PARAMETRIC STUDY OF MODERATOR HEAT EXCHANGER FOR CANDU 6 ADVANCED REACTOR

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ABSTRACT

PARAMETRIC STUDY OF MODERATOR HEAT EXCHANGER FOR CANDU 6 ADVANCED REACTOR. The passive moderator system for CANDU 6 advanced reactor require moderator heat exchanger with the small size and the low resistance coefficient of the shell-side. The study is to determine the required size of moderator heat exchanger, and to calculate the shell side of resistance coefficient have been done. Using computer code CATHENA, it is concluded that the moderator heat exchanger can be used at full power-normal operation condition, especially for the cases with 3600 to 8100 number of tube and 15.90 mm tube diameter. This study show that the proposed moderator heat exchanger have given satisfactory results.

Key words : moderator heat exchanger, passive moderator system, resistance coefficient of shell-side

ABSTRAK

STUDI PARAMETER PENUKAR PANAS UNTUK REAKTOR CANDU 6 YANG DISEMPURNAKAN. Sistem moderator pasif untuk reaktor CANDU 6 yang disempurnakan membutuhkan penukar panas dengan ukuran kecil dan mempunyai koefisien tahanan yang kecil pula pada sisi cangkangnya. Untuk itu telah dilakukan studi untuk menentukan ukuran penukar panas yang dibutuhkan dan menghitung koefisien tahanan pada sisi cangkangnya. Berdasarkan hasil analisis menggunakan program komputer CATHENA, dapat disimpulkan bahwa penukar panas sistem moderator ini dapat digunakan pada kondisi operasi normal, khususnya untuk kasus-kasus penukar panas dengan jumlah pipa 3600 sampai 8100 dan diameter pipa 15,9 mm. Studi ini menunjukkan bahwa penukar panas yang diusulkan telah memberikan hasil yang memuaskan.

Kata kunci : penukar panas moderator, sistem moderator pasif, koefisien tahanan pada sisi cangkang

I. INTRODUCTION

The current concept for the passive moderator system features single-phase forced circulation during normal operation, but can passively reject heat in the event of a loss-of-coolant accident (LOCA) [1,2,3,4]. A simplified schematic of the proposed system is shown in Figure 1. During normal operation, the flow in the moderator loop is driven by a combination of forced and free convection. In the event of an in-core loss-of-coolant accident, the heat transferred to the moderator is rejected through moderator heat exchanger that are cooled by naturally circulating light water from an overhead passive emergency water system (PEWS) tank. Hence, the pump power required for the proposed concept should be less than that of the conventional CANDU 6 moderator system.

In the proposed system, the moderator heat exchanger used for normal heat rejection are also employed for emergency heat rejection (the switch from a normal mode to an emergency mode of heat rejection is accomplished by means of valves in parallel and series arrangement, see figure 1).

With respect to the proposed concept for the passive moderator system, the specific objective of the planned analysis is to determine the required size of moderator heat exchanger, and to calculate the shell side of resistance coefficient. This study describes the cases, methodology, input information and boundary conditions to a planned analysis of the moderator heat exchanger for the passive moderator system of CANDU 6 advanced reactor.

II. BASIC DESIGN

The evaluation of performance or rating calculation of shell-and-tube heat exchanger has been analyzed in the following main ways [5] :

1. The process conditions are fully specified, just as for rating cases, but limits are usually imposed on maximum allowable pressure drop and/or flow velocity, both shell and tube side
2. As a consequence of (1), a geometric configuration must be found that will respect these limits while performing the desired heat duty most economically. This implies the combination of initial cost, operating and maintenance cost, ease of cleaning and many others.
3. It is then up to the heat exchanger designer (a) to select the basic construction elements such as shell type, tube size and layout, baffle type and baffle geometry, number of tube passes, and (b) to find the ultimate design dimensions of shell diameters and tube length that will result in complete thermal-hydraulic exchanger specification
4. The final step of converting the thermal-hydraulic solution into manufacturing specifications involves calculation of mechanical elements such as tube-sheets and flanges.

2.1. Boundary Conditions

The boundary conditions for the present study are discussed below :

a. The feed water employed for moderator cooling during normal operation is drawn from the condenser-pump, upstream of the first low-pressure pre-heater. The corresponding flow conditions are approximately 728 kg/s and 38°C.

b. For steady-state full-power operation conditions, the imposed moderator head load is 100 MW.

c. In the passive moderator system, the PEWS tank is open to the atmosphere at approximately 101 kPa

Schematic of the simplified moderator system is shown in figure 1.

