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RE-THINKING THE DESIGN OF CLOSED-LOOP COOLING STATIONS TO MITIGATE FOULING

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ABSTRACT

Parallel heat exchangers in closed-loop cooling stations are typically designed identically. Such a design strategy has some beneficial features such as simplified design and construction processes, as well as the possibility to cannibalize spare parts should that be necessary. However, in some applications, such as datacenter or district cooling, the required cooling capacity is time-dependent, e.g. over the day or season. Hence, the station will recurringly operate at part load, having flow rates lower than design flow rates, sometimes significantly so. Often, part load is met by reducing the number of heat exchangers in service but the difference to design velocity may still be large. This increases the risk for, or rather the rate of, fouling on the cold side of the station. If instead the cooling station is designed with non-equal heat exchanger capacities, there will be more station capacities to access when following a varying cooling requirement, thus increasing the possibility to operate close to design velocity, hence limiting the rate of fouling. The present numerical study shows that such a design approach may reduce the cleaning cost by up to 40 %, and the combined cleaning and pumping cost due to fouling with 15 to 20 %.

Designing the heat exchangers of a cooling station with different capacities only requires a new way of thinking for the heat exchanger provider, and should thus be possible to quote as of today. However, a re-design of the control logic would be required.

INTRODUCTION

Closed-loop cooling systems (secondary-cooling systems) is a common way to arrange the cooling of various kinds of facilities, such as power plants, oil refineries, chemical plants, district cooling network centrals, and data centers. The heat sink can be either the sea, a river, or the atmosphere via a cooling tower facility. In all these cases, fouling can be expected to occur on the coolant side of the heat exchangers of the cooling station.

Fouling, i.e. deposits on heat transfer surfaces, will lower heat transfer rates and make targeted cooling more difficult. For example, in data centers it is crucial to keep the temperature of the servers below a certain limit (typically 85 °C [1]) for safe operation. This is a fairly significant industry sector – the global energy use in data centers have been estimated to amount to 205 TWh, or 1 % of the global electricity consumption in 2018 [2] – and in our digitalized world it is a fundamental one. In this industry there is a drive to decrease the power usage effectiveness (PUE, the total data center power usage divided by the IT equipment power usage), and secondary cooling (a.k.a. free cooling) is one alternative as opposed to using compression refrigeration systems [3]. However, the issue of fouling must be managed.

Fouling can be caused by a number of phenomena, including crystallization, particulate deposition, biological growth, chemical reaction, corrosion, and freezing, often in combination [4]. Many factors influence the growth of fouling. The three most important ones are temperature, fluid velocity, and concentration of foulant precursor [4]. A high fluid velocity, or rather wall shear stress, will for most types of fouling decrease the fouling rate [4][5] but comes with a high pumping power cost. For biological and crystallization fouling there may be a maximum in fouling rate with respect to fluid velocity due to increased mass transfer rate, but a high wall shear stress is still preferred [4][5].

Means to mitigate fouling include using coolant filters, modification of coolant chemical composition, and modification of heat transfer surfaces. A summary of seawater fouling mitigation approaches – such as fouling inhibitors in the coolant, heat exchanger design and material selection, and control strategy – is given by Pugh et al. [5]. Ishiyama et al. [6] studied the interaction between heat transfer enhancement inserts and fouling in tubular heat exchangers. Santos et al. [7] reported on trials with Teflon-coated plate heat exchanger surfaces.

When retrofitting heat exchangers, Coletti et al. [8] have shown that it is important to study the effect on the whole system even if only one heat exchanger design is altered. They analyzed the system-level consequences of a simulated retrofit of one shell-and-tube heat exchanger in a crude oil pre-heat train.

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Some applications, such as district cooling and data center cooling experience daily and seasonal variation in the required cooling. This often implies a variation in wall shear stress as the heat exchangers will at times operate at part load. The standard advice is to maintain design velocity even at part load [4][5], e.g. by recirculation. However, this is not always followed in practice, as has been observed through the Alfa Laval *Smart Heat Exchanger* health monitoring system [9].

The present study focuses on the overall design of the cooling station, more specifically on the distribution of the total cooling capacity (heat transfer area) on the heat exchangers. With non-uniform distributions it is possible to achieve a smaller reduction of the shear stress at part load operation. This study tries to answer, on a high level, the questions (a) Can the operating cost due to fouling be decreased by distributing heat exchanger capacities differently? and (b) If so, by what magnitudes? Two different cooling requirement curves and three different coolant control strategies are considered.

The investment cost of the cooling stations is out of scope for the present study. However, one may note that if the cost for a heat exchanger scales linearly with size, the investment cost will be the same for all heat exchanger capacity distributions. On the other hand, if the cost scales with an exponent less than unity, the equal distribution will have the highest investment cost.

