within 500 sec., which shows the propriety of fuel control. But that of θ_f brings a permanent offset, which results from the fact that the control loop of superheater temperature by spray water was not used.

Measured points are shown in Fig. 7.

3.2 A consideration on the analytical method of dynamic characteristics

Notation

Q_0 : Feed water flow $q_0 = \Delta Q_0 / Q_0$

Q_l : Fluid flow before transition point $q_l = \Delta Q_l / Q_l$

Q_s : Fluid flow at superheater inlet $q_s = \Delta Q_s / Q_s$

Q_h : Fluid flow at superheater outlet $q_h = \Delta Q_h / Q_h$

R_0 : Fuel flow $r_0 = \Delta R_0 / R_0$

A : Valve opening at boiler outlet

$a = \Delta A / A$

Table 2 Conditions of raw water and feed water

		Total solid ppm	SiO ₂ ppm	Fe ppm	Cu ppm	Conductivity $\mu v/cm$	O ₂ ppm	PH	Turbidity
Raw water	Design	170	20.4	0.07	—	—	—	—	3.5°
	Actual	—	2.14	0.29	0.0027	160	—	—	16°
Feed water	Design	0.2	0.02	0.01	0.01	1.0	0.007	9~9.5	—
	Actual	0.17	0.02	0.01	0.0027	0.5	0.004~0.007	9~9.5	—

- P : Steam pressure $p = \Delta P/P$
 Θ : Fluid temperature $\theta = \Delta \Theta/(\Theta - \theta_i)$
 Θ_p : Metal temperature of tube $\theta p = \Delta \Theta_p/(\Theta_p - \Theta)$
 I : Enthalpy of steam $i = \Delta I/(I - I_i)$
 V : Specific volume of steam $Y = V/V_i$
 R_1 : Heat transmission to fluid per unit length of tube
 C : Heat capacity per unit length of tube
 H : Equivalent heat transfer rate per unit length of tube
 F : Cross section of tube
 L : Length in direction of flow
 T_a, T_V, T_F : Time constants
 t : time

$$k_i = \frac{I - I_i}{\Theta - \theta_i} \cdot \left(\frac{\partial \Theta}{\partial I} \right)$$

$$k_p = \frac{P}{\Theta - \theta_i} \cdot \left(\frac{\partial \Theta}{\partial P} \right)$$

$$k_r = \left(\frac{\partial \Theta}{\partial I} \right) / \left(\frac{\partial \Theta}{\partial I} \right)_i$$

(Subscript)

- i : Inlet of region in question
 k : Economizer inlet
 re : Furnace rear wall outlet
 l : Furnace left side wall outlet
 ri : Furnace right side wall outlet
 f : Boiler outlet
 T : Transition point

As Mr. Terano's theoretical equations⁽⁷⁾, which were derived from systematic analysis of dynamic characteristics of once-through boiler, are based on steam property at subcritical pressure, we can not use these indiscriminately for supercritical pressure, especially for transition region.

In the region before transition point (furnace left side wall of this test boiler), approximate response equations are derived from the fundamental equation, assuming as follows;

- (1) Response of fluid property by disturbances such as feed water flow, fuel flow, and inlet temperature is divided into two parts-response without pressure change and response with pressure change only, and these two responses can be superposed.

- (2) Change of specific volume of fluid is proportional to enthalpy.

a. Response equation of temperature

At first, the fundamental equation without pressure change is transformed into the following partial differential equation of enthalpy change, using the above-mentioned assumption⁽²⁾.

$$\frac{\partial^2 i}{\partial L^2} + \left\{ \frac{C(I - I_i)}{R_i \cdot L \left(1 + \frac{C}{H} S \right)} \left(\frac{\partial \Theta}{\partial I} \right) + \frac{F}{Q_0 \cdot V} \right\} \times S \cdot \frac{\partial i}{\partial L} = 0$$