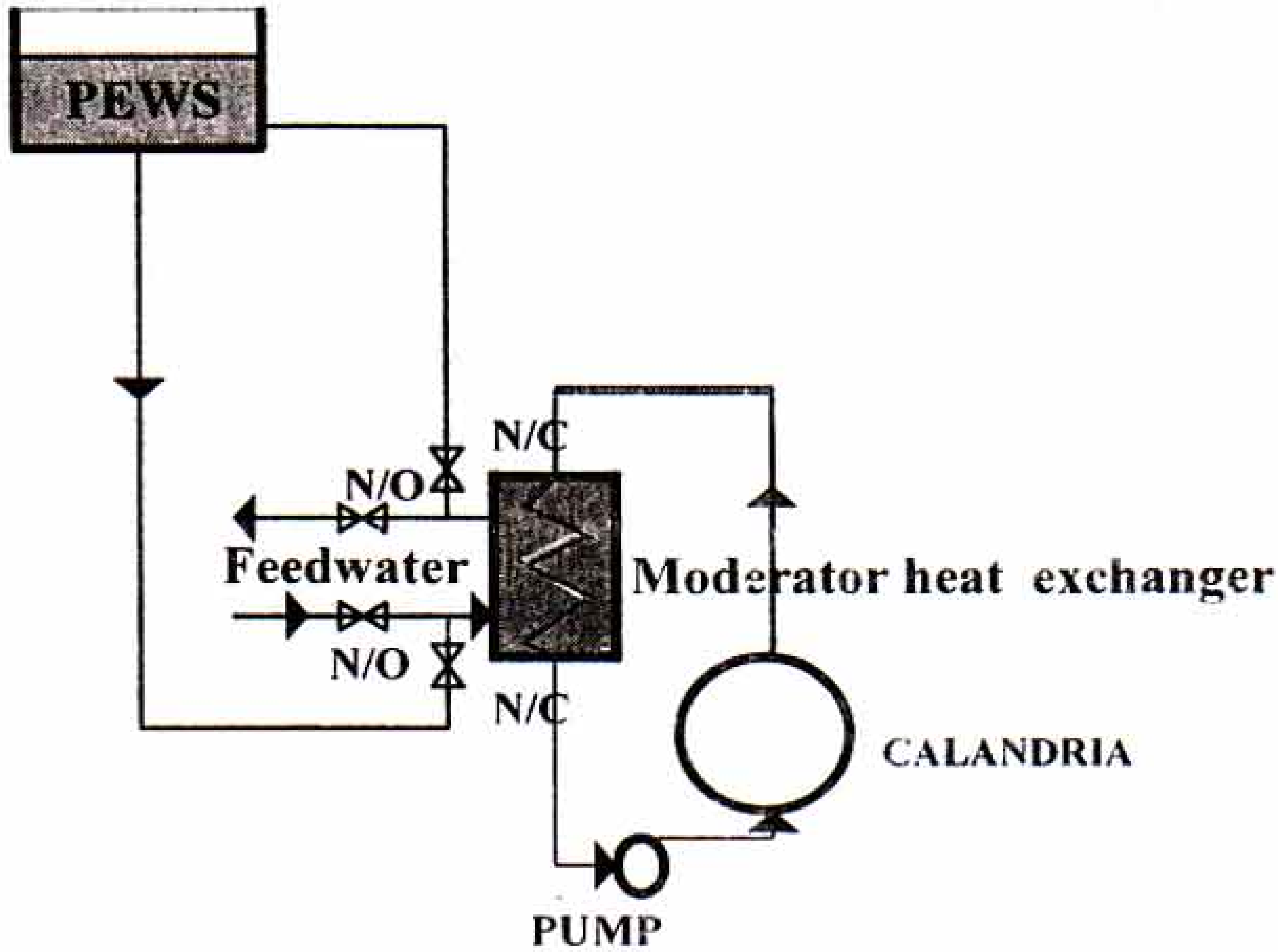

Figure 1. Simplified moderator system

2.2. Geometric Configurations

It is of utmost importance that the designer of shell-and-tube exchanger become acquainted with probable effects of elements of construction, such as tube diameter, tube layout pitch or tube layout pattern, tube bundle type, shell type, tube length, baffle type and spacing.

a. Tube outside diameter, D_t

Small tube diameters are preferred because of better heat transfer effectiveness, but cleaning considerations often limit the selection of tube diameter to ∞ 20 mm as a minimum

b. Tube wall thickness, L_{tw}

Tube wall thickness is determined according to pressure, temperature, material strength and possible corrosion [5,6].

c. Tube layout pitch, L_{tp}

A useful characteristic value is the tube pitch-to-tube diameter ratio (L_{tp}/D_t), which should be kept between the values of approximately 1.25 as a minimum and 1.5 as a maximum [7].

d. Tube layout characteristic angle, θ_{tp}

The 30° staggered layout permits the largest heat transfer surface within a given shell, but it produces the highest pressure drop for a given tube pitch. The 45° staggered layout have also a high effectiveness of pressure drop to heat transfer conversion, but permits only about 85 % of tubes within a given shell, as compared to a 30° layout. For a given pitch, the pressure drop is less than for a 30° layout. The 90° in-line layout has high effectiveness of pressure drop to heat transfer conversion in turbulent flow, especially if flow pressure drop is desired. External cleaning convenience is the same as for 45° layout.

