Automated Retrofit Design of Cooling-Water Systems

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An automated approach from the systematic investigation of interactions between cooling-tower performance and the cooling-water network design was developed for retrofit design of cooling-water systems. Introducing reuse of cooling water between different cooling duties enables cooling systems to be debottlenecked. The series arrangement of heat exchangers, however, results in a pressure drop increase because of the cooling water reuse. An optimization model for cooling-water networks developed considers pressure-drop constraints, complexity of networks, and efficient use of the cooling tower. A new node-superstructure model is suggested to consider the intensive property of pressure and the sequence of connections simultaneously. Physical insights are used to provide a good initialization for the resulting optimization problem. Issues of retrofit analysis are also addressed to give design guidelines for debottlenecking of cooling-water systems.

Introduction

Cooling of a chemical process is usually achieved by cooling water supplied by a cooling tower. The design and analysis of cooling-water systems has received much attention, as the improper design of the tower or poor control of water treatment can make the whole process inefficient or require a great deal of maintenance. Problems relating to process interaction, cooling-water treatment, the system upgrade for better cooling-tower performance, and the reduction of freshwater consumption have become major issues for cooling-water systems.

However, most researchers have focused on the individual components of cooling systems, not the system as a whole. Design and operation of cooling towers are important issues. But cooling-water systems should be designed and operated with consideration of system interactions between all the cooling system components, because the performance of the cooling-water system influences the performance of cooling-water-using systems, such as reactors, separators, the heat-exchanger network, and utility systems. Also, the return conditions from those cooling-water-using systems affect the performance of a cooling system. Since changes to operating conditions are often encountered for cooling-water-using systems, as well as the cooling system itself, this necessitates an understanding of the interactions between cooling towers and coolers.

When a cooling-water network needs to increase the heat load of an individual cooler, or a new heat exchanger needs to be introduced into an existing system, the cooling-water system can become bottlenecked. Although not all exchangers require cooling water with the supply temperature of the cooling system, heat-exchanger networks usually operate in a parallel configuration. Yet parallel configurations generally have the most difficulty accommodating increased cooling demand on the cooling system. When the cooling demand on cooling-water users is beyond the capacity of the existing cooling tower, a new cooling tower or supplementary cooling capacity needs to be added.

Investigation of the process interactions for water systems has been made in the area of water and wastewater minimization. During the last decade, various systematic methods have been suggested to save water resources and develop an environmental-friendly design for water systems. Wang and Smith (1994) introduced a design method for targeting maximum water reuse based on the graphical representation of water systems. The basic idea is that wastewater can be reused directly in other operations when water-using operations can accept the contamination level of previous operation(s). Later, Kuo and Smith (1997, 1998a,b) improved this design method-

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ology for total water-system design by attempting to take account of the interactions between water minimization, regeneration systems, and effluent treatment systems. While these methods provide physical insights and design features of water systems, energy and water implications were not investigated simultaneously.

In debottlenecking situations of cooling systems, a new design methodology has been developed by Kim and Smith (2001). This allows cooling towers to operate with an improved performance by rearranging some heat exchangers from parallel to series configurations rather than installing a new cooling capacity. While the interaction between the performance of the cooling tower and the design of cooling-water networks are explored systematically, the suggested design methodology has some drawbacks.

The design procedures developed for cooling-water networks are based on conceptual, not automated methods. It is impractical to consider other objectives (for example, pressure drop) simultaneously because this conceptual method deals with cooling-water systems by graphical representation of heat exchangers, and, therefore, targeting and design procedures are limited within a two-dimensional (2-D) analysis. Also, it is difficult to consider system constraints to be included (for example, forbidden/compulsory matches between units; complexity of networks), because the design method uses the concept of composite curves that represent cumulative characteristics of overall systems from the limiting data of individual coolers. The limitation of the conceptual method provides incentives to develop an automated method.

Although the minimization of any energy penalty and new investment for additional cooling can be achieved by the previous method, the effect of network configuration changes on cooling systems are overlooked. With debottlenecking design procedures, the design of cooling-water networks follows a reuse policy, and cooling-water flow rates for individual equipment are changed according to target conditions. In the retrofit case, pressure-drop constraints or circulation-pump performance often limits system modifications, because the performance of the pump is affected by the system flow sheet. But the previous targeting and design method does not consider pressure drop. Also, with the current design methodology, it is difficult to screen all alternatives for the network and select the best design configuration from among them.

The design problem of cooling-water systems is illustrated in Figure 1. For the cooling-tower side, the operating conditions for the tower are targeted, based on the thermal performance of the cooling tower and operating constraints. For the cooling-water-network side, the network should be designed to satisfy system changes and constraints. Also, conditions for both systems should be compatible with each other, and both system requirements should be met.

The objective of this study is to explore the retrofit design of cooling-water systems with minimum pressure drop by using an automated approach. The retrofit design of networks is necessary when cooling-water systems are bottlenecked. The problem is formulated as a mixed-integer nonlinear programming (MINLP) model with the new superstructure representation of cooling-water networks. The optimization model developed combines design requirements of cooling systems with pressure drop. In other words, the network with minimum pressure drop will be selected from among all of the possible configurations of network design that satisfy the target conditions for debottlenecking.

This article first reviews Kim and Smith's (2001) debottlenecking methods from cooling-water systems, and then pressure-drop correlations of heat exchanger and piping lines are explained briefly. The proposed MINLP model and optimal network design subject to pressure-drop constraint follows.

Review of Cooling-Water System Design

The design method developed by Kim and Smith (2001) will be reviewed first. In their research on cooling-water systems, the model of the cooling-water system and the design methodology of cooling-water networks was developed to account for the system interactions and process constraints.

Cooling-water system model

To investigate interactions within cooling-water systems, a cooling-water system model needs to include the cooling tower as well as the other system components. The model predicts the conditions of the exiting water and the air from the cooling system for a given design and operating conditions. From the predictions of the cooling-tower model, the heat removal of cooling towers can be expected to increase under conditions of high return temperature and low flow rate of cooling water. The mathematical model of the cooling tower was described in detail by Kim and Smith (2001).

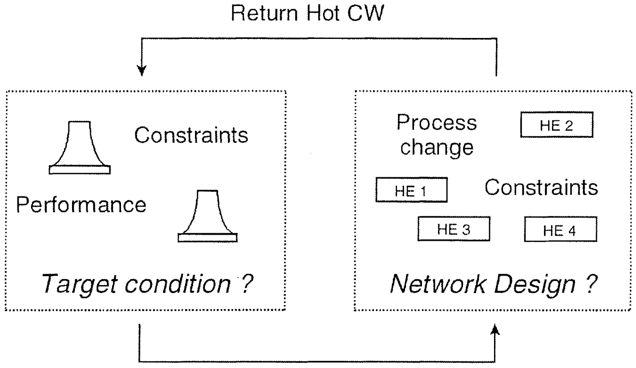


Figure 1. Design problem of cooling-water systems.

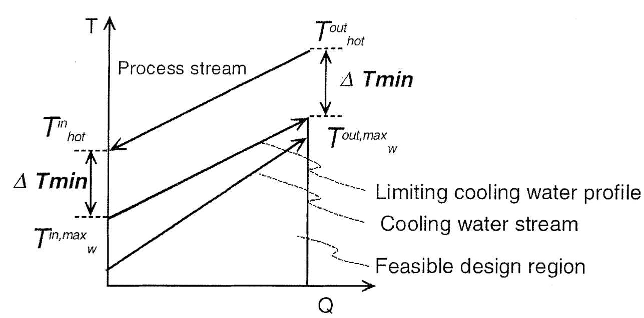


Figure 2. Limiting cooling-water profile.

Table 1. Hot-Process Stream Data: Example 1

Heat Exchanger	T_{hot}^{in} (°C)	T_{hot}^{out} (°C)	CP (kW/°C)
1	50	30	20
2	50	40	100
3	85	40	40
4	85	65	10

From the fact that all cooling duties do not require cooling water at the supply temperature from cooling systems, it is possible to change the cooling-water network from a parallel to a series design. Therefore, if the design configuration is converted from parallel to series arrangements where appropriate, the cooling tower can service a higher cooling demand for the cooling-water-using system. This overcomes the inflexibility of the traditional parallel arrangements of coolers.

Design methodology for cooling-water networks

A simple problem, Example 1 (Kim and Smith, 2001), will be used to explain the design methodology for cooling-water networks. A "limiting cooling water profile" is defined with the maximum inlet and outlet temperatures that are limited by the minimum temperature difference, corrosion, fouling, cooling water treatment, and so forth (Figure 2). In retrofit situations, the selection of limiting temperatures should be incorporated with the performance characteristics and/or constraints. The hot process stream data and limiting cooling water data for Example 1 are given in Tables 1 and 2. Since the limiting profile represents water and energy characteristics simultaneously, the cooling-water network can be manipulated on a common basis throughout all coolers.

A cooling-water composite curve is constructed by combining all individual profiles into a single curve within temperature intervals (Figure 3), and shows the overall limiting conditions of the whole network. The overall cooling-water flow rate and conditions for the network are determined by matching the composite curve with a straight line: the cooling-water supply line. The supply line touches the composite curve and creates a pinch point when targeting the minimum consumption of cooling water. With the adaptation of design procedures (Figure 4) given by Kuo and Smith (1998a), the cooling-water network with a pinch can be designed to achieve the target predicted by the supply line. The method often allows more than one network design to achieve the target. One design of a cooling-water network to achieve the minimum flow-rate target for Example 1 is shown in Figure 5.

For cooling-water networks, minimum overall flow rate of cooling water is not necessarily the optimum, because the

Table 2. Limiting Cooling-Water Data: Example 1

Heat Exchanger	$T_W^{in,max}$ (°C)	$T_W^{out,max}$ (°C)	CP (kW/°C)
1	20	40	20
2	30	40	100
3	30	75	40
4	55	75	10

Note: $\Delta T_{min} = 10°C$, cooling-water inlet temperature = 20°C.

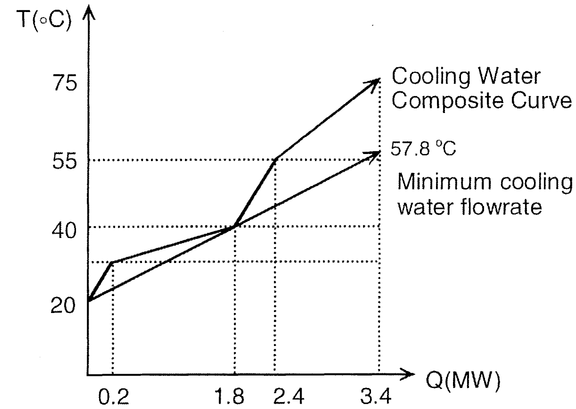


Figure 3. Targeting of minimum cooling-water flow rate.

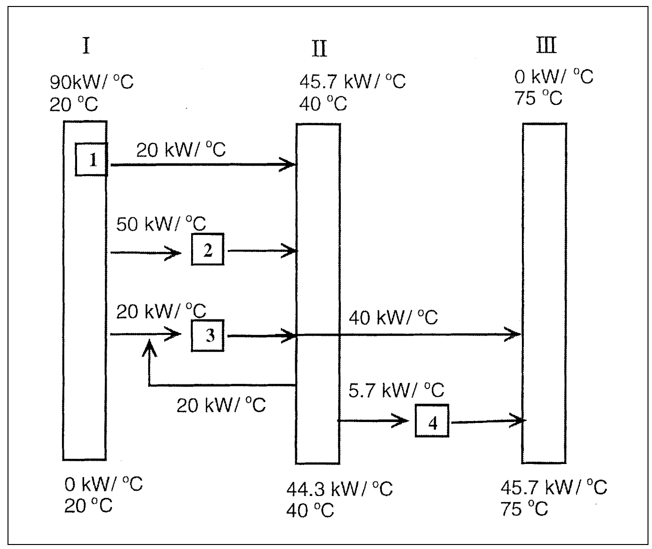


Figure 4. Cooling-water main method.

cooling-water supply line should be targeted based on consideration of the overall system. Also, cooling-water treatment problems might prevent the cooling-water system from operating beyond a specific return cooling-water temperature. For both cases, the cooling-water supply line does not correspond with minimum flow rate, and a pinch point is not created with the composite curve (Figure 6).

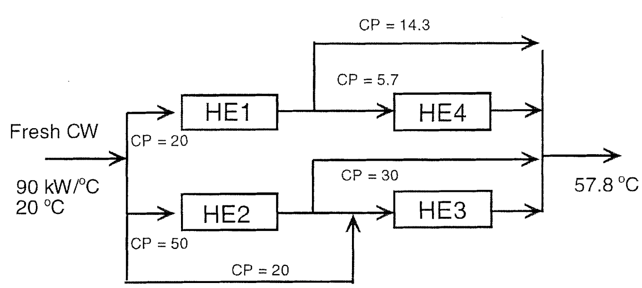


Figure 5. Cooling-water network design with maximum reuse.

