Nucl. Sci. and Tech, Vol. 4, No. 2 (2014), pp. 07-14

# Nuclear Science and Technology

Journal homepage: https://jnst.vn/index.php/nst

# Preliminary test results of anuclear pressurized thermal hydraulictesting facility.

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**Abstract:** In this work, the main features of a nuclear pressurizedthermal hydraulic testing facility named BKTF (BK testing facility) are presented. The facility consists of major systems and components similar to the standard PWR prototype, and it is a volume and pressure scaled model with the total mass of primary coolant in order of hundreds of kilograms. The model is designed to work at normal temperature and pressure less than 160 $^{0}$C and 10 bars respectively. In this part of research, preliminary results of cold test and hot test are presented. These results show that the system is qualified enough to work under designed pressure and temperature. It is seen that the similarity between temperature and pressure trend lines with these from a real power system during starting up process. In addition, natural circulation was appreciated and compared with forced circulation during the cooling down process and this may help us to have a better vision on passive safety and completely recognize about safety system of nuclear power plant.

***Keywords:*** *nuclear thermal hydraulic, pressurized water reactor, testing facility, LOCA, cold test, hot test, start up, cooling down natural circulation***.**

## I. INTRODUCTION

Testing facility is designed to perform experiments which increase man's understanding the behaviors and, sometimes the nature of phenomena happened in a power plant. It is clear that the sizes of various components and test conditions are so important that properties of plant can be properly simulated. However, it isimpossible to scale down every sizes and testing conditions of all components, so that every testing facility is designed to performed just some tests which can simulate the behaviors that are proven to happen in a real power plant system. In the field of nuclear safety related to thermal hydraulics, small break loss of coolant accident(SB LOCA)is more likely to occur and that its consequences could be sufficiently severe to warrant safety concerns. In this type of accident, system pressure do not decease rapidly and transient happens in a such quite long period of time that the facility is capable with the change of hydraulics for core cooling in very high temperature and pressure. On the other hand, large break loss of coolant accident (LB-LOCA), the controlling hydraulic phenomena are really difficult. Furthermore, the reflood has been considered as the most crucial period and properly simulated tests can provide between 200-300 $^{0}$F (150$^{0}$C) safety margin. Consequently, many testing facilities adopt system pressures during experiments only several bars [1]**.**

In this paper, a new constructed testing facility and its preliminary test results are presented. The BKTF is built with aims including: to successfully set up an experiment facility working at high temperature up to 160$^{0}$C with the pressure range of 1.0 MPa used in many

https://doi.org/10.53747/jnst.v4i2.221

Received 19 May 2014, accepted 29 September 2014

experiment systems, to connect theoretical works with experiments in the field nuclear thermal hydraulics, to strength capabilities of complex designs related to interdisciplinary science and engineering (physics, heat transfers, mechanics, materials, instrumentation and control..). The potential uses of the facility are related experiments for basic heat transfers, code validation and safety analysis, loss of coolant accidents, phenomena of water hammer andnatural circulation of low void fraction primary coolant or high void fraction steam flow, and passive safety researches.

## II. GOVERNING EQUATIONS

In testing facility we use electrical heaters instead of nuclear fission to supply thermal energy to primary circuit, thus

$$\dot{Q} = P_{heaters} \quad (1)$$

In practical, with a very good insulation we can consider the maximum useful work for control volume-reactor which contains heat generator (electrical or fission) is [2]:

$$\dot{W}_{u,max} = \dot{m}_p(h_{out} - h_{in})_{reactor} \quad (2)$$

For other components, consider each with multiple inlet and outlet flow streams operating at steady state and surround it with a non-deformable, stationary control volume. Applying the law of energy conservation for a control volume like steam generator, pump, turbine, condenser. We have:

$$\sum_{k=1}^{i}(\dot{mh})_{in,k} - \sum_{k=1}^{i}(\dot{mh})_{out,k} = \dot{W}_{shaft} - \dot{Q} \quad (3)$$

$$\dot{m} = \sum_{k=1}^{i}\dot{m}_{in,k} = \sum_{k=1}^{i}\dot{m}_{out,k} \quad (4)$$

For any heat exchanger used in the system, the basic equations for heat transfer rate, boiling and condensation process can be expressed respectively as follows [3,4]

$$\dot{q} = UA\Delta T_{over\ all} \quad (5)$$

$$\frac{C_l \Delta T_x}{h_{fg} Pr_l} = C_{sf}\left[\frac{q''}{\mu_l h_{fg}}\sqrt{\frac{g_c \sigma}{g(\rho_l - \rho_v)}}\right]^{0.33} \quad (6)$$

$$Nu^* = Co = \frac{h_m(\frac{v_l^2}{g})^{1/3}}{k_l} = 1.514\ Re_x^{-1/3} \quad (7)$$

