SCIENCE CHINA
Technological Sciences

June 2011 Vol.54 No.6: 1462–1468
doi: 10.1007/s11431-011-4297-0

Constructal design for a steam generator based on entransy dissipation extremum principle

XIAO QingHua, CHEN LinGen* & SUN FengRui

Postgraduate School, Naval University of Engineering, Wuhan 430033, China

Received October 6, 2010; accepted December 31, 2010; published online February 10, 2011

Steam generator is optimized by applying entransy dissipation extremum principle and constructal theory and adopting analytical method. The obtained results show that the optimal spacing between adjacent tubes, the mass flow rate of gas and the maximum entransy dissipation rate all depend on the dimensionless diameter of one tube, the dimensionless pressure difference number and the dimensionless length of flow channel of gas. Besides the three dimensionless groups, the optimal numbers of riser tubes and downcomer tubes and their summation all depend on the dimensionless height of one tube. The maximum entransy dissipation rate increases as the pressure difference that drives the gas flowing increases, and as the diameter of one tube and the length of flow channel both decrease. The mean heat flux in the heat transfer process of hot gas grows greatly, and the performance of the system is improved. Compared with the optimal construct with heat transfer rate maximization, the optimal construct with entransy dissipation rate maximization can improved the heat transfer effect of the steam generator more.

entransy dissipation extremum principle, constructal theory, steam generator, entransy dissipation rate, generalized thermodynamic optimization

Citation: Xiao Q H, Chen L G, Sun F R. Constructal design for a steam generator based on entransy dissipation extremum principle. Sci China Tech Sci, 2011, 54: 1462–1468, doi: 10.1007/s11431-011-4297-0

1 Introduction

In 1996, after the model building and mathematical analysis of the development of street network [1], Bejan put forward the constructal theory [2]. This theory can be expressed, for simplicity, as the structure derived from the optimal performance. Since constructal theory was applied to optimization problems involving heat conduction [3], constructal theory has been developing rapidly [4–16] and has provided new research impetus into heat transfer problems [17–31].

Steam generators are a major domain of technology development in the power generation industries. Researches on the structures of steam generators are much significant. Kim et al. [32] proposed to use constructal theory in the conceptual design of steam generators for large-scale commercial power plants and pointed out that constructal theory was ideally suited to this because it began the conceptual design with a clean slate, and suggested that the designers to recognize and consider all the possible and competing configurations. Based on the simplifying assumption that the steam generator consisted of just one downcomer tube and many riser tubes, features that resulted from constructal design were the tube diameters, the number of riser tubes, the water circulation rate, the rate of steam production, and how the flow architecture should change when the operating pressure and the size of the flow system changed. Based on ref. [32], Kim et al. [33] considered that the steam generator was free to have many downcomer tubes. Features that resulted from constructal design with heat transfer rate maximization were the numbers of downcomer and riser tubes and optimal spacing between adjacent tubes.

*Corresponding author (email: lgchenna@yahoo.com; lingenchen@hotmail.com)

However, constructal optimization of the steam generator does not reflect global heat transfer performance by maximizing heat transfer rate as in ref. [33]. Guo et al. [34] introduced definitively a new physical quantity--"entransy" ever called heat transfer potential capacity [35] and proposed the entransy dissipation extremum principle. The physical meaning of entransy was further expounded with researches into, for example, physical mechanisms of heat conduction and electro-thermal simulation experiments [36–38]. Many scholars [39–60] have shown great interest in and have also studied heat transfer optimizations based on entransy dissipation extremum principle. Some scholars combined the entransy dissipation extremum principle with constructal theory to optimize a series of heat transfer problems [61–71]. Based on ref. [33], by combining the entransy dissipation extremum principle with constructal theory the steam generator will be re-optimized, and analytical expression of performance vs. geometric configuration of the steam generator will be obtained.

2 Model of steam generator

The model of steam generator is rectangularly parallelepiped (*HLW*), which is traversed by *N* equidistant riser and downcomer tubes of diameter *D* and height *H*, as shown in Figure 1 [33]. The geometric constraints of the steam generator are the total volume

$$V = HLW, \tag{1}$$

and the volume fraction occupied by the tubes

$$\phi = \frac{\pi N D^2}{4LW}, \tag{2}$$

where *N* is the total number of riser and downcomer tubes. If one uses $V^{1/3}$ as the fixed scale of the entire architecture, then there are five dimensionless geometrical features, $(\tilde{H},\tilde{L},\tilde{W},\tilde{D},\tilde{S}) = (H,L,W,D,S)/V^{1/3}$ and eq. (1) can be non-dimensionalized as

$$\tilde{H}\tilde{L}\tilde{W} = 1. \tag{3}$$

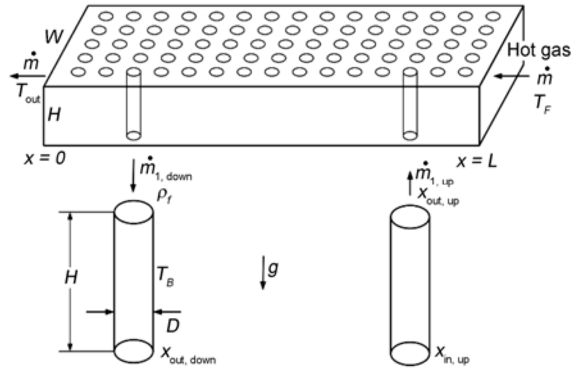

Figure 1 Model of steam generator [33].