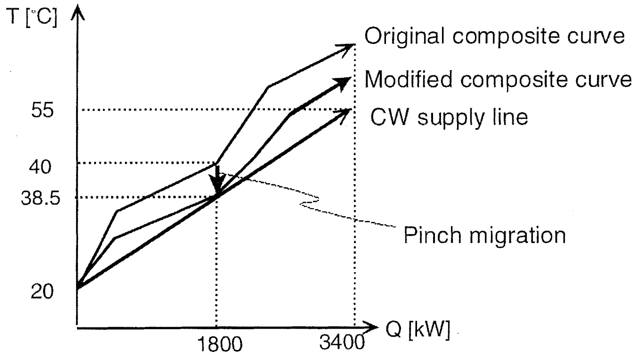


Figure 6. Pinch migration: Example 1.

To deal with network design without a pinch, the cooling-water composite curve is modified within the feasible region and the problem changed into a problem with a pinch. Combining "temperature shift" of individual limiting profiles, a "pinch migration" method is introduced to convert problems without a pinch into those with a pinch while maintaining the desired supply line. Suppose we constrain the target temperature to be 55°C for the cooling water for Example 1.

To avoid a temperature cross for the target line against the composite curve, the second kink point of the composite curve for this example is selected to migrate and a new pinch is calculated from a simple heat balance (Figure 6). Next a temperature shift is applied to individual exchangers.

The limiting cooling-water modifications have two stages. The first stage is to shift the temperature of the limiting water profiles according to the value of the temperature shift (exchangers 2 and 3). For cases when the modified profiles cross the supply line, the second stage increases the flow rate [CP] of the limiting water profile, without violating thermal design conditions (exchanger 1). After modification of the conditions for each individual exchanger, the modified cooling-water profiles are illustrated in Figure 7 and the new composite curve creates a pinch point with the desired cooling-water supply line. The cooling-water network design can now be carried out using the cooling-water mains method of Kuo and Smith (1998a). Thus, the pinch-migration and temperature-shift methods enable a design with any target temperature.

Debottlenecking of cooling-water systems

Debottlenecking procedures for cooling-water systems are explained by Example 2. A new heat exchanger is introduced into the base case, which causes the cooling-water system to become bottlenecked. The data for limiting cooling-water conditions and tower operations are given in Table 3. The performance of the cooling-water system in a parallel arrangement indicates that the cooling-water inlet temperature (T_{in} = 30.4°C) to the network is hotter than the desired inlet temperature (28.8°C). Also, the heat load of the network (15.6 MW) is greater than the heat removal of the tower (14.6 MW).

From the previous work, the best solution is obtained by modifying the cooling-water network when the cooling-water system is bottlenecked in terms of cooling capacity. First, the cooling-water composite curve is constructed from the limiting cooling-water data. Since the cooling-water network configuration can be changed between the maximum reuse supply line and the parallel design supply line (Figure 8), the feasible cooling-water supply line (line AB) can be identified. This represents the attainable outlet conditions by changing design configurations.

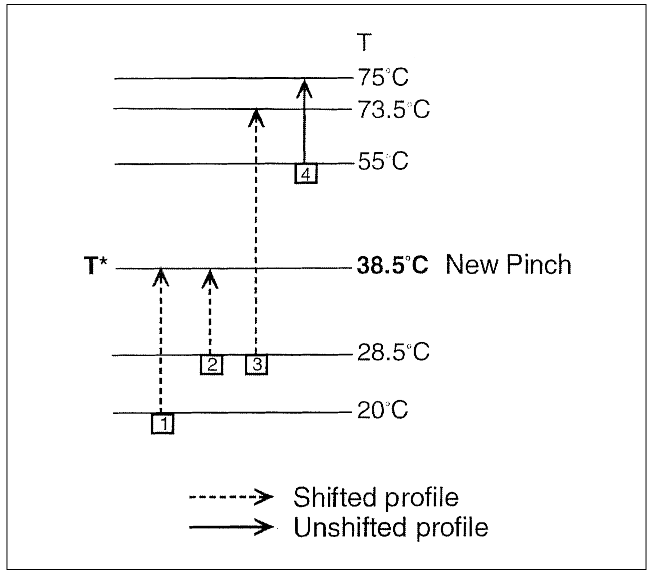


Figure 7. Temperature shift: Example 1.

Although the cooling-water supply line has the same heat load (15.6 MW), regardless of cooling-water network configuration, the heat removal of the cooling-water system is changed as inlet conditions to the cooling tower are changed. The heat removal of the cooling-water system increases as the design configuration changes from parallel to maximum reuse. The target conditions, which should satisfy the desired temperature for the cooling-water network (28.8°C) and cooling demand (15.6 MW), are found by changing the cooling-water supply conditions from A to B.

The target conditions for debottlenecking are found with the cooling-water system model and result in a CP of 725

Table 3. Cooling-Water System Data: Example 2

Heat Exchanger	$T_W^{in,max}$ (°C)	$T_W^{out,max}$ (°C)	CP (kW/°C)
Limiting cooling-water data			
1	28.8	37	200
2	30	37	635.5
3	30	52.7	488.9
4*	55	48	250
Cooling-system data			
Air flow rate = 732.24 t/h			
Air wet-bulb temperature = 23.9°C			
Air dry-bulb temperature = 29.4°C			
Blowdown = 7.6 t/h			
Makeup = 22.7 t/h at 10°C			
Evaporation = 15.1 t/h			
Cooling water inlet temperature to network = 28.8°C			
ΔT_{min} = 10°C			

*New heat exchanger.

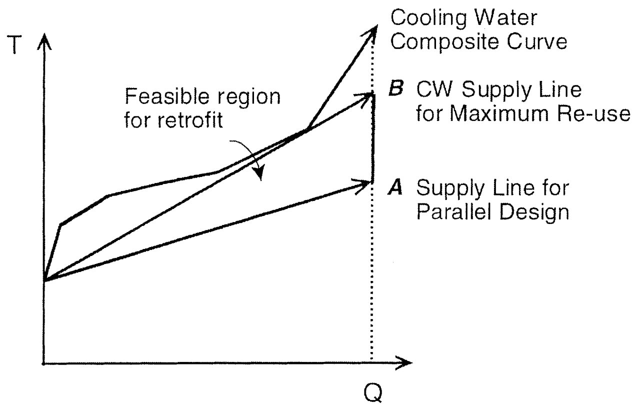


Figure 8. Identification of feasible cooling-water supply line.

kW/°C and a temperature of 50.3°C. The next stage is to design the cooling-water network with target conditions. The design for the debottlenecked cooling-water network is shown in Figure 9. The design procedure for debottlenecking cooling-water systems targets the cooling-tower conditions and then designs the cooling-water network for the new target conditions. The overall procedure is summarized as follows: (1) Find the feasible cooling-water supply line; (2) Target cooling-tower supply conditions with a cooling-system model; and (3) Design the cooling-water network with pinch migration.

Having summarized the previous work on cooling-water network design, let us now review pressure-drop correlations.

Pressure-Drop Correlations

Pressure drop of heat exchangers

Among the many types of heat exchangers, the 1-2 shell-and-tube-type heat exchanger is assumed in this study, because it is the design most commonly used. However, the method is readily adapted to other designs. Nie and Zhu (1999) proposed a pressure-drop correlation for 1-2 shell-and-tube-type heat exchangers. In their derivation of tube and shell sides pressure drop, the velocity is used as a bridge to correlate between the heat-transfer coefficient, pressure drop, and the heat-exchanger area. Although the shell-side pressure drop calculation is influenced by correction factors, these correlations give a sufficiently accurate estimation and are used for pressure-drop calculation of heat exchangers in this study.

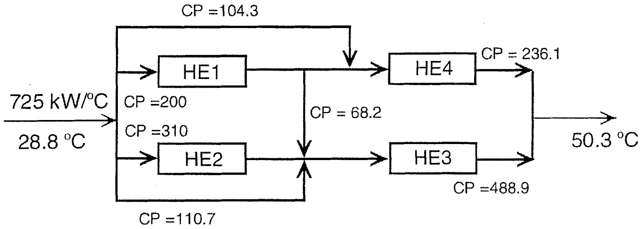


Figure 9. Debottlenecked design of cooling-water networks.

The pressure-drop correlations (Nie and Zhu, 1999) are expressed in terms of velocity as follows

For the tube-side

$$\Delta P_t = K_{t1} u_t^{2.8} + K_{t2} u_t^2 \tag{1}$$

For the shell-side

$$\Delta P_s = K_{s1} u_s^{1.83} + K_{s2} u_s^{2.83} + K_{s3} u_s^{1.83} + K_{s4} u_s^3 \tag{2}$$

It should be noted that all pressure-drop calculations in this article follow the notation of subtracting the pressure of the outlet condition from that of the inlet.

To apply the preceding correlations to cooling-water system design, the equations need to be represented in terms of flow rate, because the cooling-water system data are usually based on flow rate [*or* CP] rather than velocity.

The correlation for the tube side can be converted to an equation in terms of flow rate by using

$$V_t = \frac{\pi d_i^2}{4} \frac{N_t}{n_{tp}} u_t \tag{3}$$

By rearranging (Eq. 1), we have

$$\Delta P_t = N_{t1} V_t^{1.8} + N_{t2} V_t^2 \tag{4}$$

where

$$N_{t1} = \frac{1.115567}{\pi^{2.8}} \frac{\rho^{0.8} \mu^{0.8} n_{tp}^{2.8} A}{N_t^{2.8} d_o d_i^{4.8}} \tag{5}$$

$$N_{t2} = \frac{20}{\pi^2} \frac{n_{tp}^3 \rho}{N_t^2 d_i^4} \tag{6}$$

For the shell-side, the exact velocity value might not be calculated correctly, because each compartment of the shell side (that is, window, cross, and end zone) has a different and complicated geometry. In this work, correlating velocity with flow rate in the shell side follows Nie and Zhu's method (1999), which was used for retrofit design of heat-exchanger networks. Film heat-transfer coefficients between two cases with different flow rates can be calculated by using the following relationships

$$\frac{h_s}{h_{so}} = \left(\frac{V_s}{V_{so}} \right)^{0.6} \tag{7}$$

where

$$h_s = 0.69 F_h \frac{c_p^{1/3} k^{2/3} \rho^{0.52}}{\mu^{0.187} d_o^{0.48}} \cdot u_s^{0.52} \tag{8}$$

By rearranging Eq. 2 with Eq. 7, the shell-side pressure-drop correlation in terms of flow rate is given in Eq. 9. In this work, $F_b' = 0.8$ and $F_L' = 0.5$ are assumed and the correction factor $F_h = 0.64$ is used

$$\Delta P_s = N_{s1} V_s^{2.11154} + N_{s2} V_s^{3.26538} + N_{s3} V_s^{2.11154} + N_{s4} V_s^{3.46154} \tag{9}$$

where

$$N_{s1} = 1.947 \frac{F_b' D_s \mu^{0.17} \rho^{0.83}}{P_t d_o^{0.17}} \frac{u_{so}^{1.83}}{V_{so}^{2.11154}} \tag{10}$$

$$N_{s2} = \frac{2.596}{\pi^2} \frac{F_b' F_L' (p_t - d_o) A \mu^{0.17} \rho^{0.83}}{V_s P_t^2 d_o^{0.17}} \frac{u_{so}^{2.83}}{V_{so}^{3.26538}} \tag{11}$$

$$N_{s3} = -1.296 \frac{F_b' F_L' D_s \mu^{0.17} \rho^{0.83}}{P_t d_o^{0.17}} \frac{u_{so}^{1.83}}{V_{so}^{2.11154}} \tag{12}$$

$$N_{s4} = \frac{F_L'}{\pi^2} \left(\frac{1}{D_s} + \frac{0.075}{p_t} \right) \frac{A \rho (p_t - d_o)}{V_s P_t d_o} \frac{u_{so}^3}{V_{so}^{3.46154}} \tag{13}$$

Pressure drop of pipelines

The pressure drop in circular pipes is generally defined by

$$\Delta P_p = 4 f_o \left(\frac{\rho u_p^2}{2} \right) \left(\frac{L}{D_i} \right) \tag{14}$$

The following equation (Hewitt et al., 1994) is used to estimate the fanning friction factor

$$f_o = \frac{0.046}{Re^{0.2}} \tag{15}$$

By substituting Eq. 15 into Eq. 14, the pressure drop for the pipeline is given as

$$\Delta P_p = N_p^{\mathrm{EX}} V_p^{1.8} \tag{16}$$

where

$$N_p^{\mathrm{EX}} = \frac{1.11557}{\pi^{1.8}} \frac{\rho^{0.8} \mu^{0.2} L}{D_i^{4.8}} \tag{17}$$

Because cooling water might be reused between heat exchangers within the suggested debottlenecking design method, new pipes are introduced into the current network. Therefore, for retrofit cases, the calculation of the pressure drop is categorized into two classes: existing pipes and new pipes to be installed.