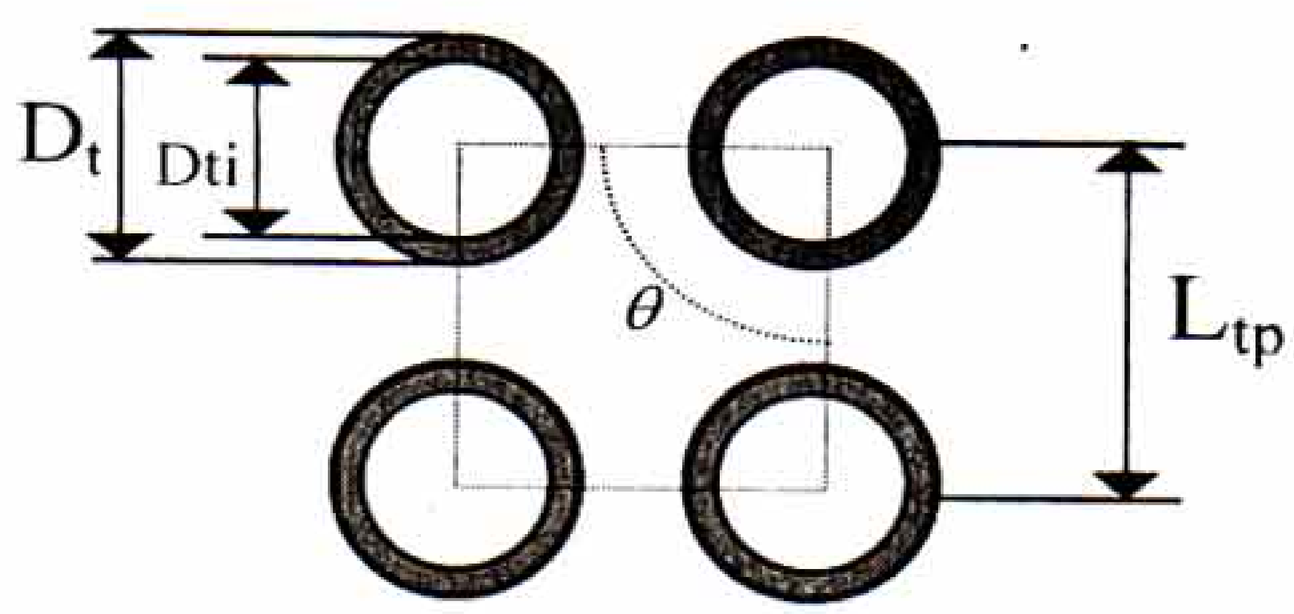

Figure 2. Tube layout geometry

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e. Tube bundle - circumscribed circle, D_{otl} and the diameter of the circle through the center of the outer tubes, D_{ctl}

Tube bundle - circumscribed circle and the diameter of the circle through the center of the outer tubes can be shown in figure 3.

f. Tube length definitions, L_{to}

In general, the longer the tube, the lowest the cost of the heat exchangers for a given surface. This is due to the resulting smaller shell diameter, thinner tube sheets and flanges. The limitations are accommodating shell - side flow areas with reasonable baffle spacing and practical design considerations.