For the hydraulic of water flow, mechanical energy is converted into heat in the viscous boundary layer along the pipe walls and is lost from the flow, then we have equation for head at any point of stream flow

$$h = z + \frac{p}{\rho g} + \frac{V^2}{2g} + \int_{x0}^{x}\frac{f}{D}\frac{V(x)^2}{2g}dx \quad (8)$$

we assume that the pipe diameter *D* stays constant. By continuity condition, we then know that the fluid velocity *V* stays constant along the pipe. With *D* and *V* constant we can integrate the viscous head equation and solve for the pressure at point B if we know pressure at point A.

$$p_B = p_A - \rho g\left(\Delta z + f\frac{L}{D}\frac{V^2}{2g}\right) \quad (9)$$

where *L* is the pipe length between points A and B, and Δ*z* is the change in pipe elevation ($z_B$ - $z_A$). Note that Δ*z* will be negative if the pipe at B is lower than at A. The viscous head term is scaled by the pipe friction factor *f*. In general, *f* depends on the Reynolds Number*Re* of the pipe flow, and the relative roughness *e*/*D* of the pipe wall [5].

## III. SYSTEM DESIGN AND MAIN FEATURES

With the reference to Babcock and Wilcox 241 system, BKTF is designed to convert heat to mechanical or electrical energy by using indirect Rankine Cycle. The heat energy from reactor core was extracted to primary coolant water which is pressurized to be in subcooled condition. The primary coolant gives heat to the secondary one at a heat exchanger called steam generator (SG) to produce saturated steam to drive aturbine. The steam, after giving its thermodynamic energy to rotate the turbine will go through a condenser to condensate steam to liquid water and this return to SG again to repeat the cycle. With the base of classic modeling that keep water dimensionless properties such a Reynolds number in order of $10^5$[6] for turbulent flow in core, the volume of the core is determined about 45 liters out of 100 liters of total volume of reactor vessel for a steady flow rate of 3.5 $m^3$ /h (~1kg/s). In contrast to a real prototype where the down comer plays the role of a water shield to resist the thermal shock and a channel leading to lower plenum to homogenize the circulating flow, a can with volume about one tenth that of the core is assembled prior to the lower plenum. The core bypass also is considered by using a pipe that connects the return flow to the outlet**.** Core scaling factor is about one tenth for characteristic length of either plenum, with the reference to linear scaling[7,8].The main features of BKTF are shown in fig. 1 with the emphasized remarks to positions of thermocouples (T1-T8) with the resolution of 0.1 $^0$C, pressure sensors (P1-P3)-accuracy of 0.3% full scale output (FSO), and vortex flow sensors (SFM, FM)-accuracy of 0.5% FSO. These sensors are used to determine the thermodynamic properties of primary and secondary flows for the heat transfer calculation and process control.

Pressurizer is design to maintain system pressure with a large relative volume margin. Normally, at the cool conditions the liquid volume fraction $f_v$of pressurizer is set to be 33% and that at operation condition is $f_{max}$up to 65%. With the system volume about 140 (ρ=997 kg/$m^3$, the calculated volume of pressurizer is about 40 l.

The two sub components installed in the pressurizer are spray head and electrical heater that allows the pressurizer regulates the system pressure. When the system pressure increase 25% up to 10 bar of upper limit , the spray head allows the cold leg water enter the pressurizer to cool down the steam which make the system pressure decreased. In turn, when system pressure decrease the electrical heater is on then it make the liquid to vaporize. For the simplification of design and construction of steam generator, once through type with reference to Babcock and Wilcox one is selected[9]. Finally, condenser is designed to condense the steam flow with capacity of 40 kg/h. One shell countercurrent type is adopted into the design. There are three tube supports to help the tubes against the hydraulic vibrations and increase the contact time of the steam flow with surface of tube banks. ANSI ¾ inch stainless steel pipes were used to main coolant and steam flows. The system main components described above and other features are listed detail in table I.