From the above equation and boundary conditions at region inlet, we have an approximate response equation as follows;

$$i_{l1} = \frac{1}{Y-1} \cdot \frac{1 - Y e^{-T_a S}}{\frac{T_a}{\ln Y} \cdot S - 1} \left(\frac{\gamma_0}{1 + T_F S} - q_0 \right) + e^{-T_a S} \cdot i_i$$

Therefore, the response equation of temperature without pressure change becomes as follows;

$$\theta_{l1} = k_i \frac{1}{Y-1} \cdot \frac{1 - Y e^{-T_a S}}{\frac{T_a}{\ln Y} \cdot S - 1} \left(\frac{\gamma_0}{1 + T_F S} - q_0 \right) + k_r \cdot e^{-T_a S} \cdot \theta_i$$

Next, according to the following consideration, we have a response equation of temperature with pressure change only. That is, in Fig. 8, when the pressure changes instantaneously from P to P' , the state at region outlet moves from A to B (constant temperature change). In this case, the required heat absorption is $(I - I_i + \Delta I)$, however only $(I - I_i)$ is given actually.

This indicates that a change of heat absorption $\Delta I/(I - I_i)$ is given to this system as a disturbance and so the change from B to C corresponds to the transfer function of temperature for the disturbance of heat absorption.

Then, we have the following equation

$$\theta_{l2} = k_p \cdot \frac{1}{Y-1} \cdot \frac{1 - Y e^{-T_a S}}{\frac{T_a}{\ln Y} \cdot S - 1} \cdot P$$

From above equation and assumption of superposition, finally we have the response equation of temperature as follows;

$$\theta_l = \theta_{l1} + \theta_{l2} = \frac{1}{Y-1} \cdot \frac{1 - Y e^{-T_a S}}{\frac{T_a}{\ln Y} \cdot S - 1} \times \left\{ k_i \left(\frac{\gamma_0}{1 + T_F S} - q_0 \right) + k_p P \right\} + k_r \cdot e^{-T_a S} \cdot \theta_i$$

Fig. 8

b. Response equation of pressure

Assumption that the change of pressure is equal at all points in the boiler simplifies the derivation of a theoretical equation.

Change of steam flow at superheater outlet q_h ;

$$q_h = a + \frac{P}{\sqrt{P/v}} \cdot \frac{\partial \sqrt{P/v}}{\partial P} \cdot p + \frac{\theta_f - \theta_{ri}}{\sqrt{P/v}} \times \frac{\partial \sqrt{P/v}}{\partial \theta_f} \cdot \theta_f \equiv a + k_1 p - k_2 \theta_f$$

Change of steam flow at superheater inlet q_s ;

$$q_s = q_h - \frac{B}{Q_h \cdot V_h} \cdot S(\theta_h' - p) \equiv q_h - T_a \cdot S(\theta_h' - p)$$

where B , V_h and θ_h' represent volume occupied by steam, the average specific volume and the change of average temperature of superheater.

Change of fluid flow before transition point q_l is obtained on the assumption that an equivalent discontinuous change of specific volume at transition point occurs by means of a change of flow due to displacement of transition point $l = \Delta L/L_l$, then

$$q_l = q_s - \frac{FL_l}{Q_s} \left(\frac{1}{V'} - \frac{1}{V''} \right) S \cdot l \equiv q_s + T_q \cdot S \cdot l$$

where V' and V'' represent equivalent specific volumes of liquid phase and steam phase, respectively. Displacement of transition point l ;

$$l = \frac{P}{\theta_l - \theta_i} \cdot \frac{\partial \theta_r}{\partial p} \cdot p - \theta_l \equiv k_T p - \theta_l$$

From above equations we have the response equation of pressure as follows;

Fig. 9 Disturbance test of feed water flow

$$p = \frac{1}{k_1} \cdot \frac{1}{1 + \frac{k_T \cdot T_q + T_a}{k_1} S} \times \{q_l - a + T_q S \theta_l + T_a S \theta_h' + k_2 \theta_f\}$$

3.3 Comparison with calculation results by analytical method**a. Independent disturbance test, case 1**

At first, in order to ascertain the theoretical equations near transition point which were obtained in the previous paragraph, we assume that all variables in the right terms of response equations are independent disturbances and calculate θ_l and p by using the measured values themselves.