In sum, the constructal design has four degrees of freedom ($\tilde{H}$, $\tilde{L}$, $\tilde{D}$ and $\tilde{S}$). Hot combustion gases are single phase with constant properties. The gases of mass flow rate $\dot{m}$, specific heat at constant pressure c_P and inlet temperature T_F flow from right to left. Steam flows in a vertical loop due to the fact that the riser tubes are close to the inlet of the gases are heated more intensely that the downcomer tubes close to the outlet of the gases. Because the temperature of the hot flue gas is not high enough to transform the water in the tubes into superheated steam at the outlet of each tube, the temperature of the water in tubes remains constant at the boiling temperature T_B.

3 Temperature distribution and entransy dissipation rate of the gas

Assuming that *N* is sufficiently large so that one can treat the gas temperature as a continuous function of position *x*. According to conversation of energy, enthalpy loss of the gas matches the transfer of heat to each tube at the located position, i.e. [33],

$$\frac{hA}{L}(T - T_B)\mathrm{d}x = \dot{m}c_P\mathrm{d}T, \tag{4}$$

where h is the heat transfer coefficient considered as a known constant or a function of the known mass flow rate $\dot{m}$. The boundary condition corresponds to eq. (4)

$$x = L,\ \ T = T_F. \tag{5}$$

By solving eq. (4) the gas temperature distribution in the steam generator is [33]

$$T = (T_F - T_B)\exp[\frac{hA}{\dot{m}c_P}\left(\frac{x}{L} - 1\right)] + T_B. \tag{6}$$

Entransy which is a new physical quantity reflecting heat transfer ability of an object was defined by Guo et al. [34]. In the heat transfer process the total entransy dissipation rate $\dot{E}_{vh\phi}$ is

$$\dot{E}_{vh\phi} = \int_v \dot{E}_{h\phi}\mathrm{d}v = \int_v \dot{\boldsymbol{q}} \cdot \nabla T\mathrm{d}v, \tag{7}$$

where ∇T is the temperature gradient. In this heat convection problem of this paper, the entransy dissipation rate is led by the heat exchange of the gas and the tube wall. In the position *x* of each tube the entransy dissipation rate is the product of the heat exchange and the temperature difference between the gas and tube wall. The entransy dissipation rate of the gas in the steam generator is

$$\dot{E}_{vh\phi} = \int_0^L (T - T_B)\dot{m}c_P\mathrm{d}T = \int_0^L (T - T_B)\dot{m}c_P\frac{\mathrm{d}T}{\mathrm{d}x}\mathrm{d}x. \tag{8}$$

The gas entransy dissipation rate can be obtained with eqs. (6) and (8).

$$\dot{E}_h = \int_0^L \dot{m}c_P (\Delta T)^2 \frac{hA}{\dot{m}c_P L} \exp\left[\frac{2hA}{\dot{m}c_P}\left(\frac{x}{L}-1\right)\right]\mathrm{d}x$$
$$= \frac{\dot{m}c_P}{2}(\Delta T)^2 [1-\exp(-2N_{tu})], \quad (9)$$

where $\Delta T = T_F - T_B$ is specified and N_{tu} is determined by $\dot{m}$, h and A, i.e.,

$$N_{tu} = \frac{hA}{\dot{m}c_P}. \quad (10)$$

Therefore, the mean heat flux with a fixed boundary temperature can be expressed as

$$\bar{Q} = \dot{E}_h / \Delta T. \quad (11)$$

The entransy dissipation extremum principle proposed by Guo et al. [34] is stated as follows: for a fixed boundary heat flux, the heat transfer process is optimized when the entransy dissipation is minimized (minimum temperature difference), while for a fixed boundary temperature, the heat transfer process is optimized when the entransy dissipation is maximized (maximum heat flux). Here, the temperature difference and heat flux both denote mean effect. Ref. [34] indicated that the irreversibility in the heat transfer process for the purpose of heating or cooling was measured by dissipation rate. The heat transfer problem not related to the transition between the heat and work belongs to the entransy dissipation extremum principle with a fixed boundary temperature. The higher the entransy dissipation rate is, the larger the mean heat flux will be and the better the heat transfer effect will be. That is, when the gas entransy dissipation rate is maximized, the mean heat flux in the heat transfer process will be maximized and the performance of the system will be the best.