METHOD

The model will be described with a cooling station in mind and fouling will be assumed to occur on the cold fluid side of the heat exchangers.[1]

In the present case, then, the load is the required cooling power, which can take any value within a given heat load range. Each heat exchanger is designed for a nominal capacity, performed at design conditions. The combined nominal capacity of the employed heat exchangers in a station ideally matches the nominal required heat load. However, since the nominal capacity of a heat exchanger is fixed, the combined nominal capacity will take on discrete values, and will thus in general not match exactly the required heat load at a given instant.

Given a known temporal distribution of the required load and a distribution of heat exchanger nominal capacities, the time-averaged metrics (defined in the Metrics section below), viz. (a) the relative cost for cleaning and (b) the relative cost for pumping power, are to be determined. The relative cost for pumping power will be evaluated for negligible fouling flow resistance and for fouling with some flow resistance, respectively.

The programming language Python 3.10 from the Python Software Foundation[2] was used for coding the calculations and post-processing the result.

Model

The cooling station is designed to handle a nominal (maximum) required heat load $w_{\max}$. However, over a period of, say, one year, the required heat load varies and can take any value in the range $w_{\min} \simeq 0$ to $w_{\max}$ (see section Heat Load Distribution below). For simplicity, it is assumed that the inlet temperatures are constant over the entire period. Also, it is assumed that the flow rates (hot and cold side flow rates) to the station, as well as to the individual heat exchangers, can be controlled.

At any given instant, the total cooling power delivered by the station is given by the total mass flow rate $\dot{m}$ of each side (hot and cold) which will result in some NTU of the heat exchangers (assumed equal for all units, see section Heat Exchanger Station below). Hence, an effectiveness $\epsilon \geq \epsilon_{\mathrm{reqd}}$ and a pressure drop $\Delta p \leq \Delta p_{\mathrm{perm}}$ must result, otherwise more heat exchanger capacity needs to be employed.

We consider a cooling case in which the cold side flow rate is higher than the hot side flow rate, as is normally the case, hence the minimum fluid capacity rate $C_{\min} = C_{\mathrm{hot}} = \left(\dot{m}c_p\right)_{\mathrm{hot}}$. The states before and after the change of required heat load are denoted using the subscripts 0 and 1, respectively.

In this discussion then, the hot side mass flow rate $\dot{m}_{\mathrm{hot}}$ is directly proportional to the required heat load. The heat transfer coefficient (HTC) $\alpha \propto \dot{m}_{\mathrm{hot}}^r$ where r is in the approximate range 0.6 to 0.8 for many common heat exchanger channels in the turbulent Reynolds region. Hence,

$$\mathrm{NTU} = \frac{kA}{C_{\mathrm{hot}}} \propto m_{\mathrm{hot}}^{r-1} \tag{1}$$

if the overall heat transfer coefficient (OHTC) k is taken as primarily dependent on the hot side HTC. A is the heat transfer surface area.

Now, if the hot side mass flow rate is reduced, $0 < C_{\mathrm{hot},1}/C_{\mathrm{hot},0} < 1$, the NTU will increase, and the resulting return temperature will depend on how the cold side flow rate is set in response. This is illustrated by the open symbols in Fig. 1. The responses will be discussed next.

[1] However, the overall rationale applies to a heating station as well; in fact, the present discussion has relevance also to other types of process equipment operating in parallel and being exposed to varying load requirements, such as pumping stations, centrifuge stations, etc.

[2] www.python.org

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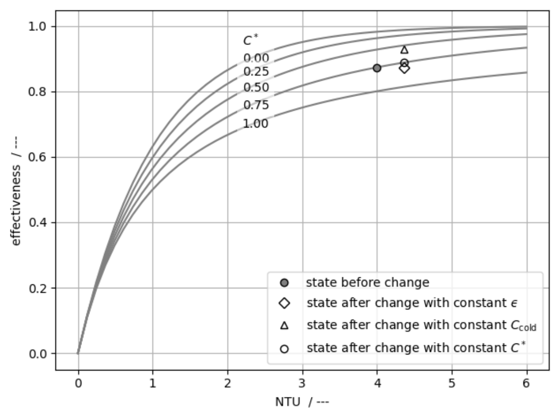

Fig. 1 Effectiveness curves for a counter-current arrangement with C^* as parameter. Also indicated are the resulting effectivenesses due to the different control strategies.