For new pipe installation, the pipe sizing is an important factor, and the pipe size should be incorporated into the pressure-drop calculation of pipe networks. A small pipe diameter yields a high velocity for the pipe and a high pressure drop. A large pipe diameter is favorable from the viewpoint of pressure drop, but results in high capital costs. To consider the economic trade-off of pipe installation, the optimal pipe size suggested by Peters and Timmerhaus (1991) is used in this study and given by

$$D_i^{\mathrm{opt}} = 0.343525 V_p^{0.45} \rho^{0.13} \quad (D_i \geq 0.0254 \text{ m}, \ Re \geq 2{,}100) \tag{18}$$

By substituting Eqs. 15 and 18 into Eq. 14, the pressure drop for the pipeline is given as

$$\Delta P_p = N_p^{NW} \frac{1}{V_p^{0.36}} \tag{19}$$

where

$$N_P^{NW} = \frac{188.318}{\pi^{1.8}} \rho^{0.176} \mu^{0.2} L \tag{20}$$

Equation 19 represents the pressure drop for new pipes when the pipe diameter is constructed at its optimal size. For existing pipes, Eq. 16 is used for the calculation of pressure drop with a fixed pipe diameter.

Mathematical Model of Optimization

In this article, we shall develop a mathematical optimization framework for cooling-water networks. For retrofit design of cooling systems, capital expenditure, such as the new cooling equipment and new pipe installations, can be expected. For basic design, capital-cost issues would be important decision factors in addition to operating costs. However, the current design methodology focuses on retrofit design and is aimed at exploring possibilities for the more efficient use of existing cooling towers, so as not to invest in an additional cooling tower or supplemental cooling capacity (such as air heat exchangers).

As discussed in the previous section, the debottlenecking design for cooling-water systems can be identified from targeting the cooling-water supply conditions for the tower by changing the cooling-water network design. As shown in Figure 10, the feasible region, which represents the possible network design area between maximum reuse and parallel supply line, can be divided into two design regions. Between the maximum reuse and target supply line (design region I), no additional cooling-tower capacity is needed for cooling-water systems, and the cooling tower can manage the increased

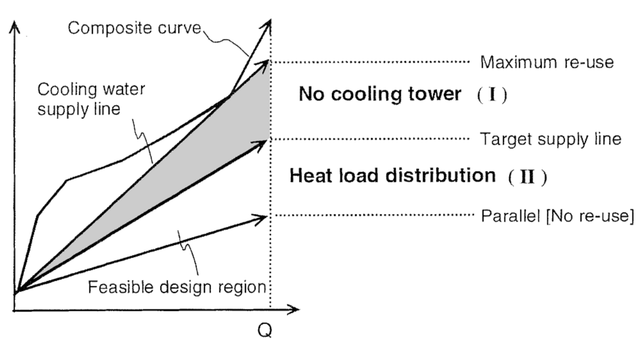


Figure 10. Design region for debottlenecking.

cooling load. However, in region II design, for the area between the target supply line and the parallel reuse line, changing the cooling-water network design cannot satisfy the increased cooling demand. Therefore, some of the heat load should be distributed with the aid of other design options. These include the introduction of air heat exchangers, change of blowdown extraction from cold to hot, a new cooling tower, upgrading of the cooling-tower components, and increasing the air flow rate to the tower. Insufficient cooling-water supply with low quality can result in a large profit loss and a significant reduction in productivity for the overall processes. Because the cooling-water network can change in design regions I and II, the overall outlet conditions of the networks can be varied between the maximum reuse and parallel supply lines.

Low quality of cooling-water supply can also be expected due to seasonal variations in air conditions that cause the cooling-tower performance to fluctuate. These problems can be solved by obtaining free capacity of the tower with the introduction of cooling water reuse and/or by implementing heat-load distribution options (Kim and Smith, 2002).

Among the several options for heat-load distribution, the installation of a new cooling tower is the most common solution in practice. Although the introduction of air heat exchangers is often carried out in practice, it has some disadvantages. Heat transfer for dry cooling is limited by dry-bulb temperature, while for wet-cooling towers it is limited by wet-bulb temperature. Also, a large heat-exchange area is required for air coolers, and, hence, results in a high capital cost, as the heat capacity of air is low. Upgrading of the cooling towers is not a desirable option, because the whole process needs to be shut down and there are limitations for upgrading. The strategic use of blowdown from the hot side, rather than the cold side, might cause environmental problems from high effluent temperatures. Increasing air flow rate is not a long-term solution and the effect is strongly dependent on ambient air conditions. Therefore, the introduction of an additional tower is a long-term and robust design solution and is considered in this study for economic evaluation when cooling systems are bottlenecked.

The economic trade-off is analyzed in Figure 11. A new cooling tower is introduced when the supply line is located in design region II, while a new tower is not required in design region I. Kim et al. (2001) exploited the optimization of cooling-system design. From their evaluation, the cooling-tower capital and operating costs can be represented as shown in Figure 11. In general, the capital cost is a function of range (temperature difference between cooling-water inlet and outlet temperatures of the tower), approach (temperature difference between cooling-water outlet temperature from the tower and wet-bulb temperature), flow rate, and wet-bulb temperature. The approach and wet-bulb temperature become fixed for a debottlenecking situation. Because the flow rate has more influence on capital cost than range, the capital cost for a new tower increases as the flow rate is increased.

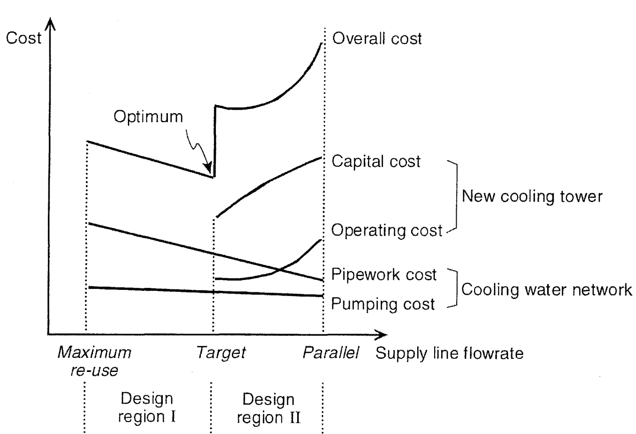


Figure 11. Debottlenecking cost for cooling-water systems.

As the cooling-water supply line is changed from parallel to maximum reuse conditions, the pressure drop of the network is increased due to the cooling-water reuse design, but the flow rate is decreased. Therefore, cooling-water pumping costs through the cooling-water network do not influence the overall cost significantly. Another expense is pipework cost. As the supply line for the cooling-water network approaches maximum reuse conditions, the reuse of cooling water between exchangers is increased, and, therefore, the pipework cost is increased. However, the cost of new piping is relatively low compared with that of a new cooling tower, as discussed by Kim and Smith (2001). From an inspection of the overall cost, the optimal design conditions are located at the target supply line because of a step change from the contribution of capital and operating costs for a new tower in design region II.

The final design at the target supply line always can be evolved for design simplicity. But this is likely to result in penalties being incurred and the design not achieving the target. When the network design does not achieve the target, that is, the supply line is located in design region II, this would likely result in a penalty for the cooling-system performance and new cooling capacity should be introduced.

Since the debottlenecking methods suggested employ the introduction of cooling-water reuse between coolers and an increase in flow rate for some coolers, the pressure drop across the cooling-water network is increased relative to the base case. In most retrofit cases, pressure-drop constraints or circulation pump performance become limiting factors for network design modifications. Therefore, the pressure-drop issue should be considered for the design of cooling-water networks with given target conditions.

As discussed previously cooling-water systems have interactions between the cooling-water network and the cooling tower. Because the cooling-water temperature and flow rate are linking variables between the cooling-water network and cooling tower, it would be best to integrate both subsystems simultaneously in the optimization framework. For executing optimization simultaneously, it is necessary to represent the thermal performance of the tower in an analytic equation. The analytic equation does not perfectly represent the thermal behavior of the tower because of its complexity and diversity. A complex expression with better correlation from rigorous regression might be obtained, but it will add high nonlinearity and nonconvexity to the optimization.

Therefore, in this study, a sequential approach to the debottlenecking design of cooling systems is proposed. First, a conceptual understanding of the problem is exploited to ex-

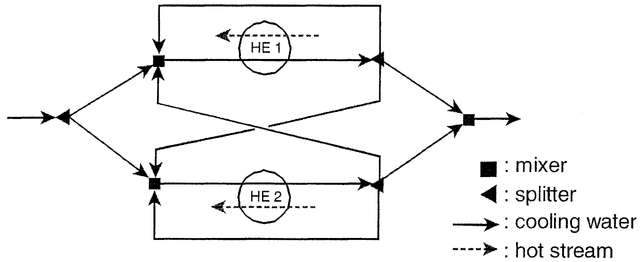


Figure 12. General superstructure.

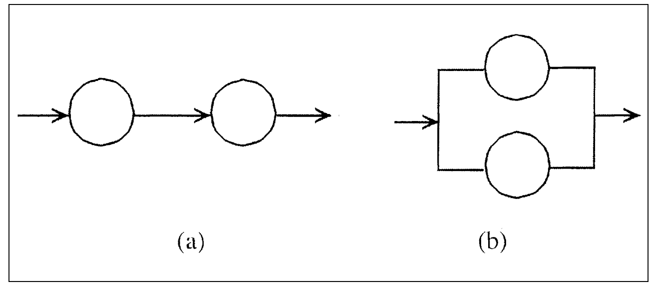


Figure 13. Pressure drop in different arrangements.

amine system interactions between the thermal performance of the tower and the cooling-water network design. The overall target temperature and flow rate are identified from the cooling-system model by changing the attainable conditions of the cooling-water network. Then, given the target conditions, the network design is optimized to find the configuration for minimizing the pressure drop. Although decomposition of targeting and design may not capture all perspective of the design problem, it can provide clear design guidelines and a practical approach to the real situation.

Since the flow rate as well as pressure drop affects the operating cost for pumping, the operating cost rather than the pressure drop of a network, is the correct criterion for the design. If the network design is searched in the whole design region, the minimization of operating cost should be chosen as an objective function. However, for this study, the target conditions, that is, the overall cooling-water flow rate and temperature to the tower are fixed from the targeting in the first stage of the approach. Then the second stage for network optimization is performed at a fixed overall cooling-water flow rate. Therefore, the minimization of pressure drop is an appropriate criterion to find an optimal solution for the network design.

Node – superstructure model

A superstructure is shown in Figure 12. The superstructure includes all possible connections between (1) the fresh cooling-water source and heat exchangers, (2) heat exchangers and cooling-water sink, and (3) heat exchangers. Some assumptions for the synthesis model are made: (1) The heat exchanger type of each cooling-water user is a 1-2 shell-and-tube type with countercurrent operation; (2) The specification of all heat exchangers is fixed and known; (3) Cooling load and limiting data (maximum inlet and outlet temperature) of the heat exchanger are known.

Although this superstructure includes all possible streams for the network, it is not sufficient to consider the pressure drop of a network, because the overall pressure drop of a network is determined by the pressure drop of individual units, as well as the configuration.

Let us consider why the calculation of overall pressure drop for complex networks gives rise to difficulties for basic or retrofit design with the simple arrangement shown in Figure 13. If two units are connected in series (Figure 13a), the pressure drop is the summation of the pressure drop of the two units. If two units are operated in parallel, the pressure drop is the maximum value of the two (Figure 13b). Therefore, the pressure drop cannot be calculated until the network configuration is fixed, because pressure drop is not an extensive property but an intensive property.

But pressure-drop issues should be addressed simultaneously when finding the optimal network design. To tackle the intensive characteristic of the pressure drop, a new superstructure model is proposed and "critical path algorithm" for a linear-programming (LP) problem is extended into an MINLP model.