4 Distribution of riser and downcomer tubes

According to eq. (6), regardless of the water direction and mass flow rate in the tubes, the gas temperature varies continuously, so does the heat transfer rate from the gas to the water in the tubes. The heat transfer rate into one tube located at any position can be expressed as [33]

$$\dot{m}_1(h_{\text{out}} - h_{\text{in}}) = \pi hDH(T - T_B), \quad (12)$$

where $\dot{m}_1$ is the water mass flow rate in the tube, and h_{in} and h_{out} are the water specific enthalpies of the inlet and outlet of the tubes. Assuming that the water received by every tube is saturated, one can obtain h_{in} and h_{out}, i.e.,

$$h_{\text{in}} = h_f,\ h_{\text{out}} = h_f + x_{\text{out}} h_{fg}, \quad (13)$$

where h_{fg} is the specific enthalpy of phase change, h_f is the specific enthalpy of the saturated water, and x_{out} is the quality of water-vapor mixture. From ref. [33] one can obtain

$$x_{\text{out}} = \frac{1}{\rho_f v_{fg}}\left(\frac{\rho_f}{\rho_{\text{out}}} - 1\right). \quad (14)$$

According to eqs. (6) and (12)–(14) one can obtain

$$\dot{m}_1 = \frac{\pi hDH\Delta T}{B}\exp\left[N_{tu}\left(\frac{x}{L}-1\right)\right], \quad (15)$$

where $B = h_{fg}(\rho_f/\rho_{\text{out}} - 1)/(\rho_f v_{fg})$. The change in flow direction of the water in the tubes is not considered in eq. (15). In fact, the mass flow rate of the water in riser tubes is positive, but in downcomer tubes is negative. The change in direction occurs at the location (x_c) obtained by the following equation:

$$\int_0^{x_c} -\dot{m}_1 \mathrm{d}x + \int_{x_c}^{L} \dot{m}_1 \mathrm{d}x = 0. \quad (16)$$

Eliminating $\dot{m}_1$ between eqs. (15) and (16), one can obtain

$$\frac{x_c}{L} = \frac{1}{N_{tu}}\ln\left(\frac{1+\mathrm{e}^{N_{tu}}}{2}\right). \quad (17)$$

The separated location between the riser tubes and downcomer tubes will be obtained when N_{tu} is fixed.

5 Geometry on the gas side

As shown in Figure 2, the geometry of the steam generator is represented by the total volume $V = HLW$, dimensions H, L, W, D and the spacing between adjacent tubes S.

Ref. [33] shows the heat transfer coefficient for a single cylinder washed by the fluid:

$$\frac{hD}{k_F} = 0.3 + \frac{0.62 Re_D^{1/2} Pr_F^{1/3}}{[1+(0.4/Pr_F)^{2/3}]^{1/4}}\left[1+\left(\frac{Re_D}{282000}\right)^{5/8}\right]^{4/5}, \quad (18)$$

where k_F and Pr_F are properties of the hot gas. When $10 < Re_D < 10^5$ eq. (18) is approximated adequately by [33]

$$\frac{hD}{k_F} \cong C Re_D^{1/2}, \quad (19)$$

where $Re_D = UD/\nu_F$ and C is dimensionless factor of order 1,

$$C = \frac{0.62 Pr_F^{1/3}}{[1+(0.4/Pr_F)^{2/3}]^{1/4}}\left[1+\left(\frac{Re_D}{282000}\right)^{5/8}\right]^{4/5}. \quad (20)$$

The pressure difference (ΔP_F) that drives the gas stream flowing is fixed. When the spacing between tubes is large,

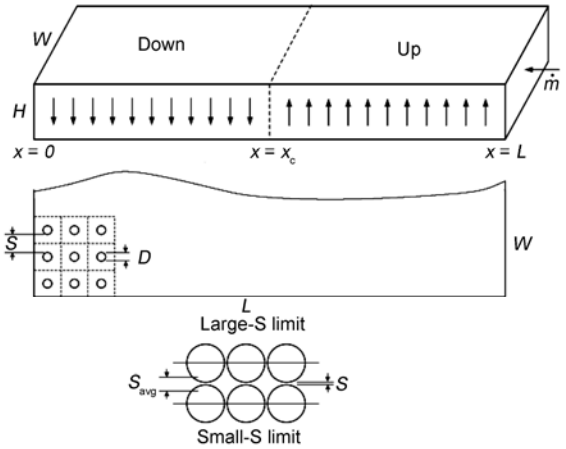

Figure 2 Geometric detail of steam generator [33].

the thermal contact between gas and tubes is poor ($N_{tu} << 1$), and the gas entransy dissipation rate $\dot{E}_h$ is small. When the spacing is small, the thermal contact is good, and N_{tu} is a large number, but $\dot{E}_h$ is also small. Between the two S extremes, there must be an optimal spacing ($S_{\rm opt}$) for which $\dot{E}_h$ is maximal ($\dot{E}_{h,\max}$). Based on the intersection of asymptotes method in refs. [14, 33], when H, L and D are fixed the optimal spacing corresponding to maximum $\dot{E}_h$ will be predicted below.

5.1 The large S limit

Because of the large spacings, the free velocity and mass flow rate of the hot gas are [33], respectively,

$$U_a \cong \frac{\dot{m}_a}{\rho_F HW}, \tag{21}$$

$$\dot{m}_a = \left(\frac{2\Delta P_F \rho_F H^2 W^2 S^2}{C_D LD} \right)^{1/2}, \tag{22}$$

where the gas density ρ_F is treated as a constant, C_D is the drag coefficient. From eq. (19) one can obtain

$$h_a \cong \frac{k_F C_a U_a^{1/2}}{\nu_F^{1/2} D^{1/2}}. \tag{23}$$

The total contact area between gas and tubes is [33]

$$A_a = \frac{\pi DV}{S^2}. \tag{24}$$

In the large S limit the number $N_{tu,a}$ approaches zero, and this means that the group $[1-\exp(-2N_{tu,a})]$ approaches $2N_{tu,a}$, and that eq. (9) can be expressed, for simplicity, as

$$\dot{E}_{h,a} = \dot{m}_a c_P (\Delta T)^2 N_{tu,a} = h_a A_a (\Delta T)^2. \tag{25}$$