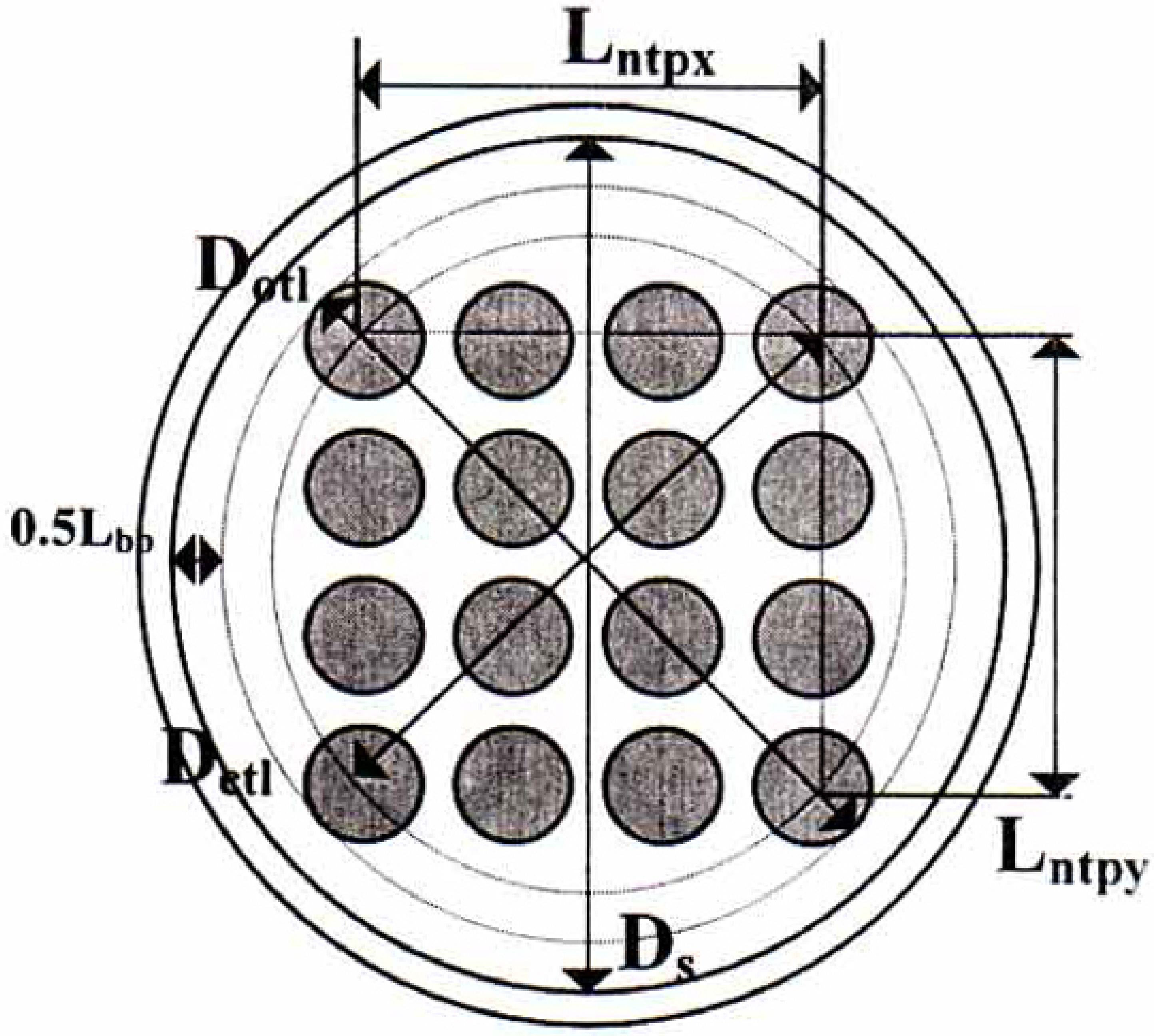

Figure 3. Basic geometry relations

Usual length - to - shell diameter ratio (L_{to}/D_s) range about 5 to 10 for best performance. We cannot plan the number of tube (n_t) without considerate the length - to - shell diameter ratio, because the heat exchanger will not operate in the best performance.

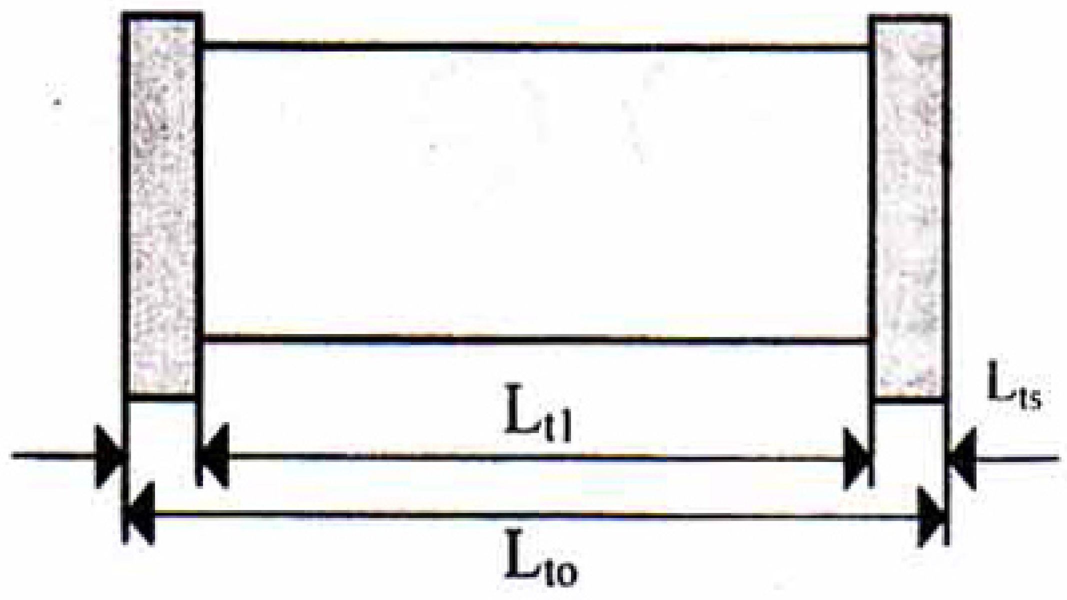

Figure 4. Tube length definition

g. Friction factor, f_i

By using the shell - side Reynolds number, the friction factor can calculate from the corresponding :

$$f_i = 0.391\left(\frac{1.33}{L_{tp}/D_t}\right)^{0.5023}\left(\mathrm{Re}_s\right)^{-0.148}$$

h. Cross flow area, S_m

The cross flow area at the shell centerline within one baffle spacing L_{bc} can calculate by using equation :

$$S_m = L_{bc}\left[L_{bb} + \frac{D_{ctl}}{L_{tp.eff}}\left(L_{tp} - D_t\right)\right]$$