To find clues to overcome the difficulties caused by the complex configuration of the network, let us consider the LP network problem that is known as a "critical path or longest route" algorithm as explained in Gass (1985). In Figure 14a, the node m (P_m) represents the pressure of starting, intermediate, or final destinations. The value of the arc (ΔP_{mn}) represents the pressure drop between nodes m and n. We need

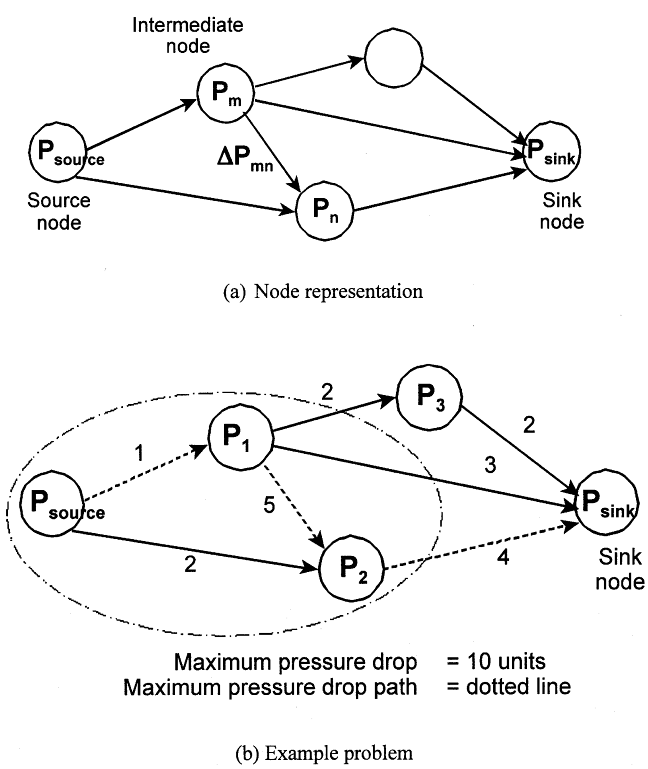


Figure 14. Pressure drop calculation for network with fixed configuration.

to find the maximum pressure-drop path among all paths that are generated in the networks from a source to a sink node. When formulating the inequality equations (Eq. 21), the maximum pressure-drop path can be found by minimizing $P_{\text{source}} - P_{\text{sink}}$.

$$P_m - P_n \geq \Delta P_{mn} \tag{21}$$

With the simple example shown in Figure 14b, let us illustrate how this algorithm works. The numbers on the arrow lines represent the pressure drops between the nodes. A large value is initially assigned to the value of P_{source} to prevent the accumulated value of pressure at the nodes to be negative. Suppose P_{source} has 20 units of pressure. When the dotted circle is considered, P_1 is 19 units, which satisfies the equality condition in Eq. 21. Then, the pressure of P_2 can be 14 units (path $P_{\text{source}} \rightarrow P_1 \rightarrow P_2$) or 18 units (path $P_{\text{source}} \rightarrow P_2$) without consideration of Eq. 21. However, if the pressure of P_2 is 18 units, Eq. 21 between P_1 and P_2 is violated: (19 units − 18 units) < 5 units. In this way, the accumulated value of pressure at each node would be assigned as $P_1 = 19$ units, $P_2 = 14$ units, $P_3 = 17$ units, and $P_{\text{sink}} = 10$ units. Therefore, the maximum pressure-drop path is $P_{\text{source}} \rightarrow P_1 \rightarrow P_2 \rightarrow P_{\text{sink}}$ with 10 units of pressure drop.

Other combinations of pressure can be assigned to nodes that will be greater than optimal value of 10 units in this example. For example, suppose that $P_1 = 19$ units, $P_2 = 13$ units, $P_3 = 15$ units, and $P_{\text{sink}} = 8$ units. Then the maximum pressure drop becomes 12 units, while all inequality equations between nodes are satisfied. As the pressure drop of the network ($P_{\text{source}} - P_{\text{sink}}$) is minimized, the optimal solution is given as 10 units of pressure, the minimum pressure drop of the network. It is important to note that the design task is to find the maximum pressure-drop path among all possible paths in a network, not the maximum pressure drop of the network. This maximum pressure drop path of the network obtained is minimized from the optimization, and the network design with minimum pressure drop is obtained.

Another important point to be noted is that the pressure at intermediate nodes and the pressure drop between nodes obtained from the optimization are no longer meaningful. Initially, the pressure-drop value between nodes is calculated with the condition that only two nodes are involved, not considering other connections following or followed by these nodes. As illustrated in Figure 13b, the overall pressure drop with a parallel branch will be the maximum pressure drop path of the two. The pressure drop for particular connections or units, which belongs to the particular path with a lower pressure drop than the maximum pressure-drop path, does not represent the initial pressure-drop value. Therefore, only the overall pressure drop between P_{source} and P_{sink} is meaningful for the overall network.

This "critical path" algorithm can be extended into the cooling-water network design, because the optimal pressure-drop path for networks depends on the quantity of the connections and units and the arrangement (sequence) of the heat exchangers and pipes.

The new "node−superstructure" combines the general superstructure with a node representation. In the general superstructure, the function of splitters and mixers is to distribute and merge streams. For the proposed node−superstructure model, the mixers and splitters are represented as nodes that contain not only the quantity of pressure, but also the distribution or merging of streams, as shown in Figure 15. The final mixer before going to the cooling tower becomes the sink node, and the first splitter from the tower becomes the source node. The intermediate nodes are splitters or mixers that are connected to each other, or sink, or source nodes. When the node i' is connected after i, the pressure difference between node i and node i' should be greater than the pressure drop of connection (i, i').

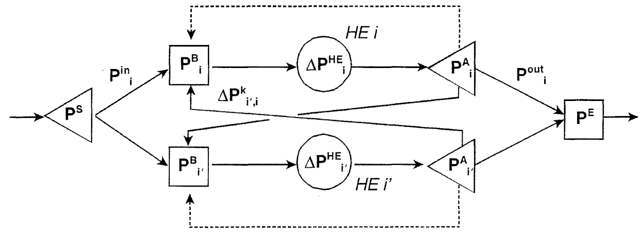


Figure 15. Node−superstructure.

However, the pressure drop cannot be calculated until the configuration is known, because some connections are selected, but others discarded by the optimization model. Therefore, an integer variable is introduced to extend the "critical path" algorithm from the LP framework into an MINLP. To consider whether a connection between a node is activated ($Y_{i,i'} = 1$) or not ($Y_{i,i'} = 0$), logic constraints for activation/deactivation are formulated by combining binary variables with a sufficiently large value. In Eq. 22, after introducing a large value (LV), if a node is connected with another node ($Y_{i,i'} = 1$), the inequality constraints are strictly activated. Otherwise ($Y_{i',i} = 0$), the inequality constraints are relaxed by a sufficiently large value. With the critical-path algorithm combined with the activation/deactivation scheme, the pressure of the node is determined according to the visiting order of the nodes. In other words, the value of each node is assigned to decide the maximum pressure-drop path of the network in the optimization process

$$P_{i'} - P_i + LV \cdot (1 - Y_{i',i}) \geq \Delta P_{i',i} \tag{22}$$

When setting the general superstructure for network synthesis, it is common to introduce a recycling stream, that is, the outlet stream from the unit is fed to the inlet of the same unit, in order to include all possibilities in network synthesis, as shown by the dotted lines in Figure 15. From the viewpoint of hydrodynamics, this is an unrealistic situation, because the recycling stream flows in the opposite direction to the stream in the unit, which is the flow against the pressure drop. It is also undesirable, since the recycling of cooling-water causes the buildup of fouling and scaling materials over a long period. Therefore, the recycling stream is excluded with the setting of $Y_{i,i} = 0$ for all coolers.

Model constraints

To obtain the minimum pressure drop for a network (this value corresponds to the maximum path among all the possi-

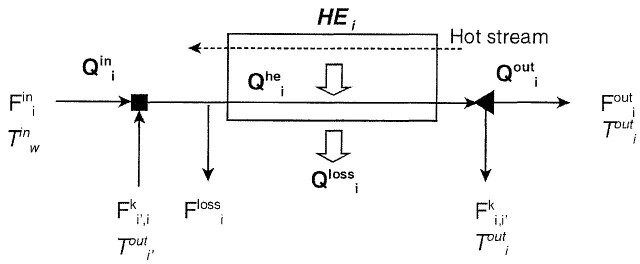


Figure 16. Around the heat exchanger.

ble paths under one given optimal configuration), the optimization problem is formulated as follows.

Since the heat load of each heat exchanger must be removed by cooling water, model constraints are identified from the node–superstructure as shown in Figure 16. The flow rate (F) is represented as the CP value in terms of kW/°C, which represents the flow rate multiplied by the heat capacity. It is assumed that the flow-rate loss occurs at the inlet point of the exchanger.

Thermal Design Constraints of Cooling-Water Networks. Heat balance in each heat exchanger

$$Q_i^{\text{in}} - Q_i^{\text{out}} + Q_i^{\text{he}} - Q_i^{\text{loss}} = 0 \tag{23}$$

Mass balance in each heat exchanger

$$F_i^{\text{in}} + \sum_{i'} F_{i',i}^k - F_i^{\text{out}} - \sum_i F_{i,i'}^k - F_i^{\text{loss}} = 0 \tag{24}$$

Flow rate in each heat exchanger

$$F_i^{\text{op}} + F_i^{\text{loss}} - F_i^{\text{in}} - \sum_{i'} F_{i',i}^k = 0 \tag{25}$$

Inlet heat load to heat exchanger

$$Q_i^{\text{in}} - \left[\sum_{i'} \left(F_{i',i}^k \cdot T_{i'}^{\text{out}} \right) + \left(F_i^{\text{in}} \cdot T_w^{\text{in,all}} \right) \right] = 0 \tag{26}$$

Outlet heat load from heat exchanger

$$Q_i^{\text{out}} - \left(F_i^{\text{op}} \cdot T_i^{\text{out}} \right) = 0 \tag{27}$$

Feasibility constraints on the inlet heat load

$$Q_i^{\text{in}} - \left(F_i^{\text{op}} \cdot T_i^{\text{in,max}} \right) \le 0 \tag{28}$$

Feasibility constraints on the outlet temperature

$$T_i^{\text{in}} - T_i^{\text{out,max}} \le 0 \tag{29}$$

Upper and lower bounds for cooling water flow rates

$$F_i^{\text{in}} - B_i^{\text{in},U} \cdot Y_i^{\text{in}} \le 0 \tag{30}$$

$$F_i^{\text{in}} - B_i^{\text{in},L} \cdot Y_i^{\text{in}} \ge 0 \tag{31}$$

$$F_{i,i'}^k - B_{i,i'}^{k,U} \cdot Y_{i,i'}^k \le 0 \tag{32}$$

$$F_{i,i'}^k - B_{i,i'}^{k,L} \cdot Y_{i,i'}^k \ge 0 \tag{33}$$

$$F_i^{\text{out}} - B_i^{\text{out},U} \cdot Y_i^{\text{out}} \le 0 \tag{34}$$

$$F_i^{\text{out}} - B_i^{\text{out},L} \cdot Y_i^{\text{out}} \ge 0 \tag{35}$$

Cooling water conditions flowing into the cooling tower are specified by the cooling-system model, according to the debottlenecking procedures. So the cooling-water network must satisfy the cooling-system requirement, that is, overall outlet temperature and flow rate from the network, regardless of pressure-drop effects. Equations for the target temperature and flow rate are formulated from the representation of the sink mixer.

Cooling-Water System Requirements. Overall outlet temperature

$$T_w^{\text{out,all}} \cdot \sum_i F_i^{\text{out}} - \sum_i \left(F_i^{\text{out}} \cdot T_i^{\text{out}} \right) = 0 \tag{36}$$

Overall outlet flow rate

$$F_w^{\text{all}} - \sum_i F_i^{\text{out}} = 0 \tag{37}$$

As presented in the previous section, a pressure-drop correlation is used to calcualte the pressure drop of the heat exchangers and pipes. Equation 16 or Eq. 19 is used for the existing or new pipes in Eqs. 38, 39, and 40. For Eq. 41, Eq. 4 or Eq. 9 is used according to the allocation of cooling water to heat exchanger.

Pressure-Drop Calculation. Pressure drop of pipe lines

$$\Delta P_i^{\text{in}} = f_N\left(F_i^{\text{in}}\right) \tag{38}$$

$$\Delta P_{i,i'}^k = f_N\left(F_{i,i'}^k\right) \tag{39}$$

$$\Delta P_i^{\text{out}} = f_N\left(F_i^{\text{out}}\right) \tag{40}$$

Pressure drop of heat exchangers

$$\Delta P_i^{\text{HE}} = f_N\left(F_i^{\text{op}}\right) \tag{41}$$

To convert the representation of flow rate from volumetic flow rate to CP, the following equation is also used for Eqs. 38, 39, 40, and 41

$$F = \frac{\rho c_p}{1{,}000} V \tag{42}$$

Maximum Pressure-Drop Path of Networks. Logic constraints for connections

$$P^S - P_i^B + LV \cdot \left(1 - Y_i^{\text{in}}\right) \ge \Delta P_i^{\text{in}} \tag{43}$$

$$P_{i'}^A - P_i^B + LV \cdot \left(1 - Y_{i',i}^k\right) \ge \Delta P_{i',i}^k \tag{44}$$

$$P_i^A - P^E + LV \cdot \left(1 - Y_i^{\text{out}}\right) \ge \Delta P_i^{\text{out}} \tag{45}$$

Heat exchangers

$$\Delta P_i^{\mathrm{HE}} = P_i^B - P_i^A \tag{46}$$

Objective Function. Minimize

$$P^S - P^E \tag{47}$$

Solution strategy

Because the MINLP model involves nonlinear and bilinear terms, nonconvexity from these terms can give multiple local minima and stationary points, or cause the failure of convergence in the nonlinear programming (NLP) algorithm. Some mathematical and physical features are investigated to overcome these difficulties.