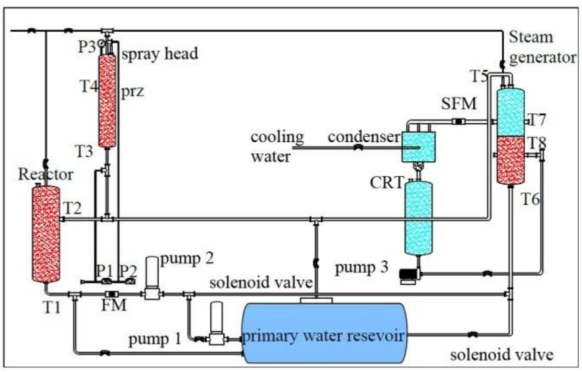


**Fig.1.** Schematic diagram of BKTF

**Table I:** Typical characteristics of thermal hydraulic testing facility (BKTF)

| Component/parameter | Values | Remarks |
|---|---|---|
| **General** | | |
| *Mass flow rate* | *1 kg/s* | *steady state* |
| *Primary coolant pressure* | *<10kg/cm$^2$* | *steady state* |
| *Primary coolant temperature* | *150$^0$C* | *steady state* |
| **Core** | | |
| *Core power* | *30 KW* | *maximum* |
| *Characteristic length* | *0.42 m* | *plenum length* |
| *Core flow area* | *0.14 m$^2$* | |
| *Reynolds number* | *~10$^5$* | *steady state* |
| **Pressurizer** | | |
| *Spray head flow* | *10 l/min* | *maximum* |
| *Heater power* | *2 KW* | *maximum* |
| *Safety valve* | *11 kg/cm$^2$* | *regulated* |
| **Steam Generator** | | |
| *Feed water inlet size* | *Φ21mm x3* | |
| *Steam outlet size* | *Φ 27mm x2* | |
| *Primary coolant inlet, outlet size* | *Φ27mm* | |
| *Water level* | *0.45-0.50m* | |
| *Total area of heat transfer* | *0.82-1.0 m$^2$* | |
| *Steam pressure* | *1,2, 3,4bar* | *saturated* |
| **Condenser** | | |
| *Max working pressure* | *9 kg/cm$^2$* | |
| *Steam inlet size* | *Φ 11mm x3* | |
| *Water well* | *Φ 80- $\phi$ 27* | *conical type* |
| *Sampling channel* | *Φ 11mm* | |
| *Vacuum level* | *< 1bar* | *ejector/vacuum pump* |
| *Total heat transfer area* | *1.0m$^2$* | |

## IV. PRELIMINARY TEST RESULTS

First, the cold test experiments were performed to check each main component solely and then are tests for system as whole. Referring to standards adopted to pressure tests, all main components of primary circuit are tested with water and compressed air with the minimum pressures are 15 kg/cm$^2$ and 10 kg/cm$^2$ respectively[10]. The test results show that all primary components are satisfied with TCVN 8366-2010 and ready to hot tests. The tests also cover the leakage test for tube bundles of steam generator and condenser to quickly detect the compressed gas leakage by using detergent solution. This work is normally useful to check complex structures of densely welded tubes.

The summation data can be seen in the table II and table III. For the system tests, the primary coolant was tested with pressure of 15 kg/cm$^2$ while the secondary one was tested with pressure of 8 kg/cm$^2$. The test for each system was repeated 3 times, each time last more than one hour. The test results show that all system can work under design pressure as seen in fig. 2

**Table II:** Pressure tests for primary main components

| Component | Water test | Gas test | N. test |
|---|---|---|---|
| *Reactor vessel* | *15(kg/cm$^2$)* | *10(kg/cm$^2$)* | *5* |
| *Pressurizer* | *15(kg/cm$^2$)* | *10(kg/cm$^2$)* | *5* |
| *SG(primary side)* | *15(kg/cm$^2$)* | *10(kg/cm$^2$)* | *5* |

**Table III:** Leakage tests for steam generator and condenser

| Component | Gas press | Observation | N. test |
|---|---|---|---|
| *Steam generator* | *6 to 10 (kg/cm$^2$)* | *No bubble* | *3* |
| *Condenser* | *6 (kg/cm$^2$)* | *No bubble* | *3* |

For the hot tests, electrical supply to the core heater is equivalent to 33% the maximum designed power. The core heater and primary coolant pump were powered on at the same time. During the heating up coolant in primary circuit, the feed water in secondary side is turned to keep constant level in the steam generator.