Consider a cooling case corresponding to the design case with the required effectiveness $\epsilon = \epsilon_{\text{reqd}}$ and pressure drops $\Delta p_{\text{hot}} = \Delta p_{\text{hot,perm}}$ and $\Delta p_{\text{cold}} = \Delta p_{\text{cold,perm}}$. The required heat load is then lowered, and the hot side flow rate is decreased while maintaining the same number of employed heat exchangers, i.e. having a constant heat transfer area. Three different responses (control strategies) to this change in heat load requirement are discerned below. The discussion will be held using the effectiveness–NTU framework [10].

Part Load Operation: Constant Effectiveness. The return temperature requirement is unaltered, hence the required effectiveness is also unaltered, which is the minimum allowed effectiveness. Keeping the effectiveness unchanged can be achieved by increasing $C^* = C_{\text{hot}}/C_{\text{cold}}$, as indicated by the diamond symbol in Fig. 1. Since C_{hot} is given C_{cold} must be reduced.

We have

$$\frac{\text{NTU}_1}{\text{NTU}_0} = \left(\frac{C_1}{C_0}\right)_{\text{hot}}^{r-1} \tag{2}$$

The required fluid capacity ratio at constant effectiveness $(C_1^*)_\epsilon$ is then obtained by solving the ϵ–NTU relationship for the parameter C^*, i.e. solvnig $\epsilon = \epsilon(\text{NTU}; C^*)$ for C^* with $\epsilon = (\epsilon_1)_\epsilon = \epsilon_0$ thus obtaining $(C_1^*)_\epsilon$.

Operating at constant effectiveness will result in an increased rate of fouling since $\dot{m}_{\text{cold}}$ must be reduced. However, the cost for pumping will be lowered.

Part Load Operation: Constant Cold Side Flow Rate. Maintaining the cold side flow rate $\dot{m}_{\text{cold}}$ (or C_{cold}) will result in an increased effectiveness, shown in Fig. 1 with the triangle symbol. This will be the maximum possible effectiveness since Δp_{cold} remains at the permitted value (if hydraulic resistance of fouling is neglected). An increased effectiveness may or may not be problematic.

Here $C_{\text{cold},1} = C_{\text{cold},0}$. Thus,

$$(C_1^*)_{C_{\text{cold}}} = \left(\frac{C_{\text{hot}}}{C_{\text{cold}}}\right)_1 = \frac{C_{\text{hot},1}}{C_{\text{cold},0}} < C_0^* \tag{3}$$

The resulting effectiveness will then be obtained from the $\epsilon-$NTU relationship as $(\epsilon_1)_{C_{\text{cold}}} = \epsilon\big(\text{NTU}_1; (C_1^*)_{C_{\text{cold}}}\big)$.

In this case there is no increased rate of fouling, but the pumping power needs to remain high. Also, the higher effectiveness may or may not need to be counter-measured.

Part Load Operation: Constant Ratio of Hot and Cold Side Flow Rates. The cases of constant effectiveness and constant pressure drop are the extremes with respect to effectiveness (and cold side pressure drop). Any state in between is of course possible. Specifically, one may choose to keep C^* constant, indicated with the open circle in Fig. 1.

In this case, $(C_1^*)_{C^*} = C_0^*$, and $(\epsilon_1)_{C^*} = \epsilon(\text{NTU}_1; (C_1^*)_{C^*})$.

We also note that $\epsilon_{\text{reqd}} < (\epsilon_1)_{C^*} < (\epsilon_1)_{C_{\text{cold}}}$ and $(\Delta p_1)_\epsilon < (\Delta p_1)_{C^*} < \Delta p_{\text{perm}} \leq (\Delta p_1)_{C_{\text{cold}}}$.

Heat Load Distribution

We will assume that the heat load distribution $\phi(w)$ is known and is distributed in a way so that it can be represented by a probability density function [11]. The function is normalized, implying its integral from w_{min} to w_{max} is unity. $\phi(w)$ thus yields the fractional time x of the considered period spent in the required duty range w to $w + dw$, and will be used as a weighting function when computing the time-averaged penalties (see section Metrics below).

Heat Exchanger Station

The count of parallel heat exchangers in the station is N. The N heat exchangers each have a nominal heat exchange capacity $\hat{w}_j$, $j = 1, 2, \ldots, N$ which meets the nominal required duty, scaled only by the flow rate. This capacity – interpreted here as the heat load (or heat transfer area) it was sized for, thus meeting the required effectiveness and permitted pressure drop – should not be confused with the fluid capacity rate $C = \dot{m}c_p$. Each heat exchanger, irrespective of capacity (size), thus provides the same NTU at the nominal (maximum heat load) conditions. In the case of plate-and-frame heat exchangers, this is accomplished by configuring each heat exchanger with the appropriate combination of plate size, free channel, mix of channels with respect to corrugation angle, and plate counts. Each heat exchanger may thus be

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individually designed, independent of the other heat exchangers.