All bilinear terms in Eqs. 26, 27, and 36 have the form of flow rate multiplied by temperature, but the temperature term in all bilinear terms is T_i^{out}. If T_i^{out} is fixed to be constant, the MINLP model has no bilinear terms, which means that T_i^{out} may be the key variable to deal with the nonconvexity caused by the bilinear terms.

Some physical insights regarding the T_i^{out} terms can be found from the characteristics of the design profile in the heat exchangers. It is evident that the inlet temperature does not change in a cooler that can use only fresh cooling water, and, hence, the design profile is exactly the same as the limiting profile. For other coolers, of which limiting inlet temperature has a higher temperature than a parallel design inlet temperature, the design profiles are illustrated in Figures 17, 18, and 19.

Since the target conditions for debottlenecking of cooling-water systems have a lower flow rate than parallel conditions, cooling water might be reused between some heat exchangers. So the inlet temperature to heat exchangers in debottlenecking situations is increased when compared with parallel

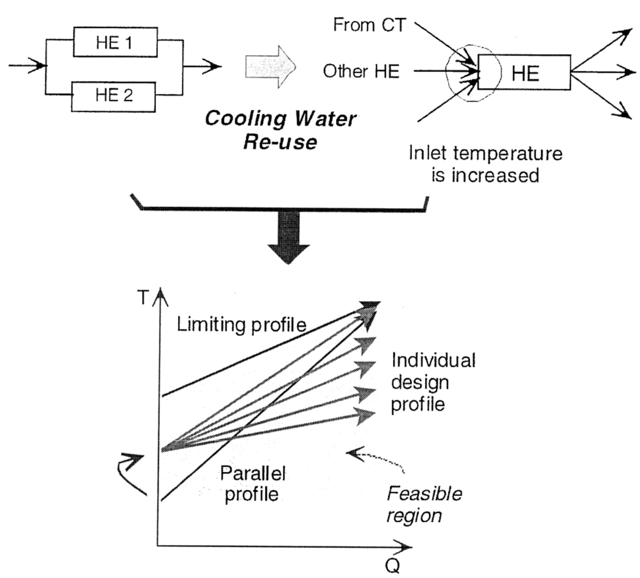


Figure 17. Characteristics of design profile (1).

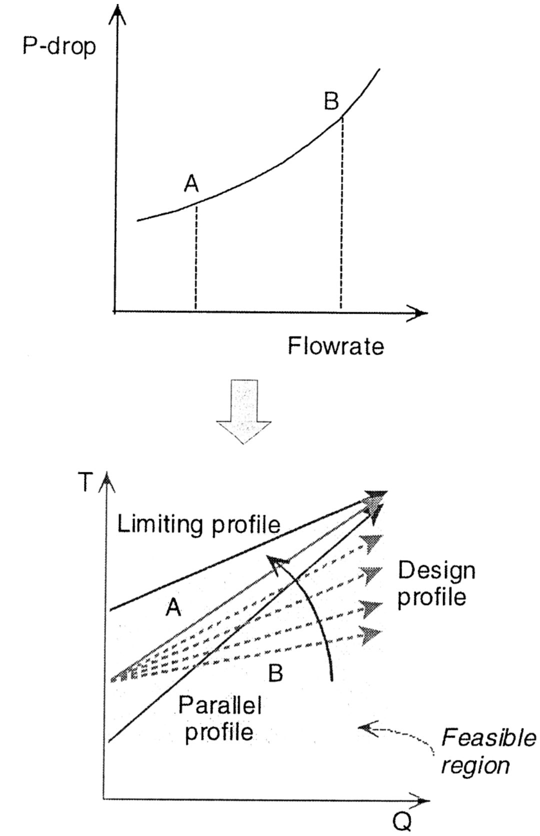


Figure 18. Characteristics of design profile (2).

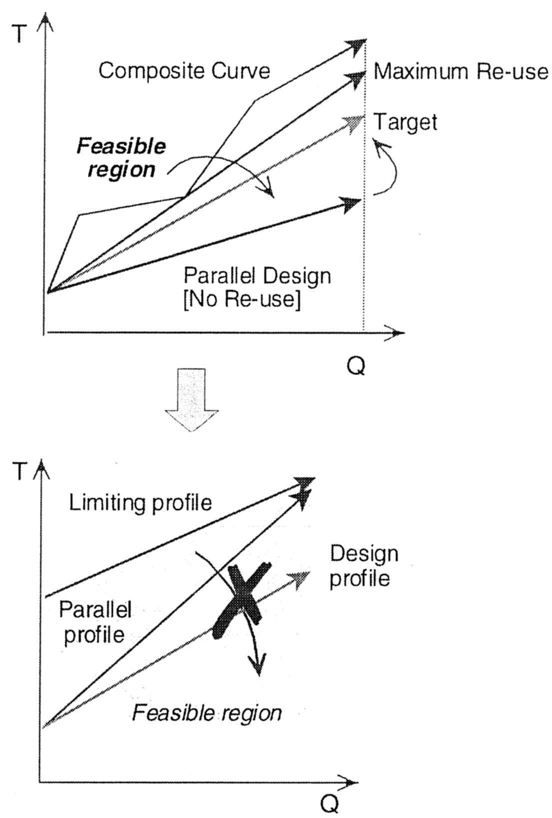
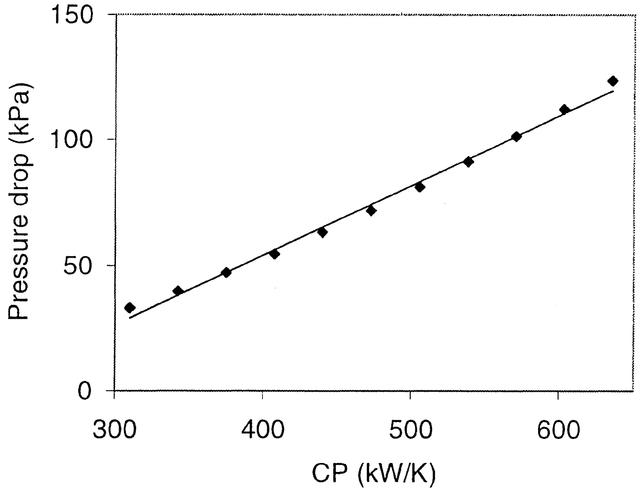
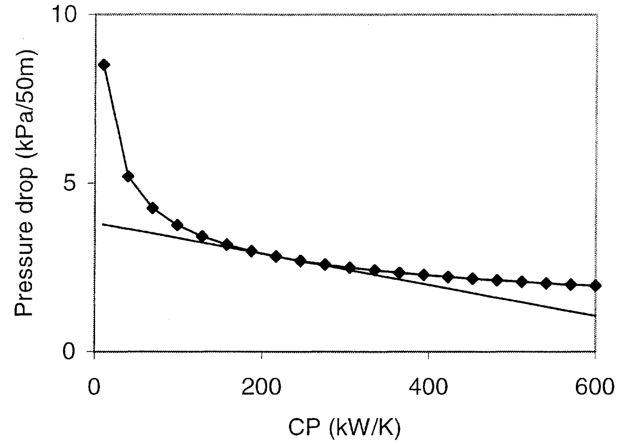


Figure 19. Characteristics of design profile (3).



(◆ : original – : linear)

Figure 20. Pressure drop in tube side of heat exchanger 2.



(◆ : original – : linear)

Figure 22. Pressure drop of new pipe.

conditions, as shown in Figure 17. In Figure 18, all design profiles with a dashed line can be feasible designs from the viewpoint of the thermal design of heat-exchangers with given inlet temperatures. But the design profile will tend not to increase the flow rate in order to decrease the pressure drop. The case when the inlet temperature of the target supply line is the same as the temperature for a parallel design is considered in Figure 19. T_i^{out} should be decreased to increase the flow rate. But this will not occur, because the overall target outlet temperature is higher than the parallel conditions. Also, the design profile keeps the outlet temperature high. From the characteristics of the design profiles, it can be concluded that the design profiles have a strong tendency to reach the maximum outlet temperature ($T_i^{out,max}$) when cooling water is reused in the network.

Given that the flow rate in the heat exchangers and pipes is changed within a limited range, pressure-drop correlations can be linearized to estimate nonlinear equations. Let us check about linearization of pressure-drop change over flow rate with Example 2. The linearization agrees well with the nonlinear expressions, with sufficient accuracy, as shown in Figures 20 and 21.

Figures 22 and 23 show the linearization for the pressure-drop calculation of the new and existing pipes. For the pressure drop of the new pipes in Figure 22, the linear equation may not be a valid estimation of the original correlation, which has a high curvature for the overall flow-rate region. But, low flow rates are seldom encountered in optimization, because the model does not choose pipelines with small flow rates to achieve the minimization of pressure drop for the network. Since the number of cooling-water sources that need fresh cooling water and reuse is usually more than two, very high

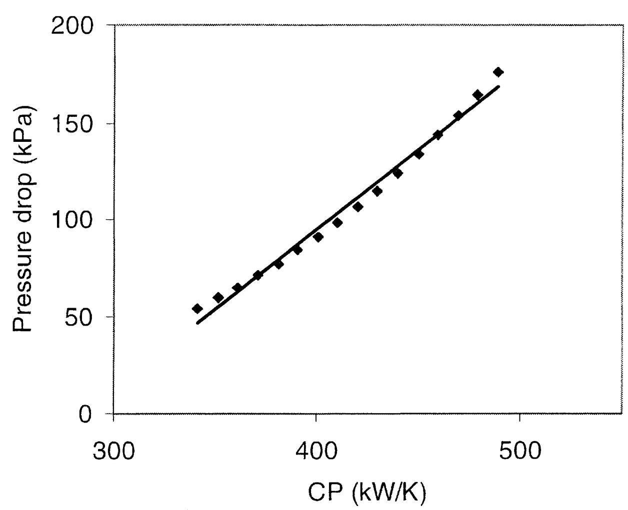
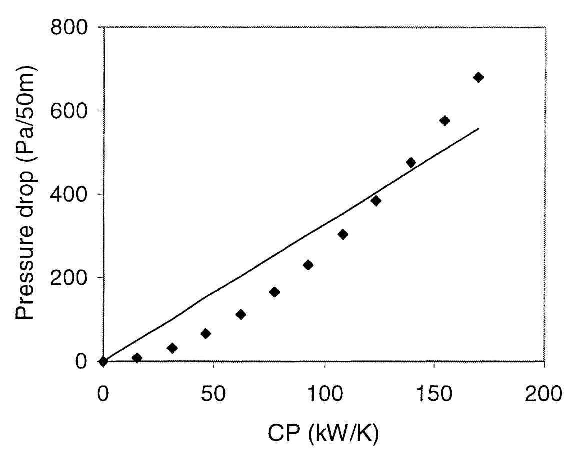


Figure 21. Pressure drop in shell side of heat exchanger 3.



(◆ : original – : linear)

Figure 23. Pressure drop of existing pipe from CT to heat exchanger 3.

flow rates for pipes rarely occur in network design. For a limited flow-rate range, the linear equation estimates the original function well.

Although the piecewise linearization method can be used to obtain a more accurate estimation of the original function, that is, estimation for quadratic function of the pressure-drop correlation, the numbers of binary variables increases significantly. From the node–superstructure of Example 2, 4 exchangers and 20 piping connections exist. If 5 binary variables are used for each exchanger and piping connection, 125 binary variables are required, which increases significantly the computational effort for optimization. Therefore, first-order linearization, rather than piecewise linearization, is adopted in this study.

Also, the use of deterministic global optimization algorithms reported by Quesada and Grossmann (1995) is another possibility for dealing with bilinear terms and nonconvexities. While their method provides a global solution, it also requires high computational effort and time, since their method is based on the spatial branch-and-bound algorithm by using a linear underestimator technique. Rather than using global optimization methods, a two-step procedure is proposed to solve the optimization model of cooling water networks, which will be discussed later in the article.

