



Bypass, Loss, and Heat Transfer in Aircraft Surface Coolers

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Surface coolers are heat exchangers with fins on the air side. When air approaches the fins, a portion is diverted away (bypass) because of the adverse pressure gradients induced by the fins. Also, for the air that does flow between the fins, a portion exits (loss) because of pressure rise along the fins due to friction. Both bypass and loss reduce the effectiveness of surface coolers to transfer heat to the air. In this study, steady RANS with the SST model (with and without conjugate heat transfer) were performed to examine how geometric and operating parameters affect bypass, loss, pressure drop, and heat transfer in two surface coolers commonly used in aircraft applications. Of the surface coolers, one has continuous fins, and the other has staggered or non-staggered segmented fins. Geometric parameter examined include: spacing between the fins ($S/H = 0.2, 0.4, 0.6, 0.8$), thickness of the fins ($t/H = 0.1, 0.2, 0.4$), length of the fins ($L/H = 1, 5, 10$), and the height of the channel, where the surface cooler is placed ($C/H = 2.5, 5, 10, 20, 40$ cm), where H is the height of the fin, and C is the half the height of the channel. Operating parameters examined include: velocity ($V_{in} = 32.5, 65, 97.5$, and 135 m/s) and temperature ($T_{in} = 300$ and 473 K) of flow approaching the surface cooler, the fins' wall temperature ($T_w = 300, 320, 350, 375, 400, 493$ K). Results obtained show C/H to significantly affect bypass and loss until C/H reaches about 20. Bypass, loss, and pressure drop all increase monotonically as the blockage created by the fins, $t/(S + t)$, increases. The ratio of the Nusselt number to the pressure coefficient is a maximum when $t/(S + t) = 0.33$ for the conjugate cases and 0.5 for the isothermal cases. V_{in} , T_{in} , and T_w were found to have negligible effects on bypass, but have appreciable effects on loss when spacing between the fins is small. For the geometries studied, segmenting the fins was found to increase loss, resulting in the worst heat-transfer rate and highest pressure drop.

Keywords: convective heat transfer, heat transfer, heat exchanger, surface cooler, fins, pressure loss

INTRODUCTION

Aircraft have many components and fluids that need to be cooled. These include any surface that is exposed to the "hot-gas path" in a gas turbine engine as well as the coolants used to cool avionics and the oils used to lubricate the gears/bearings. There are three major sources available for cooling. The first is the air that enters the engine, which can be quite cold during cruise at high altitudes, even if extracted after the fan. The second is the fuel. However, the fuel can only absorb a limited

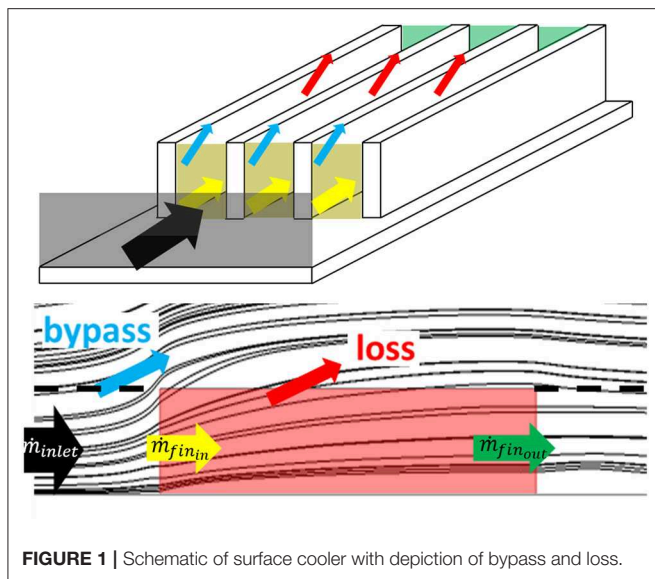


FIGURE 1 | Schematic of surface cooler with depiction of bypass and loss.

amount of thermal energy because its temperature must be kept below where coking starts unless fuel underwent deoxygenation. The third is using a refrigeration cycle.

All aforementioned sources require heat exchangers. One type of heat exchanger widely used to cool oils that lubricate engine gears and bearings on aircraft is the surface cooler; see **Figure 1**. As shown in **Figure 1**, surface coolers use the cooler air that enters the engine and has fins that protrude into the cooler air.