The reference case (or base case) will always be N identical heat exchangers, the commonest case in real situations. But in the general case, a station will have N different heat exchangers, each with an individual nominal capacity $\widehat{w}_j$. Note that the total nominal capacity $\sum_{j=1}^{N} \widehat{w}_j$ is the same irrespective of the distribution of the individual $\widehat{w}_j$'s.

For any case of capacity distribution in the station, the fractional capacity of heat exchanger j is

$$\hat{v}_j = \frac{\widehat{w}_j - w_{\min}}{w_{\max} - w_{\min}} \tag{4}$$

such that $\sum_{j=1}^{N} \hat{v}_j = 1$. In the base case, all $\hat{v}_j = 1/N$.

From N heat exchangers we can pick $\binom{N}{n}$ combinations of heat exchangers, given $1 \leq n \leq N$, the number of employed heat exchangers. Hence, there will all in all be $M = \sum_{k=1}^{N} \binom{N}{n_k}$ combinations. But of these only $1 \leq Q \leq M$ (discrete) combinations will be unique with respect to capacity. These Q unique combinations is the set of the (ordered) sets of combined capacities, each of q_k heat exchangers, i.e. $\widetilde{w}_k$ with $k = 1, 2, \dots, Q$ and $\widetilde{w}_{k-1} < \widetilde{w}_k$.

For a given required heat load w then, the combination of heat exchangers that needs to be employed will be the nearest larger $\widetilde{w}_k$, where

$$\widetilde{w}_{k-1} < w \leq \widetilde{w}_k \tag{5}$$

with $k = 1, 2, \dots, Q$

Metrics

We are interested in the implications of heat exchanger capacity distribution as well as cooling water control strategy. There are two metrics related to operation costs that are of primary concern regarding fouling on the cold side: cleaning frequency and pumping power consumption. Both are outlined below.

For a channel with constant cross-section in which an incompressible fluid flows we may set up the momentum balance

$$a_{\mathrm{w}} \tau_{\mathrm{w}} = a_{\mathrm{x}} \Delta p_f \tag{6}$$

where Δp_f is the irreversible pressure drop[3] over some channel length L. For a plate heat exchanger channel, $a_{\mathrm{x}}/a_{\mathrm{w}} = D_{\mathrm{h}}/(L\sigma)$, while for a straight tube, $a_{\mathrm{x}}/a_{\mathrm{w}} = D_{\mathrm{h}}/(4L)$. σ is the average surface enlargement factor for a pressed plate.

Penalty for Cleaning. For a given mass flow rate, the initial fouling rate is proportional to the clean wall shear stress. With a pragmatic approach of assuming that subsequent fouling rates, although changing, is related to the initial fouling rate, the characteristic time to a given degree of fouling is taken as related to the clean wall shear stress. Any effects of increasing fouling thickness, changing fouling characteristics (composition, density, etc.), and any effects of fouling on pressure drop and OHTC are thus included in the characteristic time.

According to Novak [12], time between cleanings for biological fouling varies as $\tau_{\mathrm{w}}^{0.6}$ to $\tau_{\mathrm{w}}^{0.9}$; here we use the former as this works in favor of equal heat exchanger capacity distribution. Hence, if cleaning takes place at some given degree of fouling, the number of cleaning occasions in a period varies as $\tau_{\mathrm{w}}^{-0.6}$. The cost for one cleaning, whether chemical or mechanical, is taken as dependent on the size of the heat exchanger. The combined capacity (size) S of the heat exchangers in one set $\widetilde{w}_k$ is $S(w) = \sum_{j=1}^{n_k} \widetilde{w}_{k,j}$, where $\widetilde{w}_k$ is given by Eq. (5). Hence, the relative cleaning cost κ_{C} for one set of employed heat exchangers is

$$\kappa_{\mathrm{C}} \propto S(w) \left(\frac{\tau_{\mathrm{w},1}}{\tau_{\mathrm{w},0}} \right)^{-0.6} \tag{7}$$