By setting T_i^{out} to be fixed and using linear estimation of the pressure-drop correlations, the MINLP model is converted into a mixed-integer linear programming (MILP) model. The MILP model is a good estimation of the MINLP model. The constraints for the MILP model follow.

MILP Model Constraints.

Inlet heat load

$$Q_i^{\text{in}} - \left[\sum_{i'} \left(F_{i',i}^k \cdot FT_{i'}^{\text{out}} \right) + \left(F_i^{\text{in}} \cdot T_w^{\text{in,all}} \right) \right] = 0 \tag{48}$$

Outlet heat load

$$Q_i^{\text{out}} - \left(F_i^{\text{op}} \cdot FT_i^{\text{out}} \right) = 0 \tag{49}$$

Overall outlet temperature

$$T_w^{\text{out,all}} \cdot \sum_i F_i^{\text{out}} - \sum_i \left(F_i^{\text{out}} \cdot FT_i^{\text{out}} \right) = 0 \tag{50}$$

Pressure drop of pipe line

$$\Delta P_i^{\text{in}} = f_L\left(F_i^{\text{in}} \right) \tag{51}$$

$$\Delta P_{i,i'}^k = f_L\left(F_{i,i'}^k \right) \tag{52}$$

$$\Delta P_i^{\text{out}} = f_L\left(F_i^{\text{out}} \right) \tag{53}$$

Pressure drop of heat exchanger

$$\Delta P_i^{\text{HE}} = f_L(F_i^{\text{op}}) \tag{54}$$

Because the linear equation for new pipelines has a finite value with zero flow rate, logical constraints are required to avoid inconsistency between an inactive match and a finite pressure-drop value.

Table 4. Shell-Side Specifications

Heat Exchanger	Heat Exchange Area (m^2)	Baffle Distance (m)	No. of Baffles	Shell Length (m)	Shell Dia. (m)
1	205	0.320	11	3.842	0.926
2	265.29	0.495	9	4.954	0.928
3	768.91	0.519	24	12.974	0.976
4	316.1	0.260	20	5.467	0.964

Note: 1. Overall heat-transfer coefficient is assumed as 800 W/m^2°C for heat-exchange area calculation.
2. Outer tube diameter (d_o) = 0.0254 m.
3. Tube pitch (P_t) = 1.25 × d_o.

Table 5. Tube-Side Specifications

Heat Exchanger	Heat Exchange Area (m^2)	No. of Tubes	Tube Length (m)
1	205	182	14.1
2	265.29	282	11.8
3	768.91	312	31.0
4	316.1	154	25.7

Note: 1. Overall heat-transfer coefficient is assumed as 800 W/m^2°C for heat-exchange area calculation.
2. Outer tube diameter = 0.0254 m.
3. Tube pitch diameter = 0.0211836 m.

Logical constraints

$$\Delta P_i^{\text{in}} - LV \cdot Y_i^{\text{in}} \leq 0 \tag{55}$$

$$\Delta P_i^{\text{out}} - LV \cdot Y_i^{\text{out}} \leq 0 \tag{56}$$

$$\Delta P_{i,i'}^k - LV \cdot Y_{i,i'}^k \leq 0 \tag{57}$$

The MILP model includes Eqs. 23–25, Eqs. 28–35, Eq. 37, Eqs. 42–46, and Eqs. 48–57, with the objective function of Eq. 47. The specifications of 1-2 shell-and-tube heat exchangers are shown in Tables 4 and 5. To compare changes in pressure drop on a uniform basis for heat exchangers with different capacities, the tube-side geometry is designed so that the velocity of the cooling water is 1.5 m/s. For the shell-side geometry, heat exchangers are designed so that the film heat-transfer coefficient has 3700 W/m^2°C, equivalent to a velocity of 0.8 m/s. In addition, the number of shells is taken as one. The physical properties of cooling water are assumed constant for all temperatures, as shown in Table 6. From Tables 4, 5, and 6, the parameters for N_t in Eq. 4, N_s in Eq. 9, N_p^{EX} in Eq. 16 and N_p^{NW} in Eq. 20 can be calculated.

Now, the MILP model will be applied to Example 2. When outlet temperatures for Exchangers 3 and 4 are varied, but those for 1 and 2 are fixed to their maximum values, Figure 24 shows solutions of the MILP problem that have target conditions of 50.3°C and 725.5 kW/°C. It is assumed that L_i^{in}, $L_{i,i'}^k$, and L_i^{out} are 50 m, and the cooling water is assumed to be used on the tube side of the exchangers.

Table 6. Physical Properties of Cooling Water

Density (kg/m^3)	Specific Heat Capacity (J/kg·°C)	Viscosity (Ns/m^2)	Thermal Conductivity (W/m·°C)
995.7	4178.6	7.97×10^{-4}	0.61786

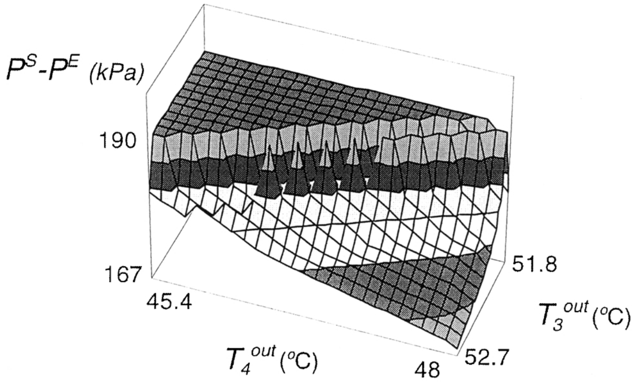


Figure 24. Optimal solution of MILP.

Because for the cooling-tower operation, the pressure drop comes mainly from the static head to lift the cooling water from the bottom of the top of the tower, the pressure drop of the cooling water for tower operation is not significant compared with that of networks. Therefore, the pumping head of the cooling water in the operation of the tower is not considered here. Flow and heat losses in each exchanger are also neglected. To avoid impractical design features, the minimum flow rate for pipes is assumed to be 10 kW/°C. Again, the pressure of first splitter node (P^S) is assigned a large value to prevent the pressure of the intermediate nodes and final mixer node being negative.

The global optimum is located where the outlet temperatures of exchangers 3 and 4 have their maximum values. The horizontal plane in Figure 24 at the lower levels of T_3^{out} and T_4^{out} represents the infeasible solution that cannot satisfy the overall outlet temperature and flow rate. Although the MILP problem does not perfectly represent the original MINLP problem, the general relationships between the outlet temperature and overall pressure drop in networks can be identified from Figure 24. The minimization of pressure drop in each heat exchanger has a dominant influence on deciding the minimization of the overall pressure drop in networks. Here we choose exchangers 3 and 4 for a variation of the outlet temperature to illustrate the features of the MILP solution, but other combinations of variation of T_i^{out} give the same result.

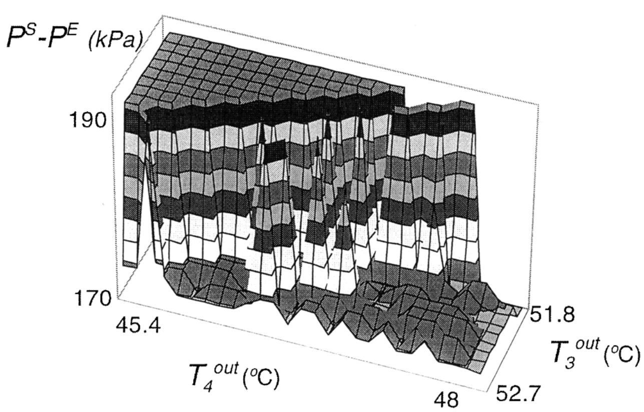
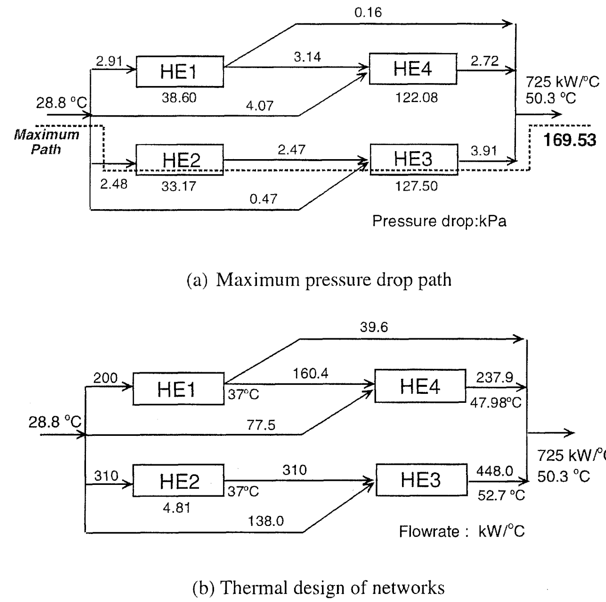


Figure 25. Optimal solution of MINLP.



(a) Maximum pressure drop path

(b) Thermal design of networks

Figure 26. Target configuration of tube-side usage.

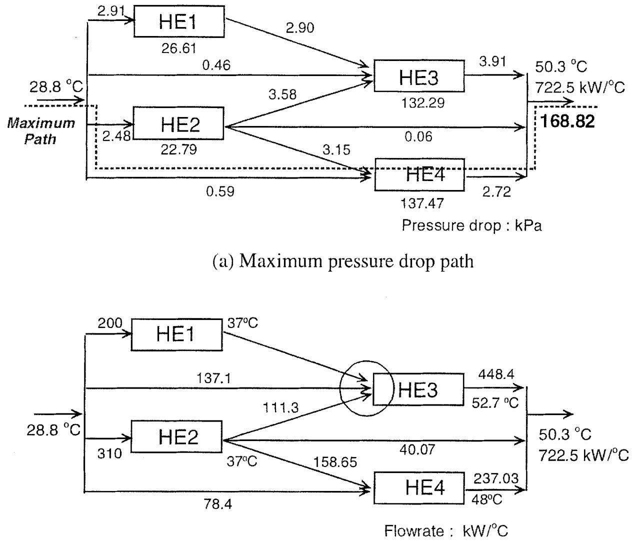
Now we have the optimum from linear approximations, and the MINLP optimum can be found by using nonlinear models. Figure 25 shows the MINLP solution using the solution of the MILP in Figure 24 as an initial point for the MINLP. Several optimal MINLP solutions with the same value are found from other initial solutions of the MILP with different settings of T_3^{out} and T_4^{out}. The optimal solution is located at the maximum levels of outlet temperature.

From physical insights, a solution strategy is suggested as follows. First, all outlet temperatures of heat exchangers are set as their maximum value and the MILP solved. Then the solution of the MILP is used as the initial point for the MINLP problem. The optimization was performed in the GAMS (Brooke et al., 1988) environment, and OSL was used for MILP and DICOPT++ for the MINLP.

Cooling water systems that operate with minimum pressure drop through the network are designed with the MINLP model satisfying cooling-tower target conditions for Example 2. Figure 26 shows the target configuration with minimum pressure drop when the cooling water is used in the tube side.

Because the allocation of cooling water streams on the tube or shell side should consider practical issues—corrosion, fouling, fluid temperature, operating pressure, viscosity, and stream flow rate—use of cooling water on the shell side is also investigated. Figure 27 shows the target configuration with minimum pressure drop when the cooling water is used on the shell side. The combined usage of cooling water on the tube and shell side can also be handled by the current method by changing the appropriate equations from the tube to shell side or from the shell to tube side.

Since the optimization model has no restrictions on the design complexity of networks, the target design can include impractical design features, which often happen when designing large networks. As shown in Figure 27, exchanger 3 has three cooling-water sources. The complexity of the network can be reduced by including complexity constraints in the



(a) Maximum pressure drop path

(b) Thermal design of networks

Figure 27. Target configuration for shell-side usage.