From eqs. (10) and (21)–(25), one can obtain

$$\dot{E}_{h,a} = \frac{2^{1/4} \pi C_a \Delta P_F^{1/4} k_F D^{1/4} V (\Delta T)^2}{C_D^{1/4} \rho_F^{1/4} \nu_F^{1/2} S^{3/2} L^{1/4}}. \tag{26}$$

This shows that the gas entransy dissipation rate decreases monotonically as S increases in the large S limit. For this reason, the tendency of the gas entransy dissipation rate as S decreases in the small S limit will be considered below.

5.2 The small S limit

Because of the small spacings, the average spacing between adjacent tubes along the gas channel L can be expressed as [33]

$$S_{avg} = \sigma S, \tag{27}$$

where σ is a factor of order 1, but greater than 1. For the elemental channel represented by the volume $S_{avg} \times H \times L$, the number of such channels is W/D and the total mass flow rate of the gas is [33]

$$\dot{m}_b = \frac{\Delta P_F \sigma^3}{12\nu_F} \frac{S^3 V}{L^2 D}. \tag{28}$$

In the small S limit the number $N_{tu,b}$ becomes large, and this means that the group $\exp(-2N_{tu,a})$ approaches zero, and that eq. (9) can be expressed, for simplicity, as

$$\dot{E}_{h,b} = \frac{\dot{m}_b c_P}{2} (\Delta T)^2. \tag{29}$$

From eqs. (27)–(29) one can obtain

$$\dot{E}_{h,b} = \frac{\Delta P_F \sigma^3 (\Delta T)^2 V S^3}{24\nu_F L^2 D}. \tag{30}$$

This shows that the gas entransy dissipation rate decreases monotonically as S decreases in the small S limit.

5.3 The intersection of asymptotes

Based on the intersection of asymptotes method in refs. [14, 33], the optimal spacing S corresponding to maximum entransy dissipation rate is obtained by intersecting the two asymptotes (i.e., eqs. (26) and (30)).

$$\tilde{S}_{\rm opt} = \left[\frac{12 \times 2^{1/4} \pi C_a}{\sigma^3 C_D^{1/4} Pr_F^{1/4}} \right]^{2/9} 2^{2/9} Be^{-1/6} \tilde{L}^{7/18} \tilde{D}^{5/18}, \tag{31}$$

where $C_a \cong C_b$, $Pr_F = \dfrac{\nu_F}{a_F}$, the dimensionless pressure difference number $Be = \dfrac{\Delta P_F V^{2/3}}{a_F \mu_F}$ and the thermal diffusivity of the gas $a_F = \frac{k_F}{\rho_F c_F}$. The factor in square brackets in eq. (31) is approximately 1 and can be neglected, and thus, eq. (31) becomes

$$\tilde{S}_{\text{opt}} = 2^{2/9} Be^{-1/6} \tilde{L}^{7/18} \tilde{D}^{5/18}, \tag{32}$$

and the corresponding mass flow rate and maximum entransy dissipation rate are, respectively,

$$\dot{m}_{\text{opt}} = \left[\frac{\sigma^3}{12}\right] 2^{2/3} \frac{k_F}{c_P} V^{1/3} Be^{1/2} \tilde{L}^{-5/6} \tilde{D}^{-1/6}, \tag{33}$$

$$\dot{E}_{h,\max} = \left[\frac{\sigma^3}{12}\right] 2^{-1/3} k_F (\Delta T)^2 V^{1/3} Be^{1/2} \tilde{L}^{-5/6} \tilde{D}^{-1/6}. \tag{34}$$

If one regards the factors in square brackets in eqs. (33) and (34) as two numbers of order 1 [33], one can obtain

$$\dot{m}_{\text{opt}} \cong 2^{2/3} \frac{k_F}{c_P} V^{1/3} Be^{7/10} \tilde{D}^{1/2} \tilde{L}^{-11/10}, \tag{35}$$

$$\dot{E}_{h,\max} \cong 2^{-1/3} k_F (\Delta T)^2 Be^{2/5} \tilde{D}^{-1} \tilde{L}^{-6/5}. \tag{36}$$

The optimal number of the tubes in the steam generator is

$$\begin{aligned} n_{\text{opt}} &= \frac{L}{S}\frac{W}{S} = \frac{1}{\tilde{S}^2 \tilde{H}} \\ &\cong 2^{-4/9} Be^{1/3} \tilde{D}^{-5/9} \tilde{L}^{-7/9} \tilde{H}^{-1}. \end{aligned} \tag{37}$$

From eq. (36), one can see that the maximum entransy dissipation rate depends on two of the remaining free dimensions of the assembly, $\tilde{D}$ and $\tilde{L}$. The $\dot{E}_{h,\max}$ value increases as both $\tilde{D}$ and $\tilde{L}$ decrease. That is, thinner tubes and a shorter gas flow length are better for increasing the total entransy dissipation rate of gas, and the performance of the system can be improved. It is because that when both $\tilde{D}$ and $\tilde{L}$ decrease, the spacing between adjacent tubes becomes smaller, and thus, the thermal contact between gas and tubes becomes better and the performance of the system is improved.