The initial irreversible pressure drop depends on the flow rate as $\Delta p_f \propto \dot{m}^{2+r_f} \propto C^{2+r_f}$. We thus have

$$\frac{\tau_{\mathrm{w},1}}{\tau_{\mathrm{w},0}} = \frac{\Delta p_{f,1}}{\Delta p_{f,0}} = \left(\frac{C_{\mathrm{cold},1}}{C_{\mathrm{cold},0}} \right)^{2+r_f} \tag{8}$$

which is approximately valid for any type of heat exchanger if channel geometry can be regarded as (essentially) unchanged. The averaged relative cleaning cost K_{C} for the considered period is then

$$\begin{aligned} K_{\mathrm{C}} &= \int_{w_{\min}}^{w_{\max}} \kappa_{\mathrm{C}}\, \phi(w)\, dw \\ &= \int_{w_{\min}}^{w_{\max}} S(w) \left(\frac{C_{\mathrm{cold},1}}{C_{\mathrm{cold},0}} \right)^{-0.6(2+r_f)} \phi(w)\, dw \end{aligned} \tag{9}$$

Note that also $C_{\mathrm{cold},1}/C_{\mathrm{cold},0}$ depends on w.

For plate heat exchangers, $-0.2 \lesssim r_f \lesssim +0.1$ in the fully turbulent regime, depending on the channel corrugations.

Penalty for Coolant Pumping with Insignificant Fouling Flow Resistance. If the hydraulic resistance of the fouling layer can be

[3] The irreversible pressure drop is due to both skin friction and form drag within the heat exchanger channel.

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neglected, the required power P (ignoring pump and motor efficiencies) for pumping a fluid is

$$P = \frac{\dot{m}}{\rho} \Delta p_f \propto C \Delta p_f \tag{10}$$

The relative cost of pumping $\kappa_\mathrm{P}^{(0)}$ is proportional to the pumping power, i.e.

$$\begin{aligned} \kappa_\mathrm{P}^{(0)} &= \frac{P_\mathrm{cold,1}}{P_\mathrm{cold,0}} = \frac{C_\mathrm{cold,1}}{C_\mathrm{cold,0}} \frac{\Delta p_{f,1}}{\Delta p_{f,0}} \\ &= \left(\frac{C_\mathrm{cold,1}}{C_\mathrm{cold,0}}\right)^{3+r_f} \end{aligned} \tag{11}$$

The superscript (0) denotes that the fouling flow resistance has been ignored. The averaged relative pumping cost $K_\mathrm{P}^{(0)}$ becomes

$$\begin{aligned} K_\mathrm{P}^{(0)} &= \int_{w_\mathrm{min}}^{w_\mathrm{max}} \kappa_\mathrm{P}^{(0)} \, \phi(w) \, dw \\ &= \int_{w_\mathrm{min}}^{w_\mathrm{max}} \left(\frac{C_\mathrm{cold,1}}{C_\mathrm{cold,0}}\right)^{3+r_f} \phi(w) \, dw \end{aligned} \tag{12}$$

Penalty for Coolant Pumping with Significant Fouling Flow Resistance. If the hydraulic resistance of fouling is significant, pumping penalty will increase more for low-flow cases. An approximate approach to include the fouling flow resistance is making the same assumption as above regarding the fouling growth, $\left(\tau_\mathrm{w,1}/\tau_\mathrm{w,0}\right)^{-0.6}$ in Eq. (7), and assume that the flow resistance change is proportional the fouling growth rate. The relative pumping power penalty including fouling resistance then becomes

$$\begin{aligned} \kappa_\mathrm{P}^{(\mathrm{f})} &= \kappa_\mathrm{P}^{(0)} \left(\frac{\tau_\mathrm{w,1}}{\tau_\mathrm{w,0}}\right)^{-0.6} \\ &= \left(\frac{C_\mathrm{cold,1}}{C_\mathrm{cold,0}}\right)^{3+r_f} \left(\frac{C_\mathrm{cold,1}}{C_\mathrm{cold,0}}\right)^{-0.6(2+r_f)} \\ &= \left(\frac{C_\mathrm{cold,1}}{C_\mathrm{cold,0}}\right)^{1.8+0.4r_f} \end{aligned} \tag{13}$$

The exponent of Eq. (13) is smaller than the one of Eq. (11). This implies that a high C_cold is less punishing if fouling has a significant flow resistance, as expected.

The averaged relative pumping cost $K_\mathrm{P}^{(\mathrm{f})}$ becomes

$$\begin{aligned} K_\mathrm{P}^{(\mathrm{f})} &= \int^{w_\mathrm{max}} \kappa_\mathrm{P}^{(\mathrm{f})} \, \phi(w) \, dw \\ &= \int_{w_\mathrm{min}}^{w_\mathrm{max}} \left(\frac{C_\mathrm{cold,1}}{C_\mathrm{cold,0}}\right)^{1.8+0.4r_f} \phi(w) \, dw \end{aligned} \tag{14}$$