We believe this is the best analytical method for judging whether our consideration with regard to the region in question is right or not.

Calculation results coincide well with the measured values as shown by chain lines in Figs. 9, 10 and 11.

b. Independent disturbance test, case 2

Next, we reduced disturbances to only three feed water flow, fuel flow, and valve opening at boiler outlet and connected other variables within a closed circuit, and analyzed the dynamic characteristics of the whole system by analogue computer. The economizer, furnace rear wall and superheater were calculated by Mr. Terano's theoretical equation, and the furnace left wall was calculated based on our consideration in the previous paragraph.

Fig. 10 Disturbance test of fuel flow

Whole block diagram is shown in Fig.12, in which step change was given to q_0 , γ_0 , and a respectively.

Calculated results are plotted in Figs. 9, 10 and 11 in dotted lines.

c. A consideration for independent disturbance test

(1) Under sudden change of feed water flow, temperature near the transition point deviates temporarily in an unexpected direction, because the pressure also changes when the feed water flow changes and it responds to the temperature in reverse direction against the effect of flow, and besides it affects very much. Therefore, until the effect of temperature change at inlet becomes large, the response will be kept in reverse direction. In other words it is because of $\partial\theta/\partial I$ near transition point being small and $\partial\theta/\partial P$ being large. For instance, when the ratio of feed water flow and fuel flow was changed, by which condition transition point moved in inlet direction and $\partial\theta/\partial I$ at furnace left side wall outlet became large, a response of θ_l in reverse direction did not appear at all.

(2) Under sudden change of fuel flow, steam pressure responds in cubic fashion. First a peak value occurs due to the change of evaporation with sudden change of fuel flow.

Since this change of evaporation has derivative characteristics, the pressure will temporarily recover until a second peak takes place under the effect of temperature change at boiler outlet.

(3) Under sudden change of the valve opening at boiler outlet, temperature θ_l shows the quickest

response, which is because of the biggest value of $\partial\theta/\partial I$ near transition point. As for temperature behavior of superheater which is cubic, the following process is considered. Temperature change by flow change at boiler outlet appears first and it recovers once due to derivative characteristic of this flow

Fig. 11 Disturbance test of valve opening at boiler outlet

Fig. 12 Block diagram

Fig. 13 Control characteristic test

change, but soon the effect of temperature disturbance at superheater inlet appears, forcing the first change to continue. Difference between analysed results and measured values is due to the assumption that time delay in superheater temperature is approximated by an integral delay of second order.

d. Control characteristics

In the block diagram shown in Fig. 12, dotted lines indicate parts of control equipments.

Now, dotted lines in Fig. 13 show calculation results for the step change of "a" by analogue computer. In this case, proportional constants of controller were chosen equal to the set point during testing and the reset time constants were set at the most appropriate theoretical values. The actual responses of steam pressure and feed water flow coincide very well with the calculated, which fact indicates the propriety of the response equation about steam pressure. Responses of temperature, however, show considerable discrepancy between calculated and measured values, which is considered to be due to improper choice of reset time constant

of controllers and change of distribution of heat absorption. Permanent off-set of temperature at boiler outlet will become zero if a spray control circuit is used.

4. Conclusions

The results obtained from the steady and dynamic test are summarized as follows.

(1) The boiler is held stable in the load range of 4.5 to 13 t/hr.

(2) The fluid temperature and pressure distribution close to the designed condition, especially the location of the transition region occupied in the boiler coincide closely with the anticipated location.

(3) Heat transfer in the transition region is considerably great.

(4) The fluid flow distribution in the parallel tubes is within the allowable ranges.

(5) The feed water condition was maintained within the limits.

(6) The approximate response formula to be applied with sufficient accuracy was introduced from the analysis of dynamic test in the supercritical region.

(7) It became evident that the balance of fire and water can be achieved rapidly by controlling the pressure with the feed water flow and controlling the combustion rate with both the PI action of the temperature located in the transition region and the P action of the feed water flow.

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