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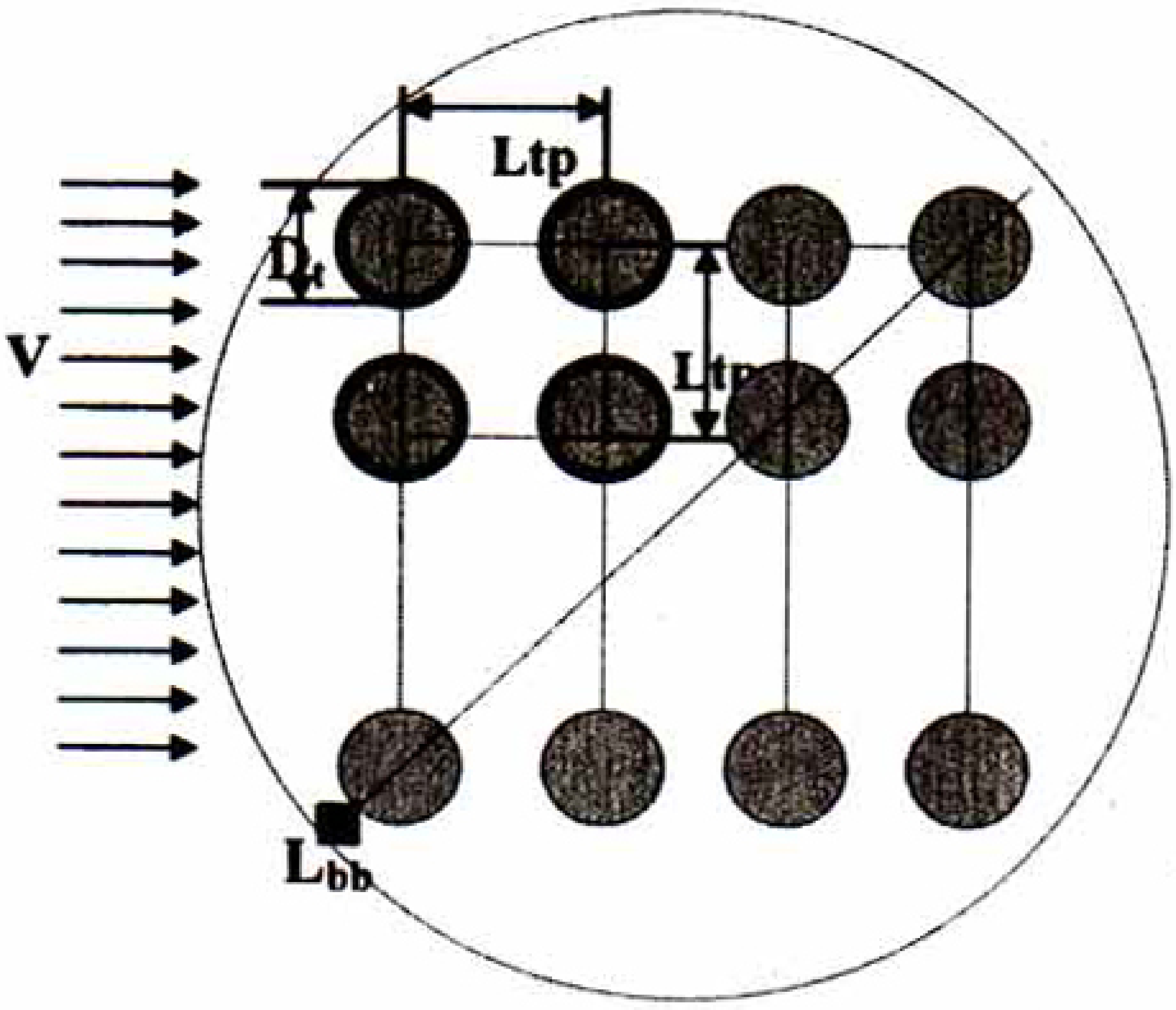