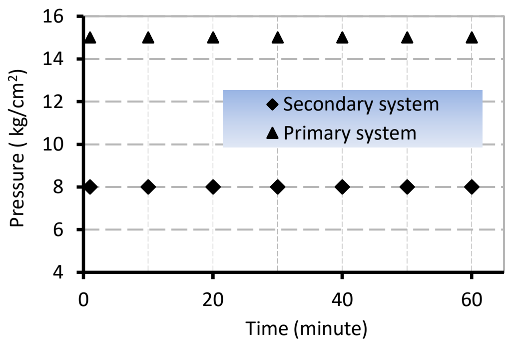


**Fig2.** Pressure test for primary and secondary system

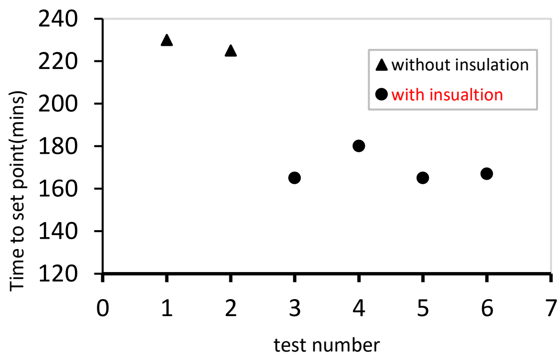


**Fig3.** Variation on time period to reach a set temperature

It is easily to see the effect of piping insulation by comparing the time to reach the temperature set point of 100 degree Celsius. The time period to reach the set point of primary coolant accountably decreased from about 225 minutes (without piping insulation) to 167 minutes (with piping insulation) with little change of outside temperature. It means that the pipe system with insulator (glass fiber) covered acts effectively to save the electrical energy by cutting heat loss about approximately 25%.  In the next figure, the collected data show the trends of temperature and pressure in primary coolant and secondary coolant during the process of leveling up to the operational point.

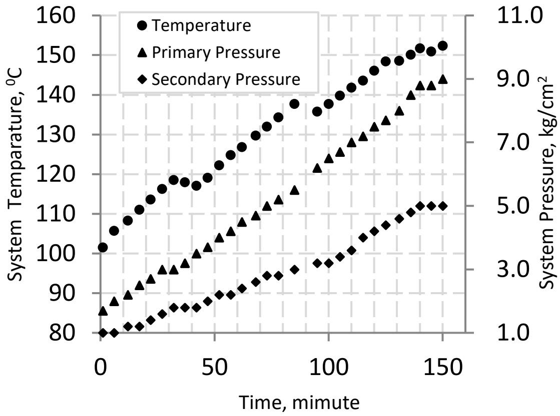


**Fig.4:** Time dependence of temperature andpressure during the leveling process

One special phenomenon was observed during leveling up the temperature, when the primary coolant reached 118.5 $^0$C and 3.0 kg/cm$^2$ (under saturated pressure), the circulate coolant temperature equaled to that at pressurizer. To avoid the inverse temperature gradient between coolant pressurizer and circulated coolant that make the control of system pressure in long term impossible or very difficult, the core power is turned off for a period of time about ten minutes

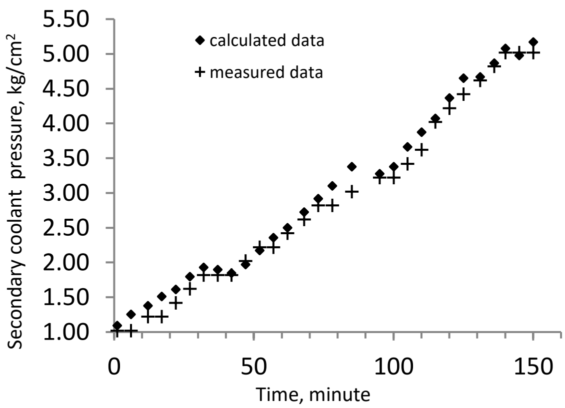


**Fig. 5:** Comparison between the measured and calculated secondary coolant pressure

The observation on temperature rising line also shows us its linear manner with a changing rate about 0.45$^0$C/min to avoid the excessive stress of material caused by thermal expansion. In an actual power reactor represented by Fuji, the temperature rise shows a similar trend with the same rising rate[11]. On the other hand, primary coolant pressure increases to the operational one in similar trend of linear in the same period of time. It's necessary to emphasize that pressure still continue when core power is turned off. This is caused by the heater in pressurizer which never stops working during the process to operation condition. In addition, when the steam phase in pressurizer did not reach the saturated state, it continued to rise to press the closed space that make the pressure keep going up. The lowest line in the figure shows the pressure in the steam generator-secondary side. It reflects the relation between temperature and pressure on a real boiler at saturated condition. The more analysis on this relation can be found at fig. 5.