6 Geometry on the steam side

The gas side and steam side are not uncoupled [33]. The location of the flow reversal depends on the group N_{tu} in eq. (17). $N_{tu,\text{opt}}$ led by optimal performance of the system is determined by

$$\begin{aligned} N_{tu,\text{opt}} &= \frac{\pi C_a}{Pr_F^{1/2}} \left[\frac{\sigma^3}{12}\right] \left[\frac{12 \times 2^{1/4} \pi C_a}{\sigma^3 C_D^{1/4} Pr_F^{1/4}}\right]^{-7/9} \\ &\times 2^{-7/9} Be^{-3/10} \tilde{L}^{9/10} \tilde{D}^{-3/2} \\ &\cong 2^{-7/9} Be^{1/12} \tilde{L}^{5/36} \tilde{D}^{1/36}, \end{aligned} \tag{38}$$

and thus, the location of the flow reversal ($x_{c,\text{opt}}$), the number of riser tubes (n_{up}) and the number of downcomer tubes (n_{down}) are, respectively,

$$\frac{x_{c,\text{opt}}}{L} = \frac{1}{N_{tu,\text{opt}}} \ln\left[\frac{1+\exp(N_{tu,\text{opt}})}{2}\right], \tag{39}$$

$$n_{\text{up}} = \left(1 - \frac{x_{c,\text{opt}}}{L}\right) n_{\text{opt}}, \tag{40}$$

$$n_{\text{down}} = \frac{x_{c,\text{opt}}}{L} n_{\text{opt}}. \tag{41}$$

According to eqs. (37)–(41), n_{up} and n_{down} are both the functions of Be, $\tilde{L}$, $\tilde{D}$ and $\tilde{H}$. Figure 3 shows the number of riser and downcomer tubes and their ratio ($n_{\text{down}}/n_{\text{up}}$) vs. $N_{tu,\text{opt}}$ characteristics. From the figure, one can see that the number of downcomer tubes is greater than the number of riser tubes, however, the ratio is $n_{\text{down}}/n_{\text{up}} \sim 1$. This means that the mass flow rate in one tube has the same scale in both riser and downcomer tubes. From the analyses of this section and last section, constructal design of the gas side determines the flow reversal position (or the optimal number of riser and downcomer tubes) and the relationships between the geometric features of the design for the steam generator (eqs. (32)–(41)).

Table 1 shows the optimal constructs of the steam generator based on entransy dissipation rate maximization (this paper) and heat transfer rate maximization (ref. [33]), respectively. The optimal constructs involve the optimal spacing, mass flow rate of the gas, the number of tubes, maximum entransy dissipation rate and maximum heat transfer rate. The optimal construct with heat transfer rate maximization does not reflect global heat transfer performance as in ref. [33]. However, the optimal construct with entransy dissipation rate maximization indicates the mean heat flux in the heat transfer process of the steam generator, and reflects the global heat transfer performance. Compared

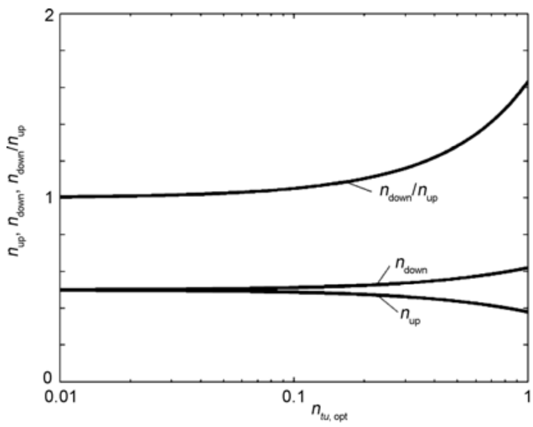

Figure 3 Numbers of riser and downcomer tubes.

Table 1 Optimal constructs of steam generator with different optimization objectives

Optimization objective	$\tilde{S}_{\text{opt}}$	n_{opt}	$\dot{m}_{\text{opt}}c_P/(k_F V^{1/3})$	$\dot{q}_{\max}/(k_F \Delta T V^{1/3})$	$\dot{E}_{h,\max}/[k_F V^{1/3}(\Delta T)^2]$
Entransy dissipation rate maximization (this paper)	$2^{\frac{2}{9}} Be^{-\frac{1}{6}} \tilde{D}^{\frac{5}{18}} \tilde{L}^{\frac{7}{18}}$	$2^{\frac{-4}{9}} Be^{\frac{1}{3}} \tilde{D}^{\frac{-5}{9}} \tilde{L}^{\frac{-7}{9}} \tilde{H}^{-1}$	$2^{\frac{2}{3}} Be^{\frac{1}{2}} \tilde{D}^{-\frac{1}{6}} \tilde{L}^{-\frac{5}{6}}$	$2^{\frac{-1}{3}} Be^{\frac{1}{2}} \tilde{D}^{-\frac{1}{6}} \tilde{L}^{-\frac{5}{6}}$	$2^{\frac{2}{3}} Be^{\frac{1}{3}} \tilde{D}^{-\frac{11}{9}} \tilde{L}^{-\frac{10}{9}}$
Heat transfer rate maximization (ref. [33])	$Be^{-\frac{1}{6}} \tilde{D}^{\frac{5}{18}} \tilde{L}^{\frac{7}{18}}$	$Be^{\frac{1}{3}} \tilde{D}^{-\frac{5}{9}} \tilde{L}^{-\frac{7}{9}} \tilde{H}^{-1}$	$Be^{\frac{1}{2}} \tilde{D}^{-\frac{1}{6}} \tilde{L}^{-\frac{5}{6}}$	$Be^{\frac{1}{2}} \tilde{D}^{-\frac{1}{6}} \tilde{L}^{-\frac{5}{6}}$	$Be^{\frac{1}{3}} \tilde{D}^{-\frac{11}{9}} \tilde{L}^{-\frac{10}{9}}$