Figure 5. Schematic sketch of cross flow area

i. Central baffle spacing, L_{bc}

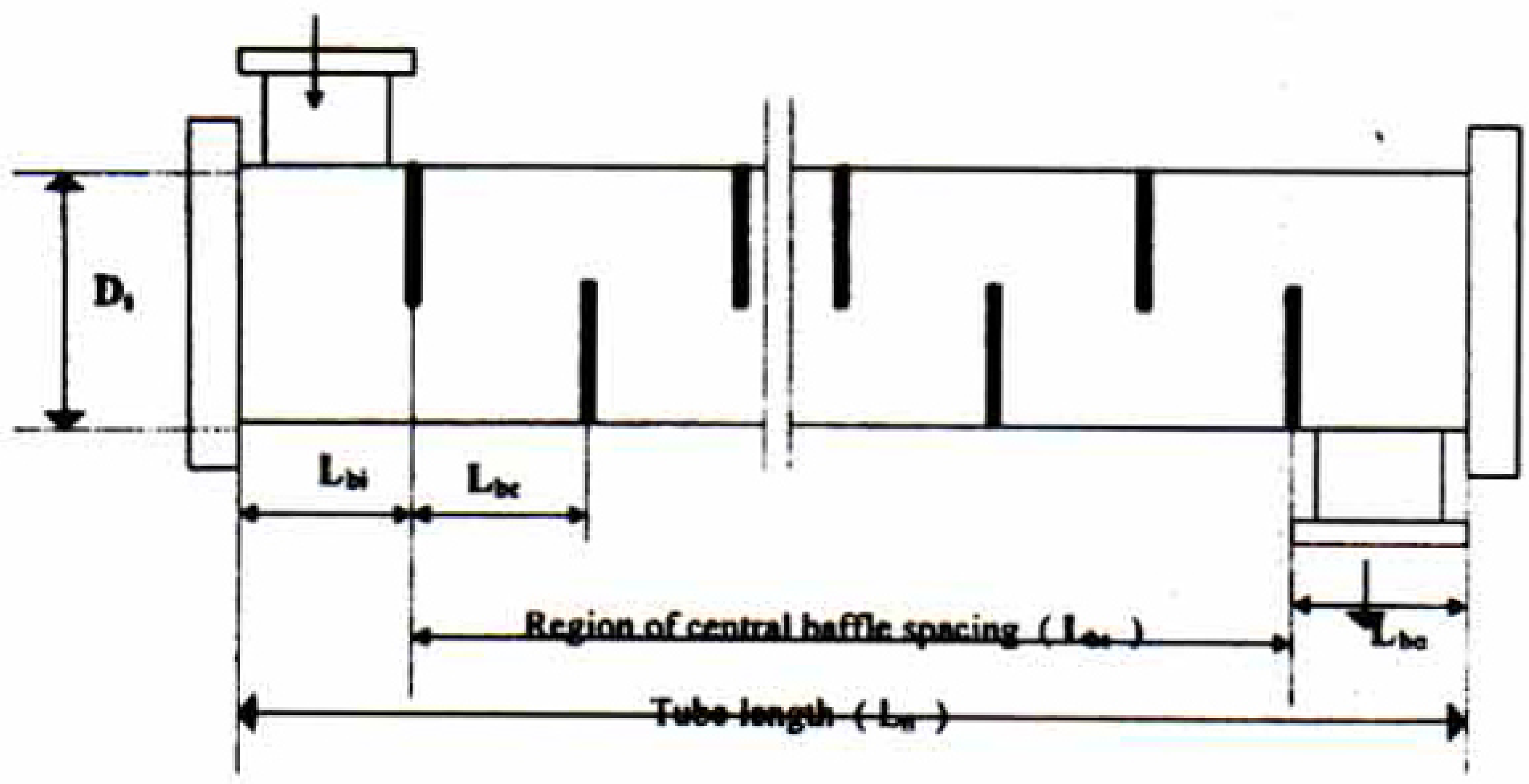

Figure 6. Schematic sketch of baffle distribution

j. Resistance coefficient of tube bank, K

The ideal tube bank pressure drop (ΔP_i) is used for a part of the flow between the baffle tips.

$$K = 4\, f_i\, N_{tcc}\, (N_b + 1)$$

where :

$$N_{tcc} = \frac{D_s}{L_{tp}}\left[1 - 2\left(\frac{B_c}{100}\right)\right]$$

III. RESULTS AND DISCUSSIONS

In the present study, the heat exchanger with 90^0 in-line layout is used for the moderator heat exchanger because the heat exchanger with 90^0 in-line layout has high effectiveness of pressure drop to heat transfer conversion in turbulent flow. Furthermore, the moderator heat exchanger with the low resistance coefficient in turbulent flow condition is needed in this simulation because the moderator heat exchanger is not only used at full-power normal operation, but also used at the emergency operating condition.

In the present study, the sizes of moderator heat exchanger were calculated for three cases.

Case 1 : Tube outside diameter = 19.05 mm
Length of tube depend on "length to-shell diameter ratio = 5"

Case 2 : Tube outside diameter = 19.05 mm
Length of tube = 9.0 m

Case 3 : Tube outside diameter = 15.90 mm
Length of tube = 9.0 m

Each of cases were considered fifth different size of moderator heat exchanger. The number of tubes in the square array ranges from 2500 (60x60) to 8100(90x90), and the corresponding heat transfer area ranges from about 1267 m^2 to 7371 m^2 for case 1, 1347 m^2 to 4363 m^2 for cases 2 and 1124 m^2 to 3461 m^2 for case 3. For comparison, the total heat transfer area of the moderator heat exchangers in typical CANDU 6 design is on the order of 2900 m^2