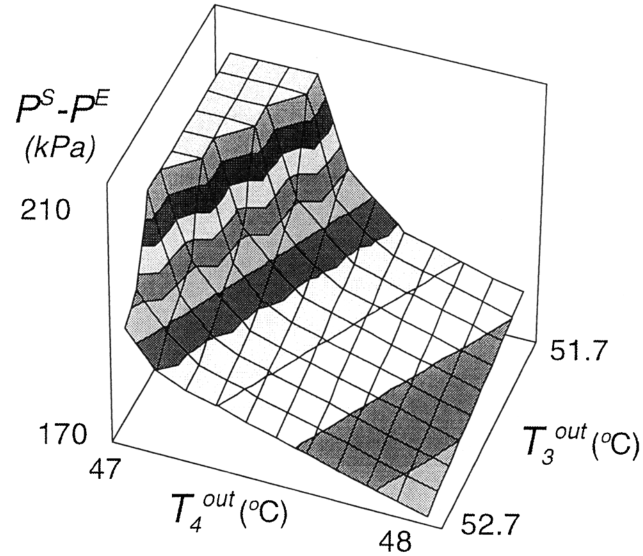
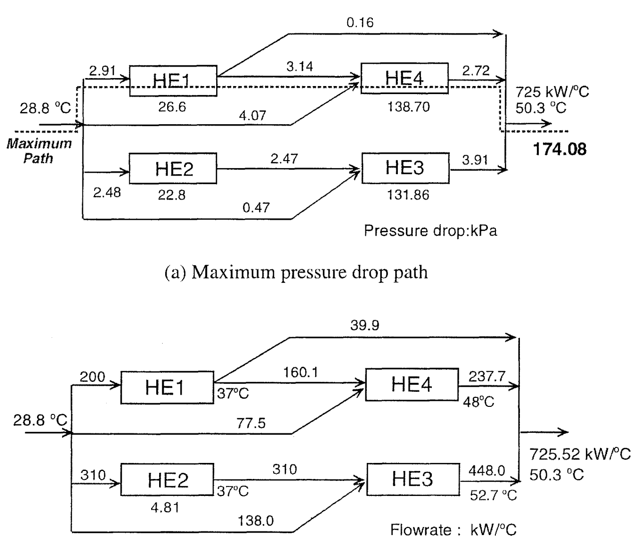


Figure 29. Optimal solution of MILP with different pipe length.

model (Eq. 58). When the maximum number of sources to any operation is restricted to be say two, the target network design appears as in Figure 28. The pressure drop is increased from 168.82 kPa to 174.08 kPa as a result of the complexity constraint

$$Y_i^{\text{in}} + \sum_i Y_{i',i}^k \leq M_{ns} \tag{58}$$

Because the MINLP solution with the standard algorithm is strongly dependent on the starting point (Zamora and Grossmann, 1998), it is necessary to verify whether the proposed solution strategy is reasonable or not. As the pressure drop of pipes (< 10 kPa) is far less than that of the heat exchangers (20 ~ 140 kPa) at the optimal configuration in Figures 26, 27, and 28, the pipeline pressure drop does not seem to have much effect when targeting the overall pressure drop of the network. Also, in practice, the pipes in the heat exchangers are not all of the same length. It is necessary to check whether pressure drop in the pipes influences the network structure. Figure 29 shows the result of the MILP solution when pipes have different lengths, as shown in Table 7. The general features remain the same, though the pressure drop of the pipe is large enough to affect the overall pressure drop of the network.



(a) Maximum pressure drop path

(b) Thermal design of networks

Figure 28. Target configuration for shell-side usage with complexity constraint.

As the target conditions have a considerably higher temperature (50.3°C) than the parallel design (44.1°C), it can be deduced that the individual outlet temperature is inevitably increased as much as possible to satisfy the overall target temperature, regardless of pressure drop. A different target temperature (47°C) is investigated to consider the relationship of the solution strategy with target temperature. As shown in Figure 30, the proposed strategy is also effective for network design in which the outlet temperature of the heat exchangers do not reach their maximum levels.

Again, as pointed out in an earlier section, the values of pressure drop shown in Figures 26, 27, 28, and 36 are used to find the maximum pressure-drop path for the network during

Table 7. Pipe-Length Data

Distance (m)	HE 1	HE 2	HE 3	HE 4
From CT	200	150	300	450
HE 1	0	100	500	350
HE 2	100	0	250	400
HE 3	500	250	0	200
HE 4	350	400	200	0
To CT	350	150	200	500

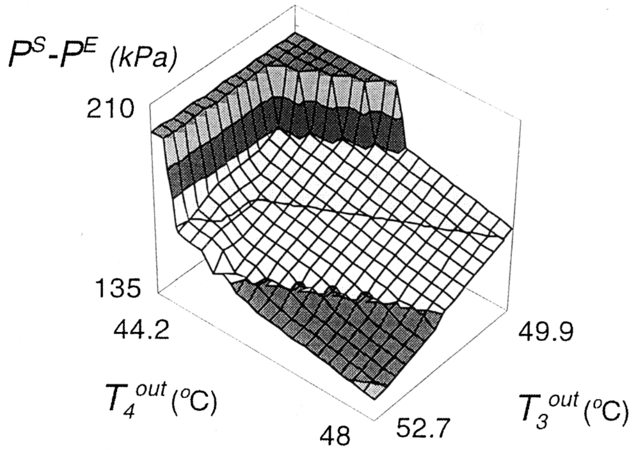


Figure 30. Optimal solution of MILP with lower target temperature.

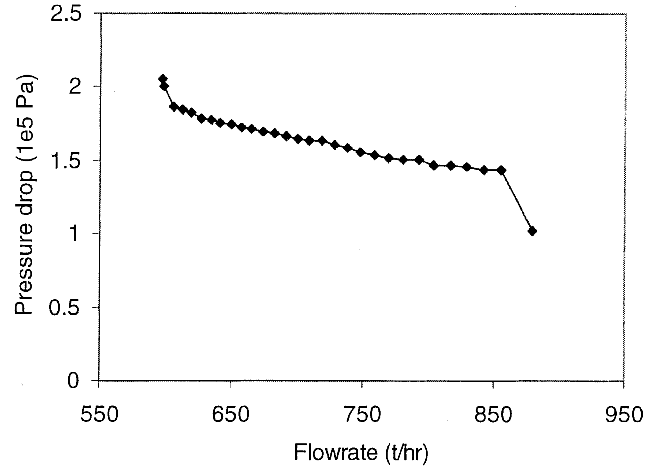


Figure 32. Retrofit pressure-drop curve for tube side.

optimization, and the overall pressure drop is only a meaningful indication of the pressure drop of the network.

Since less than 1.5 CPU s of computational time is normally required for the examples in this article (Pentium PC; 128 MB RAM; 600 MHz), the proposed model is expected to consume a reasonable CPU time even for large problems.

Retrofit of Cooling-Water Systems

At target conditions, the best network configuration can be found with the MINLP model. Let us now incorporate debottlenecking procedures. Typically, when a process is designed, the operating conditions of the pump are selected by considering the system requirement curve that represents the pressure drop of the system at various flow-rate conditions (Figure 31). The system curve is calculated from a fixed system configuration. For retrofit of cooling-water systems, the network configuration is changed according to the target conditions. The system requirement curve for pressure drop should be determined by the MINLP as target conditions of the network are varied. That system curve will be termed the retrofit pressure-drop curve.

As shown in Figure 10, the feasible region can be divided into design regions I and II. The retrofit pressure drop curve for design region I and II are illustrated in Figure 32 for tube-side flow and in Figure 33 for shell-side flow. Each point on the system curve represents the optimal value found by the MINLP model at different target conditions. As seen in Figures 32 and 33, the curve increases sharply just after the parallel design point. That is because there is at least one series match between heat exchangers, though the overall cooling-water flow rate is decreased slightly when compared with the parallel configuration. When the overall flow rate is approached near the maximum reuse point, the pressure drop increases steeply because the flow rate of Exchanger 3 must increase significantly to satisfy the overall outlet temperature.

Now the retrofit pressure-drop curve is superimposed on the performance curve of the centrifugal pump. Two cases can be expected. First, the target conditions belong to design region I, and the pump can handle the pressure-drop increment by changing impeller diameter. As shown in Figure 34, the impeller diameter needs to be increased to satisfy the retrofit pressure-drop curve in this case, because the retrofit pressure-drop curve exceeds that of the pump head availability with a fixed diameter.

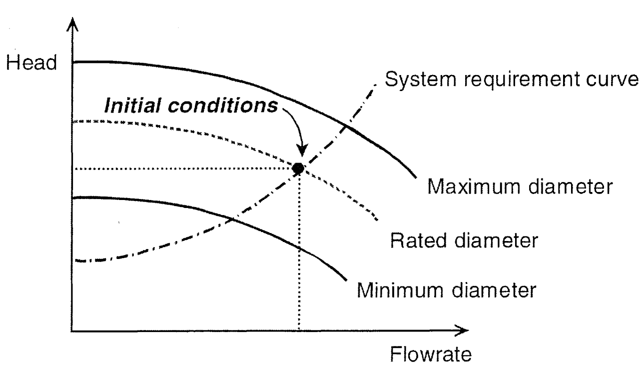


Figure 31. Centrifugal pump performance curve.

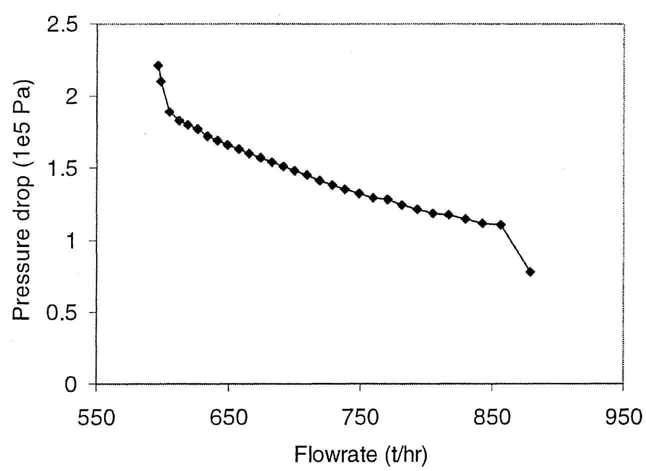


Figure 33. Retrofit pressure-drop curve for shell side.

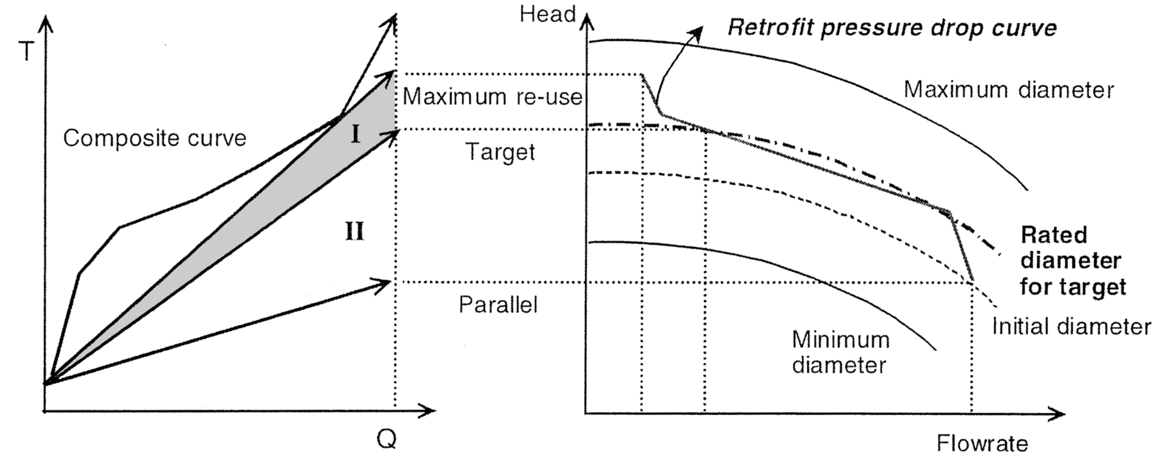


Figure 34. Retrofit of cooling-water systems for design region I.

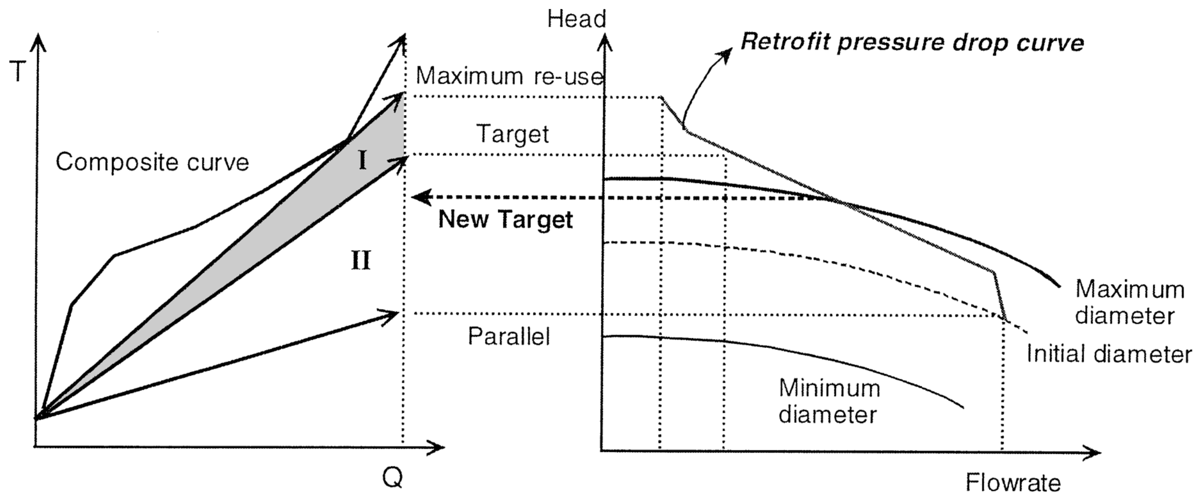


Figure 35. Retrofit of cooling-water systems for design region II.