with the dimensionless mean heat flux ($\bar{\tilde{Q}}_{h,t}$) with heat transfer rate maximization, the dimensionless mean heat flux ($\bar{\tilde{Q}}_h$) with entransy dissipation rate maximization increases by 58.7%. Therefore, the optimal construct based on the latter improves the global heat transfer performance of the steam generator obviously.

7 Conclusions

Steam generator is optimized by applying entransy dissipation extremum principle and constructal theory and by adopting analytical method. This paper similar to ref. [33] assumes the steam generator has a large number of tubes that the temperature distribution in the gas channel may be modelled as continuous, and also assumes that the tubes are all isothermal, and that the fluid in the tubes is single phase. According to these assumptions one can obtain the analytical expression of the mass flow rate in each tube along the gas channel, and the flow reversal that separates the riser tubes from downcomer tubes vs. the number of heat transfer units (N_{tu}) characteristic.

On the gas side, by adopting the method of intersecting the asymptotes, one can obtain the optimal spacing between adjacent tubes, the maximum entransy dissipation rate and optimal mass flow rate of gas with corresponding scaling relations, and the optimal number of steam tubes. The results show that the optimal spacing, mass flow rate and maximum entransy dissipation of gas all depend on the dimensionless pressure difference number of the gas, the dimensionless tube diameter, and the dimensionless length of the gas channel. Besides the three dimensionless groups, the optimal number of steam tubes depends on the length of the tubes. The maximum entransy dissipation rate increases as the pressure difference that drives the gas flowing increases, and as the diameter of one tube and the length of flow channel both decrease. The mean heat flux in the heat transfer process of hot gas grows greatly, i.e., the heat transferred to the water in the tubes of the steam generator becomes more, and the performance of the system is improved. Compared with the optimal construct with heat transfer rate maximization, the optimal construct with entransy dissipation rate maximization can greatly improve the heat transfer effect of the steam generator. In this paper, constructal design for the steam generator based on entransy dissipation extremum principle has led to some significant results, and fully enriched constructal theory and the theory of entransy dissipation.

This work was supported by the National Natural Science Foundation of China (Grant No. 10905093), the Program for New Century Excellent Talents in University of China (Grant No. NCET-04-1006) and the Foundation for the Author of National Excellent Doctoral Dissertation of China (Grant No. 200136).

1 Bejan A. Street network theory of organization in nature. J Adv Trans, 1996, 30: 85–107
2 Bejan A. Shape and Structure, from Engineering to Nature. Cambridge: Cambridge University Press, UK, 2000
3 Bejan A. Constructal-theory network of conducting paths for cooling a heat generating volume. Trans ASME, J Heat Transfer, 1997, 40: 799–816
4 Zhou S B, Chen L G, Sun F R. Constructal theory: a new direction for generalized thermodynamic optimization. J Thermal Sci Tech, 2004, 3: 283-292 (in Chinese)
5 Rosa R, Reis A, Miguel A. Proceedings of the Symposium Bejan's Constructal Theory of Shape and Structure. Evora: University of Evora, Portugal, 2004. 219
6 Bejan A, Lorente S. The Constructal Law (La Loi Constructale). Paris: L'Harmatan, 2005
7 Bejan A, Lorente S. Constructal multi-scale and multi-objective structures. Int J Energy Res, 2005, 29: 689–710
8 Bejan A, Lorente S, Miguel A, et al. Along with Constructal Theory. UNIL FGSE Workshop Series No. 1, Hernandez J and Cosinschi M, eds., University of Lausanne, Faculty of Geosciences and the Environment, Switzerland, 2006
9 Reis A. Constructal theory: From engineering to physics, and how flow systems develop shape and structure. Appl Mech Rev, 2006, 59: 269–282
10 Bejan A, Lorente S. Constructal theory of generation of configuration in nature and engineering. J Appl Phys, 2006, 100: 041301
11 Bejan A, Marden J H. Unifying constructal theory for scale effects in running, swimming and flying. J Experimental Biology, 2006, 209: 238–248
12 Bejan A, Merkx G (Editors). Constructal Theory of Social Dynamics. Ed. 1. New York: Springer; 2007
13 Bejan A, Lorente S. Design with Constructal Theory. New Jersey: Wiley, 2008
14 Bejan A, Marden J H. The constructal unification of biological and geophysical design. Phys Life Rev, 2009, 6: 85–102
15 Bejan A, Lorente S, Miguel A, et al. Constructal Human Dynamics, Security & Sustainability. Amsterdam: IOS Press, 2009
16 Bejan A, Lorente S. The constructal law of design and evolution in nature. Phil Trans R Soc B, 2010, 365: 1335–1347
17 Neagu M, Bejan A. Constructal-theory tree networks of "constant" thermal resistance. J Appl Phys, 1999, 86: 1136–1144
18 Neagu M, Bejan A. Three-dimensional tree constructs of 'constant' thermal resistance. J Appl Phys, 1999, 86: 7107–7115
19 Almogbel M, Bejan A. Constructal optimization of nonuniformly distributed tree-shaped flow structures for conduction. Int J Heat Mass Transfer, 2001, 44: 4185–4194
20 Ghodoossi S, Egrican N. Conductive cooling of triangular shaped electronics using constructal theory. Energy Convers Mgmt, 2004, 45: 811–828
21 Zhou S B, Chen L G, Sun F R. Constructal theory in designing photovoltaic cell. Prog Natural Sci, 2006, 16: 1500–1505 (in Chinese)
22 Wu W J, Chen L G, Sun F R. Improvement of tree-like network constructal method for heat conduction optimization. Sci China Ser E-Tech Sic, 2006, 49: 332–341