Total Penalty. In order to compare the total penalty – cleaning and fouling – an assumption regarding the cost needs to be made. If the reference (base) situation – equal capacity distribution and constant coolant pressure drop control – is in some sense a cost-optimized one, the costs (penalties) for cleaning and pumping ought to be equal, or

$$K_\mathrm{C,ref} = z K_\mathrm{P,ref} \tag{15}$$

Hence $z = K_\mathrm{C,ref}/K_\mathrm{P,ref}$ is the penalty translation factor. The total penalty for any situation is thus $K_\mathrm{C} + zK_\mathrm{P}$ and the relative total penalty K^* is

$$\begin{aligned} K^* &= \frac{K_\mathrm{C} + zK_\mathrm{P}}{K_\mathrm{C,ref} + zK_\mathrm{P,ref}} \\ &= \frac{K_\mathrm{C}/K_\mathrm{C,ref} + K_\mathrm{P}/K_\mathrm{P,ref}}{2} \end{aligned} \tag{16}$$

EXAMPLE CASES

Two heat load distribution cases are studied, A and B. Case A is a unimodal distribution with the mode at around 70 % of the maximum heat load. Case B is a bimodal case with the modes at approximately 25 % and 95 % of the maximum heat load. The distributions of cases A and B are shown in Fig. 2.

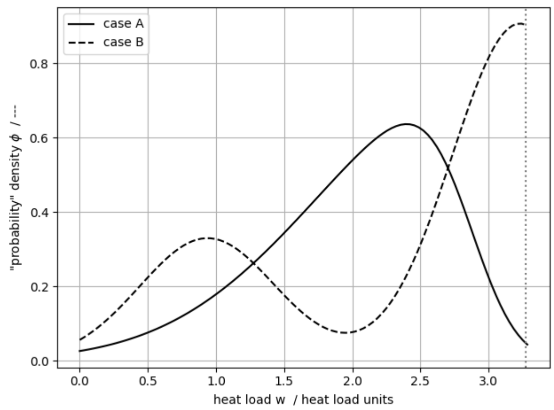

Fig. 2 Heat load distributions $\phi(w)$ for the two cases.

All three coolant control strategies described above are considered. For the heat exchangers in operation at a given cooling requirement, it is assumed that the total flow is evenly distributed in the sense that the flow rate of each heat exchanger relative to its design value is the same.[4]

[4] Having all but one of the employed heat exchangers at nominal duty, and the last one at part load is another control approach. This approach is, however, out of the present scope.

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Three different heat exchanger capacity distributions were used: “equal”, “doubled”, and “hybrid”, listed in Table 1. The “equal” case is also referred to as the base case. The “hybrid” case can be regarded as a mixture of “equal” and “doubled”, with both differently sized heat exchangers and redundancy.

Table 1 Heat exchanger capacity distributions

name	distrib.	$\hat{\mathrm{v}}$	Q
equal	1:1:1:1	{0.25, 0.25, 0.25, 0.25}	4
doubled	1:2:4:8	{0.067, 0.133, 0.267, 0.533}	15
hybrid	1:1:2:2	{0.167, 0.167, 0.333, 0.333}	6

RESULT

The computed averaged cleaning and pumping costs (penalties) for cases A and B are shown in Table 2. The results of the two cases are not qualitatively different. Equal distribution of heat exchanger capacity has a higher cleaning penalty level than the other two distributions, but a lower pumping penalty level for both cases. Within each distribution type, the constant C_{cold} control strategy has the lowest cleaning penalty and the highest pumping penalty, as expected. Here, $K_{\mathrm{P}}^{(0)}$ represent the lower limit of pumping penalties. Note that $K_{\mathrm{P}}^{(0)}$ and $K_{\mathrm{P}}^{(\mathrm{f})}$ cannot be directly compared to each other.

Table 3 and Table 4 list the relative penalties – having as reference the base case (equal capacity distribution and constant coolant flow rate; first entry line in Table 2) – for the cases of insignificant and significant fouling flow resistance, respectively. Equal capacity distribution has the highest total penalty for both case A and case B. The lowest is found for the doubled distribution. Within each distribution type, the constant C_{cold} control strategy has the highest total penalty while the lowest is found for the constant effectiveness strategy. Specifically, one may note that the cleaning penalty K_{C} for the doubled distribution is 30–42 % lower than that of the equal distribution.

For a graphical overview, Fig. 3 shows the relative total penalties with significant fouling flow resistance in matrix form. It is clear that capacity distribution, rather than control strategy, contributes most to the reduction in total penalty.

The largest penalty reduction, approximately 20 %, is found for doubled distribution and constant effectiveness in case A (insignificant fouling flow resistance).

As noted above, there is not a large difference in the result of cases A and B, but the result suggests that the benefit of a double distribution is overall somewhat higher for case A.