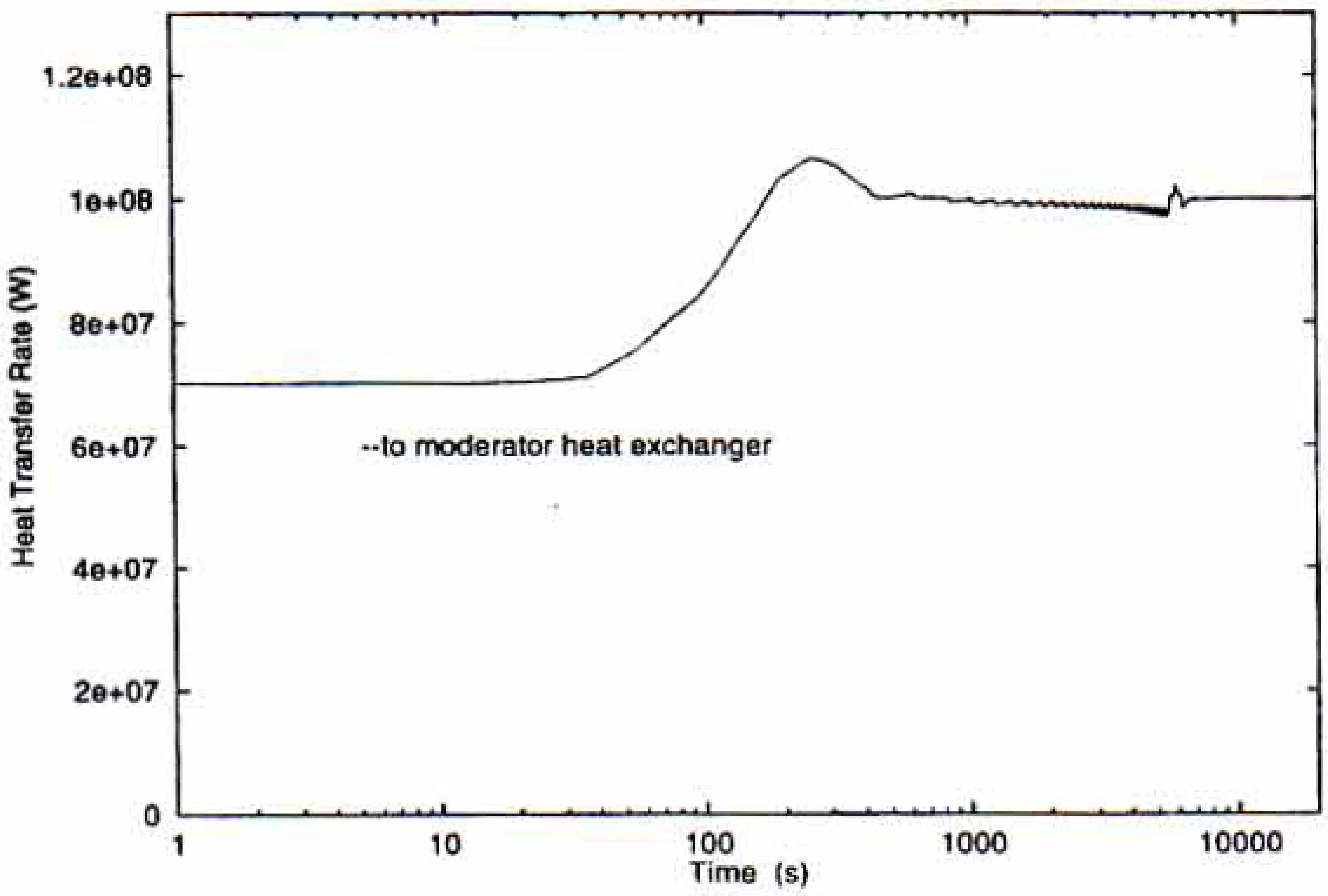

Figure 7. CATHENA prediction of heat transfer rate in moderator heat exchanger

based on Wolsong 2/3/4. [9]. Table 1,2 and 3 contains the computed values of the geometry configurations and the resistance coefficient for the moderator heat exchanger. As indicated in table 1,2 and 3, the magnitude of the resistance coefficient is directly related to the length of tube, especially for higher dimension of moderator heat exchanger. Moreover, the resistance coefficient of the moderator heat exchanger for the small pipe diameter is higher than the big pipe diameter (see table 2 and 3), but the absolute differences are not significant. Furthermore, if the cost of heat exchanger and the magnitude of heavy water hold up in the moderator heat exchanger are considered, the heat exchanger with the small pipe is more advantage than the big pipe. Based on this result, the moderator heat exchanger with the small pipe (15.90 mm diameter and 9 m length) is proposed to use in the passive moderator system. Compared to the previous design [2], the proposed moderator heat exchanger has a lower resistance coefficient of the shell-side moderator heat exchanger.

The proposed moderator heat exchanger have been tested in the CATHENA analysis of the passive moderator system [3,4]. Results of the CATHENA analysis show that the moderator heat exchanger with the number of tube 3600 to 8100 have given satisfactory results because the average rate of heat removal from the moderator equal to average rate of heat input to the moderator over a long - term period (see figure 7).