23 Zhou S B, Chen L G, Sun F R. Entropy generation minimization for "volume-point" conduction based on constructal theory. J Thermal Sci Tech, 2007, 6: 294–299 (in Chinese)
24 Wu W J, Chen L G, Sun F R. On the "area to point" flow problem based on constructal theory. Energy Convers Mgmt, 2007, 48: 101– 105
25 Zhou S B, Chen L G, Sun F R. Optimization of constructal volume-point conduction with variable cross-section conducting path. Energy Convers Mgmt, 2007, 48: 106–111
26 Zhou S B, Chen L G, Sun F R. Constructal optimization for solid-gas reactors based on triangular element. Sci China Ser E-Tech Sci, 2008, 51: 1554–1562
27 Karakas A, Camdali U, Tunc M. Constructal optimisation of heat generating volumes. Int J Exergy, 2009, 6: 637–654
28 Boichot R, Luo L, Fan Y. Tree-network structure generation for heat conduction by cellular automaton. Energy Convers Mgmt, 2009, 50: 376–386
29 Xie Z H, Chen L G, Sun F R. Constructal optimization of a vertical insulating wall based on a complex-objective combining heat flow and strength. Sci China: Tech Sci, 2010, 53: 2278–2290
30 Xie Z H, Chen L G, Sun F R. Geometry optimization of T-shaped cavities according to constructal theory. Math Comp Modell, 2010, 53: 1538–1546
31 Wei S H, Chen L G, Sun F R. The volume-point constructal optimization for discrete variable cross-section conducting path. Appl Energy, 2009, 86: 1111–1118
32 Kim Y S, Lorente S, Bejan A. Constructal steam generator architecture. Int J Heat Mass Transfer, 2009, 52: 2362–2369
33 Kim Y S, Lorente S, Bejan A. Steam generator structure: Continuous model and constructal design. Int J Energy Research, 2010, doi: 10.1002/er.1694
34 Guo Z Y, Zhu H Y, Liang X G. Entransy—A physical quantity describing heat transfer ability. Int J Heat Mass Transfer, 2007, 50: 2545–2556
35 Guo Z Y, Cheng X G, Xia Z. Least dissipation principle of heat transport potential capacity and its application in heat conduction optimization. Chin Sci Bull, 2003, 48: 406–410
36 Han G Z, Zhu H Y, Cheng X G, et al. Transfer similarity among heat conduction, elastic motion and electric conduction. J Eng Thermophys, 2005, 26: 1022–1024 (in Chinese)
37 Han G Z, Guo Z Y. Physical mechanism of heat conduction ability dissipation and its analytical expression. Proc CSEE, 2007, 27: 98–102 (in Chinese)
38 Zhu H Y, Chen J, Guo Z Y. Electricity and thermal analogous experimental study for entransy dissipation extreme principle. Prog Natural Sci, 2007, 17: 1692–1698 (in Chinese)
39 Cheng X T, Meng J A, Guo Z Y. Potential capacity dissipation minimization and entropy generation minimization in heat conduction optimization. J Eng Thermophys 2005, 26: 1034–1036 (in Chinese)
40 Han G Z, Guo Z Y. Two different thermal optimization objective functions: dissipation of heat transport potential capacity and entropy production. J Eng Thermophys, 2007, 27: 811–813 (in Chinese)
41 Liu H B, Guo Z Y, Meng J A. Analyses for entransy dissipation and heat resistance in heat exchangers. Prog Natural Sci, 2008, 18: 1186–1190 (in Chinese)
42 Chen Q, Ren J X. Generalized thermal resistance for convective heat transfer and its relation to entransy dissipation. Chin Sci Bull, 2008, 53: 3753–3761
43 Liu X B, Meng J A, Guo Z Y. Entropy generation extremum and entransy dissipation extremum for heat exchanger optimization. Chin Sci Bull, 2009, 54: 943–947
44 Liu X B, Guo Z Y. A novel method for heat exchanger analysis. Acta Physica Sinica, 2009, 58: 4766–4771 (in Chinese)
45 Wu J, Liang X G. Application of entransy dissipation extremum principle in radiative heat transfer optimization. Sci China Ser E-Tech Sci, 2008, 51: 1306–1314
46 Hu T, Min J, Song Y Z. Analysis of the effects of the heat of sorption on the process of heat transfer in moisture exchange across a membrane. Sci China Ser E-Tech Sci, 2008, 51: 2120–2127
47 Cheng X G, Xu X, Liang X G. Homogenization of temperature field and temperature gradient field. Sci China Ser E-Tech Sci, 2009, 52: 2937–2942