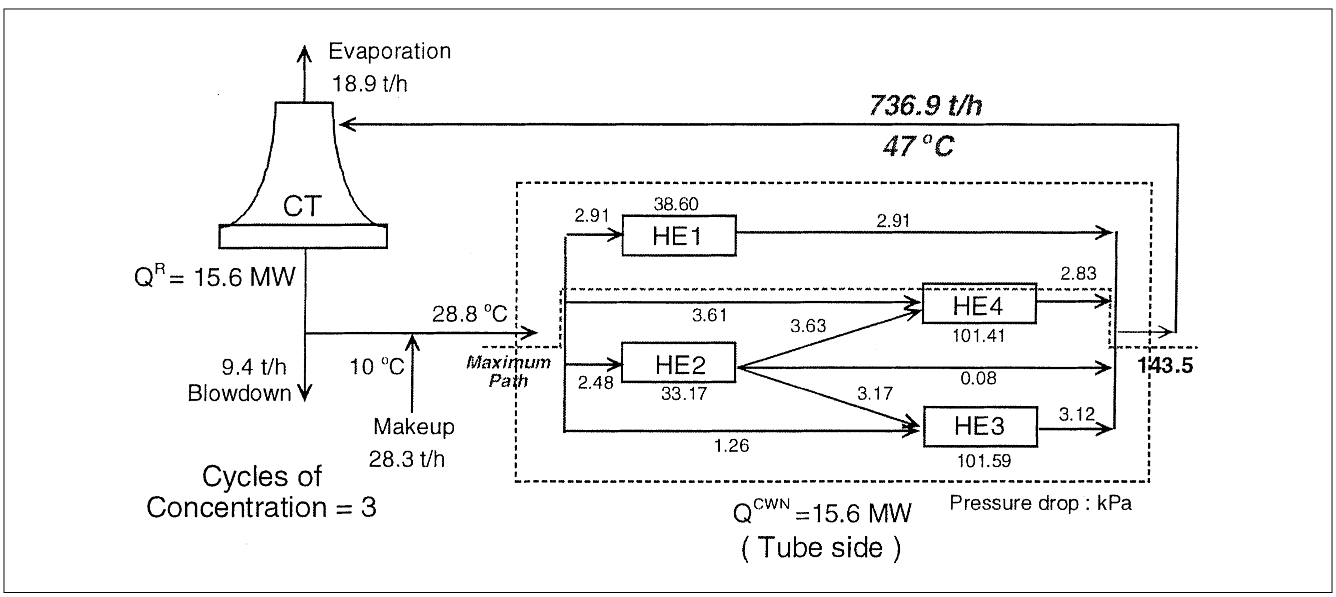


Figure 36. Retrofit of cooling-water systems with heat-load distribution.

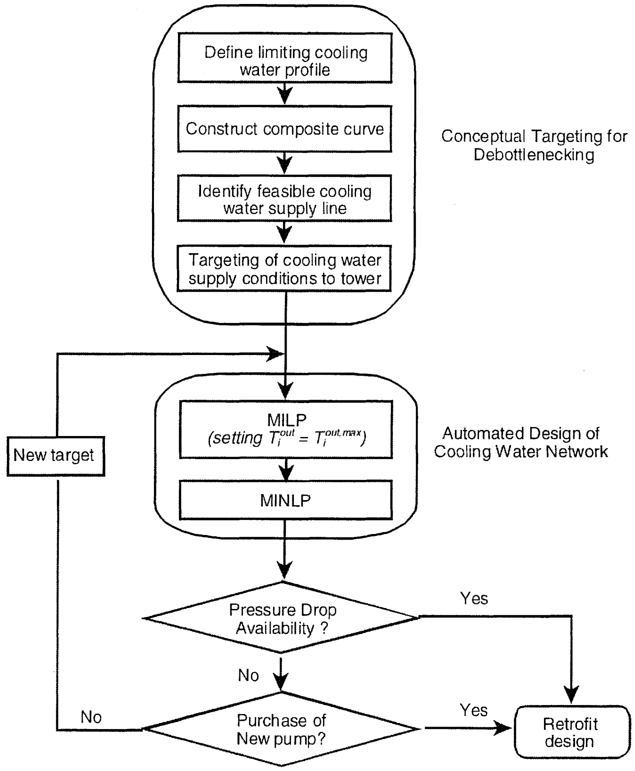


Figure 37. Overall retrofit design algorithm.

For the second case, the target conditions for the cooling system are located in design region II, and the current pump capacity is short of the desired pressure head and the maximum impeller diameter cannot satisfy the requirements (Figure 35). As explained previously, combining heat-load distribution design options is necessary in addition to the current cooling tower if new pumping systems are not desirable.

Among the several design options, the case for increasing air flow rate is illustrated in Figure 36 when the new target conditions for the tower are taken as 47°C and 736.9 t/h. For this case, air flow rate is increased to 846 t/h, but the pressure drop of the network is decreased from 169.53 kPa to 143.5 kPa. Detailed descriptions of the heat-load distribution can be found in Kim and Smith (2001).

The rated impeller diameter is selected by the target conditions and the retrofit pressure-drop curve by simple evaluation. There is a trade-off between the selection of the impeller diameter and the desired target conditions. This will be the subject of future work. As the desired target supply line approaches the parallel line, a new cooling capacity is installed, and, therefore, the capital cost increases, but a change of impeller may be avoided. However, as the desired target supply line approaches the maximum reuse line, the pump impeller should be changed to a new one with a large diameter or installation of a new pump may be considered because of limitations on the maximum diameter, while new cooling equipment is not necessary for the current systems.

The overall algorithms for the retrofit design of cooling systems are summarized in Figure 37. A two-step approach is used. First, conceptual targeting is carried out to obtain the overall target temperature and flow rate for debottlenecking. In the next stage, given the target conditions, the network structure to minimize the pressure drop of the network is determined. If the pump head availability and new purchase of a pump is not acceptable, the new target conditions are adjusted.

Conclusions

An automated design procedure for cooling-water networks has been developed in order to consider pressure-drop constraints, complexity of networks, and the efficient use of cooling towers. In debottlenecking situations for cooling-water systems, the cooling-water network design is incorporated with pressure-drop aspects into an MINLP model.

A new node–superstructure has been proposed to overcome the difficulties in calculating the pressure drop in the network and for considering the intensive property of the pressure drop. The new network representations can deal with the process integration task as well as network problems like determining the critical path of properties.

From physical insights by observing design profile characteristics in coolers, a solution strategy is suggested in order to give a good initial point for MINLP by fixing the outlet temperature of the individual heat exchangers at their maximum levels. The strategy suggested is a simple, but a robust method for solving the MINLP model.

Retrofit analysis is incorporated with the automated design method and gives design guidelines for the debottlenecking of cooling-water systems. When the cooling-water networks cannot operate within pressure-drop availability, the required change in impeller diameter can be determined, with consideration of the cost penalty caused by the installation of additional cooling capacity.

Notation

The sets, decision variables and parameters for optimization problem are given as:

Sets

$I = \{i | i \text{ is a heat exchanger}\}$

Variables

F_i^{in} = cooling-water flow rate from cooling tower to heat exchanger, $i \in I$
F_i^{out} = cooling-water flow rate from heat exchanger to cooling tower, $i \in I$
$F_{i,i'}^k$ = cooling-water flow rate from heat exchanger i to i', i, $i' \in I$
F_i^{op} = cooling-water flow rate in heat exchanger, $i \in I$
P^S = pressure of first splitter node in the cooling-water network
P^E = pressure of final mixer node in the cooling-water network
P_i^A = pressure of splitter node after heat exchanger, $i \in I$
P_i^B = pressure of mixer node before heat exchanger, $i \in I$
ΔP_i^{HE} = pressure drop of heat exchanger, $i \in I$
ΔP_i^{in} = pipe pressure drop for the stream of F_i^{in}, $i \in I$
$\Delta P_{i,i'}^k$ = pipe pressure drop for the stream of $F_{i,i'}^k$, i, $i' \in I$
ΔP_i^{out} = pipe pressure drop for the stream of F_i^{out}, $i \in I$
Q_i^{in} = inlet heat load of heat exchanger, $i \in I$
Q_i^{out} = outlet heat load of heat exchanger, $i \in I$
T_i^{out} = outlet temperature of cooling water from heat exchanger, $i \in I$
Y_i^{in} = binary variable for the stream of F_i^{in}, $i \in I$

$Y_{i,i'}^{k}$ = binary variable for the stream of $F_{i,i'}^{k}$, i, $i' \in I$
Y_{i}^{out} = binary variable for the stream of F_{i}^{out}, $i \in I$

Parameters

$B_{i}^{in,U}$ = upper bound on the stream of F_{i}^{in}, $i \in I$
$B_{i,i'}^{k,U}$ = upper bound on the stream of $F_{i,i'}^{k}$, i, $i' \in I$
$B_{i}^{out,U}$ = upper bound on the stream of F_{i}^{out}, $i \in I$
$B_{i}^{in,L}$ = lower bound on the stream of F_{i}^{in}, $i \in I$
$B_{i,i'}^{k,L}$ = lower bound on the stream of $F_{i,i'}^{k}$, i, , $i' \in I$
$B_{i}^{out,L}$ = lower bound on the stream of F_{i}^{out}, $i \in I$
F_{i}^{loss} = cooling-water flow-rate loss in heat exchanger, $i \in I$
F_{w}^{all} = overall outlet cooling-water flow rate from networks
FT_{i}^{out} = fixed outlet cooling-water temperature of heat exchanger, $i \in I$
L_{i}^{in} = pipe length for the stream of F_{i}^{in}, $i \in I$
$L_{i,i'}^{k}$ = pipe length for the stream of $F_{i,i'}^{k}$, i, $i' \in I$
L_{i}^{out} = pipe length for the stream of F_{i}^{out}, $i \in I$
N_t = coefficient for nonlinear pressure-drop correlation of tube side
N_s = coefficient for nonlinear pressure-drop correlation of shell side
N_p^{EX} = coefficient for nonlinear pressure-drop correlation of existing pipe-line
N_p^{NW} = coefficient for nonlinear pressure-drop correlation of new pipe-line
Q_i^{he} = heat load of heat exchanger, $i \in I$
Q_i^{loss} = heat loss of heat exchanger, $i \in I$
$T_w^{in,all}$ = overall inlet cooling water temperature to networks
$T_w^{out,all}$ = overall outlet cooling water temperature from networks
$T_i^{in,max}$ = limiting maximum inlet temperature of cooling water to heat exchanger, $i \in I$
$T_i^{out,max}$ = limiting maximum outlet temperature of cooling water from heat exchanger, $i \in I$

General

A = heat-exchanger area
CP = flow rate multiplied by heat capacity, kW/°C
c_p = heat capacity, J/kg·°C
CT = cooling tower
CW = cooling water
$D(d)$ = diameter, m
F = flow rate, kW/°C
F_b' = bypass correction factor
F_h = correction factor for h value
F_L' = leakage correction factor
f_L = linear equation
f_N = nonlinear equation
f_o = fanning fraction factor
h = heat-transfer coefficient, W/m^2·°C
K = coefficient for pressure-drop correlation in terms of velocity
L = pipe length, m
LV = large value
k = thermal conductivity, W/m·°C
M_{ns} = maximum number of sources to operation
n_t = number of tubes
n_{tp} = number of tube passes
P = pressure, N/m^2
p = tube pitch, m
ΔP = pressure drop, N/m^2
Re = Reynolds number
Q = heat load or cooling load
T = temperature, °C
ΔT_{min} = minimum approach temperature, °C
u = velocity, m/s
V = volumetric flow rate, m^3

Greek letters

ρ = density, kg/m^3
μ = viscosity, Ns/m^2

Subscripts and superscripts

DBT = dry-bulb temperature of air, °C
hot = hot process stream
i = heat-exchanger unit or inside of tube
in = inlet stream to heat exchanger
m = node of network
max = maximum
min = minimum
n = node of network
o = outside of tube
out = outlet stream from heat exchanger
p = pipeline
s = shell side
so = shell side and reference point
shift = temperature shift
sink = sink node
source = source node
t = tube side
V = volumetric flow rate, m^3/s
WBT = web-bulb temperature of air, °C
w = cooling water
CWN = cooling-water network
opt = optimum
R = removal heat load of cooling tower
$*$ = new pinch point

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