Table 2 Cleaning and pumping penalties for cases A and B

		case A			case B		
distribution	control	K_{C}	$K_{\mathrm{P}}^{(0)}$	$K_{\mathrm{P}}^{(\mathrm{f})}$	K_{C}	$K_{\mathrm{P}}^{(0)}$	$K_{\mathrm{P}}^{(\mathrm{f})}$
equal	const. C_{cold}	9.41	1.00	1.00	10.07	1.00	1.00
	const. C^*	11.35	0.65	0.76	11.58	0.68	0.78
	const. ϵ	11.85	0.59	0.72	11.97	0.63	0.74
hybrid	const. C_{cold}	7.51	1.00	1.00	9.01	1.00	1.00
	const. C^*	8.46	0.74	0.83	9.88	0.75	0.83
	const. ϵ	8.71	0.69	0.79	10.10	0.71	0.80
doubled	const. C_{cold}	6.46	1.00	1.00	7.91	1.00	1.00
	const. C^*	6.79	0.87	0.92	8.25	0.88	0.92
	const. ϵ	6.88	0.85	0.90	8.33	0.85	0.90

Table 3 Relative penalties of cleaning and pumping with insignificant fouling flow resistance, and the total penalty for cases A and B

		case A			case B		
distribution	control	$K_{\mathrm{C}}/K_{\mathrm{C,ref}}$	$K_{\mathrm{P}}^{(0)}/K_{\mathrm{P,ref}}^{(0)}$	$K^{*(0)}$	$K_{\mathrm{C}}/K_{\mathrm{C,ref}}$	$K_{\mathrm{P}}^{(0)}/K_{\mathrm{P,ref}}^{(0)}$	$K^{*(0)}$
equal	const. C_{cold}	1.00	1.00	1.00	1.00	1.00	1.00
	const. C^*	1.21	0.65	0.93	1.15	0.68	0.91
	const. ϵ	1.26	0.59	0.92	1.19	0.63	0.91
hybrid	const. C_{cold}	0.80	1.00	0.90	0.89	1.00	0.95
	const. C^*	0.90	0.74	0.82	0.98	0.75	0.87
	const. ϵ	0.93	0.69	0.81	1.00	0.71	0.86
doubled	const. C_{cold}	0.69	1.00	0.84	0.79	1.00	0.89
	const. C^*	0.72	0.87	0.80	0.82	0.88	0.85
	const. ϵ	0.73	0.85	0.79	0.83	0.85	0.84

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Table 4 Relative penalties of cleaning and pumping with significant fouling flow resistance, and the total penalty for cases A and B

		case A			case B		
distribution	control	$K_\mathrm{C}/K_\mathrm{C,ref}$	$K_\mathrm{P}^{(\mathrm{f})}/K_\mathrm{P,ref}^{(\mathrm{f})}$	$K^{*(\mathrm{f})}$	$K_\mathrm{C}/K_\mathrm{C,ref}$	$K_\mathrm{P}^{(\mathrm{f})}/K_\mathrm{P,ref}^{(\mathrm{f})}$	$K^{*(\mathrm{f})}$
equal	const. C_cold	1.00	1.00	1.00	1.00	1.00	1.00
	const. C^*	1.21	0.76	0.98	1.15	0.78	0.96
	const. ϵ	1.26	0.72	0.99	1.19	0.74	0.96
hybrid	const. C_cold	0.80	1.00	0.90	0.89	1.00	0.95
	const. C^*	0.90	0.83	0.86	0.98	0.83	0.91
	const. ϵ	0.93	0.79	0.86	1.00	0.80	0.90
doubled	const. C_cold	0.69	1.00	0.84	0.79	1.00	0.89
	const. C^*	0.72	0.92	0.82	0.82	0.92	0.87
	const. ϵ	0.73	0.90	0.82	0.83	0.90	0.86