48 Wang S P, Chen Q L, Zhang B J, et al. A general theoretical principle for single-phase convective heat transfer enhancement. Sci China Ser E-Tech Sci, 2009, 52: 3521–3526
49 Chen Q, Ren J X, Guo Z Y. The extremum principle of mass entransy dissipation and its application to decontamination ventilation designs in space station cabins. Chin Sci Bull, 2009, 54: 2862–2870
50 Wang S P, Chen Q L, Zhang B J. An equation of entransy and its application. Chin Sci Bull, 2009, 54: 3572–3578
51 Xia S J, Chen L G, Sun F R. Optimization for entransy dissipation minimization in heat exchanger. Chin Sci Bull, 2009, 54: 3587–3595
52 Guo J F, Cheng L, Xu M T. Entransy dissipation number and its application to heat exchanger performance evaluation. Chin Sci Bull, 2009, 54: 2708–2713
53 Chen L, Chen Q, Li Z, et al. Optimization for a heat exchanger couple based on the minimum thermal resistance principle. Int J Heat Mass Transfer, 2009, 52: 4778–4784
54 Chen L, Chen Q, Li Z, et al. Moisture transfer resistance method for liquid desiccant dehumidification analysis and optimization. Chin Sci Bull, 2010, 55: 1445–1453
55 Chen Q, Wang M R, Pan N, et al. Optimization principles for convective heat transfer. Energy, 2009, 34: 1199–1206
56 Chen Q, Yang K, Wang M R, et al. A new approach to analysis and optimization of evaporative cooling system I: Theory. Energy, 2010, 35: 2448–2454
57 Guo J F, Xu M T, Cheng L. Principle of equipartition of entransy dissipation for heat exchanger design. Sci China Tech Sci, 2010, 53: 1309–1314
58 Xia S J, Chen L G, Sun F R. Entransy dissipation minimization for liquid-solid phase processes. Sci China Tech Sci, 2010, 53: 960–968
59 Xia S J, Chen L G, Sun F R. Optimal paths for minimizing entransy dissipation during heat transfer processes with generalized radiative heat transfer law. Appl Math Modell, 2010, 34: 2242–2255
60 Guo Z Y, Liu X B, Tao W Q, et al. Effectiveness–thermal resistance method for heat exchanger design and analysis, Int J Heat Mass Transfer, 2010, 53: 2877–2884
61 Wei S H, Chen L G, Sun F R. "Volume-point" heat conduction constructal optimization with entransy dissipation minimization objective based on rectangular element. Sci China Ser E-Tech Sci, 2008, 51: 1283–1295
62 Wei S H, Chen L G, Sun F R. Constructal entransy dissipation minimization for "volume-point" heat conduction based on triangular element. Thermal Sci, 2010, 14: 1075–1088
63 Wei S H, Chen L G, Sun F R. Constructal entransy dissipation minimization for "volume-point" heat conduction without the premise of optimized last-order construct. Int J Exergy, 2010, 7: 627–639
64 Wei S H, Chen L G, Sun F R. Constructal multidisciplinary optimization of electromagnet based on entransy dissipation minimization. Sci China Ser E-Tech Sci, 2009, 52: 2981–2989
65 Wei S H, Chen L G, Sun F R. Constructal complex-objective optimization of electromagnet based on magnetic induction and maximum temperature difference. Rev Mexi Fis, 2010, 56: 245–250
66 Wei S H, Chen L G, Sun F R. Constructal optimization of discrete and continuous variable cross-section conducting path based on entransy dissipation rate minimization. Sci China Tech Sci, 2010, 53: 1666–1677
67 Xie Z H, Chen L G, Sun F R. Constructal optimization for geometry of cavity by taking entransy dissipation minimization as objective. Sci China Ser E-Tech Sci, 2009, 52: 3504–3513
68 Xie Z H, Chen L G, Sun F R. Constructal optimization on T-shaped cavity based on entransy dissipation minimization. Chin Sci Bull, 2009, 54: 4418–4427
69 Xiao Q H, Chen L G, Sun F R. Constructal entransy dissipation rate minimization for "disc-to-point" heat conduction, Chin Sci Bull, 2011, 56, doi: 10.1360/972010-544
70 Xiao Q H, Chen L G, Sun F R. Constructal entransy dissipation rate and flow-resistance minimizations for cooling channels. Sci China Tech Sci, 2010, 53: 2458–2468
71 Xiao Q H, Chen L G, Sun F R. Constructal entransy dissipation rate minimization for umbrella-shaped assembly of cylindrical fins. Sci China Tech Sci, 2011, 54: 211–219