Table 1. The computed values of the geometry configurations and the resistance coefficient of case 1

Array	nt	Dt	Ltp	Ntt	Lntpx	Lntpy	Dctl
50 x 50	2500	19.05	23.8125	50	1166.8125	1166.8125	1650.1721
60 x 60	3600	19.05	23.8125	60	1404.9375	1404.9375	1986.9317
70 x 70	4900	19.05	23.8125	70	1643.0625	1643.0625	2323.6413
80 x 80	6400	19.05	23.8125	80	1881.1875	1881.1875	2660.4009
90 x 90	8100	19.05	23.8125	90	2119.3125	2119.3125	2997.1605

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Dotl	Lbb	Ds	Lto	Lts	Lti	Lbc	Ao
1669.172	25	1.6942	8.4709	0.1694	8.1320	0.3388	1267.4
2005.931	25	2.0309	10.1547	0.2031	9.7485	0.4062	2187.8
2342.691	25	2.3677	11.8385	0.2368	11.3649	0.4735	3471.7
2660.400	25	2.7045	13.5223	0.2704	12.9814	0.5409	5179.3
3016.210	25	3.0412	15.2061	0.3041	14.5978	0.6082	7371.4

Nb	Sm	m	Re	fi	Bc	Ntcc	K
24	0.1203	727.9	226022	0.066	20	43	283.8
24	0.1716	727.9	158479	0.069	20	52	358.8
24	0.2319	727.9	117243	0.072	20	60	432.0
24	0.3013	727.9	90234	0.075	20	69	517.5
24	0.3798	727.9	71587	0.078	20	77	600.6

Table 2. The computed values of the geometry configurations and the resistance coefficient of case 2

Array	nt	Dt	Ltp	Ntt	Lntpx	Lntpy	Dctl
50 x 50	2500	19.05	23.8125	50	1166.8125	1166.8125	1650.1721
60 x 60	3600	19.05	23.8125	60	1404.9375	1404.9375	1986.9317
70 x 70	4900	19.05	23.8125	70	1643.0625	1643.0625	2323.6413
80 x 80	6400	19.05	23.8125	80	1881.1875	1881.1875	2660.4009
90 x 90	8100	19.05	23.8125	90	2119.3125	2119.3125	2997.1605

Dotl	Lbb	Ds	Lto	Lts	Lti	Lbc	Ao
1669.172	25	1.6942	9.000	0.1694	8.6612	0.3388	1346.6
2005.931	25	2.0309	9.000	0.2031	8.5938	0.4062	1939.1
2342.691	25	2.3677	9.000	0.2368	8.5265	0.4735	2639.3
2660.400	25	2.7045	9.000	0.2704	8.4591	0.5409	3447.2
3016.210	25	3.0412	9.000	0.3041	8.3918	0.6082	4362.9

Nb	Sm	m	Re	fi	Bc	Ntcc	K
26	0.1203	727.9	226022	0.066	20	43	301.5
21	0.1716	727.9	158479	0.069	20	52	318.0
18	0.2319	727.9	117243	0.072	20	60	328.4
16	0.3013	727.9	90234	0.075	20	69	344.4
14	0.3798	727.9	71587	0.078	20	77	355.5

Table 3. The computed values of the geometry configurations and the resistance coefficient of case 3

Array	nt	Dt	Ltp	Ntt	Lntpx	Lntpy	Dctl
50 x 50	2500	15.90	19.8750	50	973.8750	973.8750	1377.2647
60 x 60	3600	15.90	19.8750	60	1172.2422	1172.2422	1658.3422
70 x 70	4900	15.90	19.8750	70	1371.3750	1371.3750	1939.4171
80 x 80	6400	15.90	19.8750	80	1570.1250	1570.1250	2220.4921
90 x 90	8100	15.90	19.8750	90	1768.8750	1768.8750	2501.5670

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Dotl	Lbb	Ds	Lto	Lts	Lti	Lbc	Ao
1393.1672	25	1.4182	9.000	0.1418	8.7164	0.2836	1123.9
1674.2422	25	1.6992	9.000	0.1699	8.6602	0.3398	1618.4
1955.3171	25	1.9803	9.000	0.1980	8.6039	0.3961	2202.9
2236.3921	25	2.2614	9.000	0.2261	8.5477	0.4523	2877.2
2517.4670	25	2.5425	9.000	0.2542	8.4915	0.5085	3641.5

Nb	Sm	m	Re	fi	Bc	Ntcc	K
31	0.0852	727.9	266296	0.064	20	43	349.3
25	0.1212	727.9	187218	0.067	20	52	369.1
22	0.1635	727.9	138773	0.070	20	60	381.8
19	0.2122	727.9	106962	0.073	20	69	400.9
17	0.2671	727.9	84956	0.076	20	77	414.3

A_o = Surface area
B_c = baffle cut (%)
FWHX = feed water heat exchanger (moderator heat exchanger)
D_s = shell inside diameter
D_t = tube outside diameter
D_{otl} = tube bundle - circumscribed circle
D_{ctl} = the diameter of the circle through the center of the outer tubes
f_i = shell side cross - flow friction factor
K = resistance coefficient
L_{bb} = tube bundle - to - shell by pass clearance or by pass channel diametrical gap
L_{bc} = central baffle spacing
L_{bi} = inlet baffle spacing
L_{bo} = outlet baffle spacing
L_{ti} = the distance between the inside of the tube sheet or length of summation of all baffle spacing
L_{to} = the nominal tube length
L_{ts} = tube sheet thickness