The experimental data from fig. 5 show that the measured pressure always is lower calculated pressure. This can be explained as follows: when feed water in SG reaching temperature of 100 $^0$C and pressure 1.02kg/cm$^2$, it becomes boiling and steam phase is created in a closed plenum. While the temperature continue to rise with the rate about 0.4 $^0$C/min, the steam phase also continue to fill up this closed plenum that make the pressure rise but with a little delay in comparison with the rate of temperature rise.

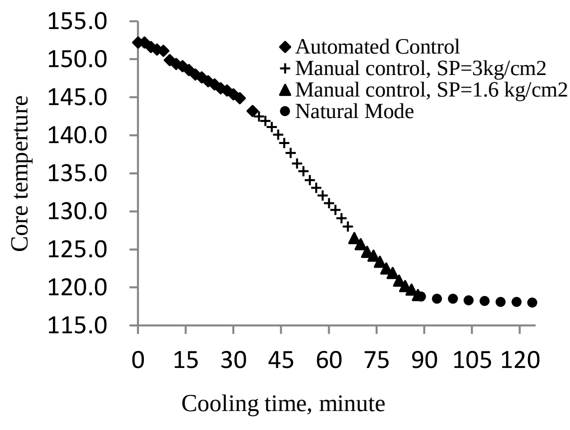


**Fig. 6**. Cooling process

The final presented data in fig. 6 show the cooling process which is separated into four segments. The cooling condition for the first segment can be described as follows: after reaching the operation set point, namely 152.2 $^0$C the cooling test was conducted by stop core heater while primary coolant pump still was in service. It's clear that when secondary is kept at high pressure in automated mode, the secondary coolant is also high that make temperature gradient between two sides of SG is small. Thisresults in a small rate of temperature decrease of 0.22 $^0$C/min. In the second segment, the different condition of cooling is created by leveling down pressure lower than saturated pressure in secondary side of SG that make secondary temperature decreased. In addition, some amount of steam consumed make the internal energy and temperature of secondary coolant in SG strongly decreased. From this moment, the temperature gradient between two sides of SG increases so that the heat transfer rate increase or cooling down process happen more rapidly at rate about 0.5 $^0$C/min. The third segment, although the pressure was once leveled down to 1.6 kg/cm$^2$ to increase heat transfer. However this pressure is very near saturated pressure so that the rate of temperature decrease in secondary side does not increase but little decrease. The last segment shows the natural cooling process when the primary coolant pump is turn off. Obviously, it's very difficult to cool down the core when the forced convection disappears and there no chance to get an observation of natural circulation in this circumstance when the core temperature nearly keep the same. The explanation for this low natural circulation may come from the high pressure loss of piping and structure system and this phenomenon will be investigated more in the near future.

## V. CONCLUSIONS

During start up process, the inverse gradient that makes the coolant boiling must be eliminated to maintain the subcooled condition in the core. The rate of linear temperature rise needs to keep reasonable to avoid excessive stress caused by thermal expansion. The cooling down process is heavily affected by forced convection. Like a real power system, the period of time for cooling down is shorter than that of start-up process.

## ACKNOWLEDGEMENT

This work has been performed at Laboratory of Nuclear Power Engineering Group at Hanoi University of Science and Technology under the grant 04/HĐ-ĐVPX 2012 from Ministry of Industry and Trade.

## NOMENCLATURE

$\dot{Q}$: thermal power (kW)
P: electrical power (kW)
$\dot{W}$: work rate (kW)
$\dot{m}_p$: flow rate of primary coolant (kg/s)
h: enthalpy (kJ/kg)
$\dot{q}$: heat transfer rate (kW)
q'': heat flux (W/m$^2$)
$\Delta T_x$: degree of superheat ($^0$C)
c: specific heat (J/kg.$^0$C)
$h_{fg}$: enthalpy of vaporization (kJ/kg)
$Pr_l$: Prantl number of liquid (-)
ρ: density (kg/m$^3$)
$h_{fg}$: enthalpy of vaporization (kJ/kg)
$Pr_l$: Prantl number of liquid (-)
ρ: density (kg/m$^3$)
μ: liquid viscosity (kg/m.s)
σ: surface tension (N/m)
$C_{sf}$ : surface coefficient (-)
g: gravitational acceleration (m/s$^2$)
Nu$^*$ modified Nusselt number ( )
Co: condensation number ( )
Re: Reynolds number ( )
*h:* head (m)
z: elevation (m)
*V:* flow velocity (m/s)

*p:* pressure ($N/m^2$)
*D*: pipe diameter (m)
*f:* friction loss ( )
*(*Superscript and subscript)
i: inlet; o: outlet; max: maximum; l: liquid, g; gas

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