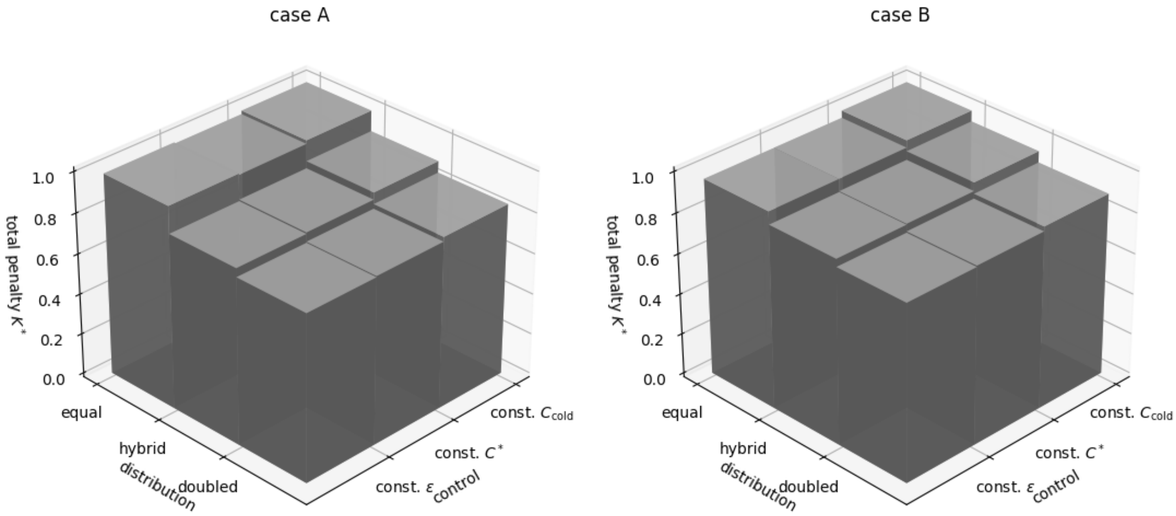

Fig. 3 Relative total penalties with significant fouling flow resistance for cases A (left) and B (right).

CONCLUSION

Different ways of distributing the heat exchanger capacities of a secondary cooling station were discussed with a focus on the time-averaged costs of cleaning and pumping for some example cases. An idealized model has been set up for simulating the impact of various distributions of heat exchanger capacity in a system with fouling on the coolant side. The influence of part load coolant control strategy was included in the model. Two simplified, but realistic, cooling load temporal distributions were employed.

The result implies that it is possible to lower the combined cleaning and pumping power cost (total penalty) if heat exchanger capacity is distributed in an unequal fashion. Maintaining the coolant design flow rate for part loads will keep fouling at a minimum. However, the present study indicates that the control strategy of constant coolant flow rate incurs the highest total penalty, irrespective of heat exchanger capacity distribution.

Reductions of approximately 15–20 %, of the total penalty was obtained for the doubled distribution and constant effectiveness control when compared to the base case (equal distribution and constant pressure drop control). Cleaning penalty reductions of up to approximately 40 % were obtained with the doubled distribution and constant effectiveness control strategy.

NOMENCLATURE

Roman

a area, m^2
A heat transfer area, m^2
c_p heat capacity, $\mathrm{J/(kg\ K)}$
C fluid capacity rate, $\mathrm{W/K}$
C^* ratio of minimum fluid capacity rate to maximum fluid capacity rate, —
D_h hydraulic diameter, m
k overall heat transfer coefficient, $\mathrm{W/(m^2\ K)}$
K period-averaged relative cost, —
K^* relative total cost (penalty), —

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L channel length, m
$\dot{m}$ mass flow rate, $\mathrm{kg/s}$
n count of employed heat exchangers, —
M count of all possible capacities, given the count of employed heat exchangers, —
N count of available heat exchangers, —
P pumping power, W
Q count of unique capacities, given the count of employed heat exchangers, —
r exponent in thermohydraulic relations, —
S combined capacity (size) of employed heat exchangers, W or m^2
$\hat{v}$ fractional nominal capacity of heat exchanger, —
$\hat{\mathrm{v}}$ set of unique combinations of employed fractional nominal capacities, —
w required capacity, W or m^2
$\hat{w}$ nominal capacity of heat exchanger, W or m^2
$\widetilde{\mathrm{w}}$ set of unique combinations of employed nominal capacities, W or m^2
x fractional time, —

Greek

α heat transfer coefficient, $\mathrm{W/(m^2\ K)}$
Δp pressure drop, Pa
ϵ heat exchanger effectiveness, —
ϕ normalized load distribution
κ relative cost, —
ρ mass density, $\mathrm{kg/m^3}$
σ average surface enlargement factor, —
$\tau_{\square}$ shear stress, Pa

Subscripts

0 state before a change in conditions
1 state after a change in conditions
C cleaning
cold cold side of heat exchanger
f flow irreversibilities
hot hot side of heat exchanger
j index for heat exchanger
k index for heat exchanger combination
max maximum
min minimum
P pumping
perm permitted
ref reference
reqd required
w channel wall
x channel cross-section

Superscripts

(0) insignificant fouling flow resistance
(f) significant fouling flow resistance

Acronyms

IT information technology
HTC heat transfer coefficient
NTU number of transfer units
OHTC overall heat transfer coefficient
PUE power usage effectiveness

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