The literature on heat exchangers with fins is quite extensive (see e.g., Kays and London, 1964; Afgan and Schlunder, 1974; Webb, 1994; Rohsenow et al., 1997; Nagarani et al., 2014). Work has been done to explore different fins shapes (Kays and London, 1964; Afgan and Schlunder, 1974; Metzger et al., 1984; Sparrow and Grannis, 1991; Webb, 1994; Chen et al., 1997; Rohsenow et al., 1997; Nagarani et al., 2014) and their arrangements (Kays and London, 1964; Afgan and Schlunder, 1974; Metzger et al., 1984; Sparrow and Grannis, 1991; Webb, 1994; Rohsenow et al., 1997; Nagarani et al., 2014). Work has also been done to optimize fin shapes (Metzger et al., 1984; Sparrow and Grannis, 1991; Chen et al., 1997) and arrangement of fins for a wide range of operating conditions (Sparrow and Beckey, 1981; Metzger et al., 1984; Sparrow and Grannis, 1991; Wirtz et al., 1994; Lee, 1995; Sata et al., 1996; Chen et al., 1997; Jonsson and Palm, 1998). Most of the literature on surface coolers are connected to the cooling of electronics, where the flow speeds are typically low ($M \ll 1$ and $Re < 5,000$), and the clearance between the fin tip and the channel is small (typically no more than the height of the fin). If there is a clearance, then flow can “bypass” the fins (i.e., flow over the fins instead of through the fins). With less flow through the fins, heat transfer from fins are greatly reduced. Sparrow and Beckey (1981) studied the effects of spacings between fins and clearances. They showed that the frictional loss between the fins is relatively unaffected by the clearance if it is less than the fin height. Wirtz et al. (1994) studied the effects of fin height and spacing on heat transfer. Though bypass was mentioned, it was not quantified. Jonsson and Palm (1998) developed correlations

to predict the thermal resistance and pressure drop for heat sinks with bypass, recognizing the role of fin height and duct aspect ratio. Lee (1995), Sata et al. (1996), Simons and Schmidt (1997), Khan et al. (2008) developed models to predict the amount of bypass. However, these models require experimental input to predict the impact of bypass for each heat sink design. In all of the aforementioned studies, bypass and the flow through the fins were treated together via a pressure drop and an effective discharge coefficient. Bypass was never quantified.

The literature on surface coolers for aircraft applications is considerably less. Surface coolers for aircraft applications differ from surface coolers for electronic cooling in two ways. First, flow rates in electronic cooling are low, whereas the flow rates for aircraft applications can be quite high (Mach number can approach unity). Second, the clearance for electronic cooling is typically quite low (less than the fin height), whereas it can be 20 or more times the fin height in aircraft applications. When the clearance is small, most of the flow is guaranteed to enter between the space between fins. However, when the clearance is large, considerable amount of the flow approaching the surface cooler could be diverted away or bypass it. Furthermore, a portion of the flow that enters the space between the fins will leave the space between the fins before reaching the end of the surface cooler. This flow that leaves the space between the fins is referred to as “loss.”

For surface coolers in aircraft applications, Villafañe et al. (2011) developed a transonic wind tunnel to reproduce aero-engine flow conditions with $M = 0.7$. Surface coolers with both continuous and segmented fins were studied (Sousa et al., 2014; Villafañe and Paniagua, 2018). Results were obtained for heat transfer and how fin arrays disturbed the flow. Bypass and loss were not measured, so their effects on heat transfer were not studied. Kim et al. (2014) performed an experimental and computational study on surface coolers for aircraft operating at $M = 0.6$. They measured bypass and the flow rates through the spacings between the fins with focus on heat transfer for one configuration under three operating condition: ground idle, cruise, and flight idle. Kim et al. (2016) studied the effects of segmenting fins on heat transfer by comparing a continuous fin with a segmented fin. They examined the effects of pitch between the segmented fins in the streamwise and spanwise directions. In their study, they showed that mass flow rate through the spacing between the fins decreases along the length of the fin, which is the first reference to loss. Kim et al. (2018) studied the effects of slanting one continuous and one segmented fin configuration with clearance up to 30 times the fin height on frictional losses and heat transfer.

As noted, few studies have been performed on surface coolers for aircraft applications. Though two studies to date have shown that bypass and loss can occur in surface coolers and that they affect frictional losses and heat transfer, bypass and loss were not quantified. More importantly, the detailed fluid mechanics of bypass and loss and how they affect heat transfer have not been examined. For aircraft applications, bypass and loss can be significant. The objective of this paper is to use computational methods to study how geometry and operating conditions affect bypass and loss in surface coolers for aircraft applications.

Since bypass and loss reduce the mass-flow rate of air through the fins for heat transfer, bypass and loss need to be quantified. In this study, bypass and loss are defined as follows:

$$bypass = \frac{\dot{m}_{inlet} - \dot{m}_{finin}}{\dot{m}_{inlet}} \quad (1)$$

$$loss = \frac{\dot{m}_{finin} - \dot{m}_{finout}}{\dot{m}_{fininlet}} \quad (2)$$

where \dot{m}_{inlet} is the mass-flow rate of the air intercepted by the frontal cross-sectional area of the surface cooler with the fins far

upstream of it; \dot{m}_{finin} is the mass-flow rate that enters the surface cooler at its leading edge; and \dot{m}_{finout} is the mass-flow rate that exits the surface cooler at its trailing edge (see **Figure 1**).

DESCRIPTION OF THE PROBLEM

In this study, two surface coolers widely used in aircraft applications were studied (**Figure 2**). Both surface coolers have fins of height $H = 1$ cm and thickness t , but one has fins that extend across the entire surface cooler (referred to as continuous

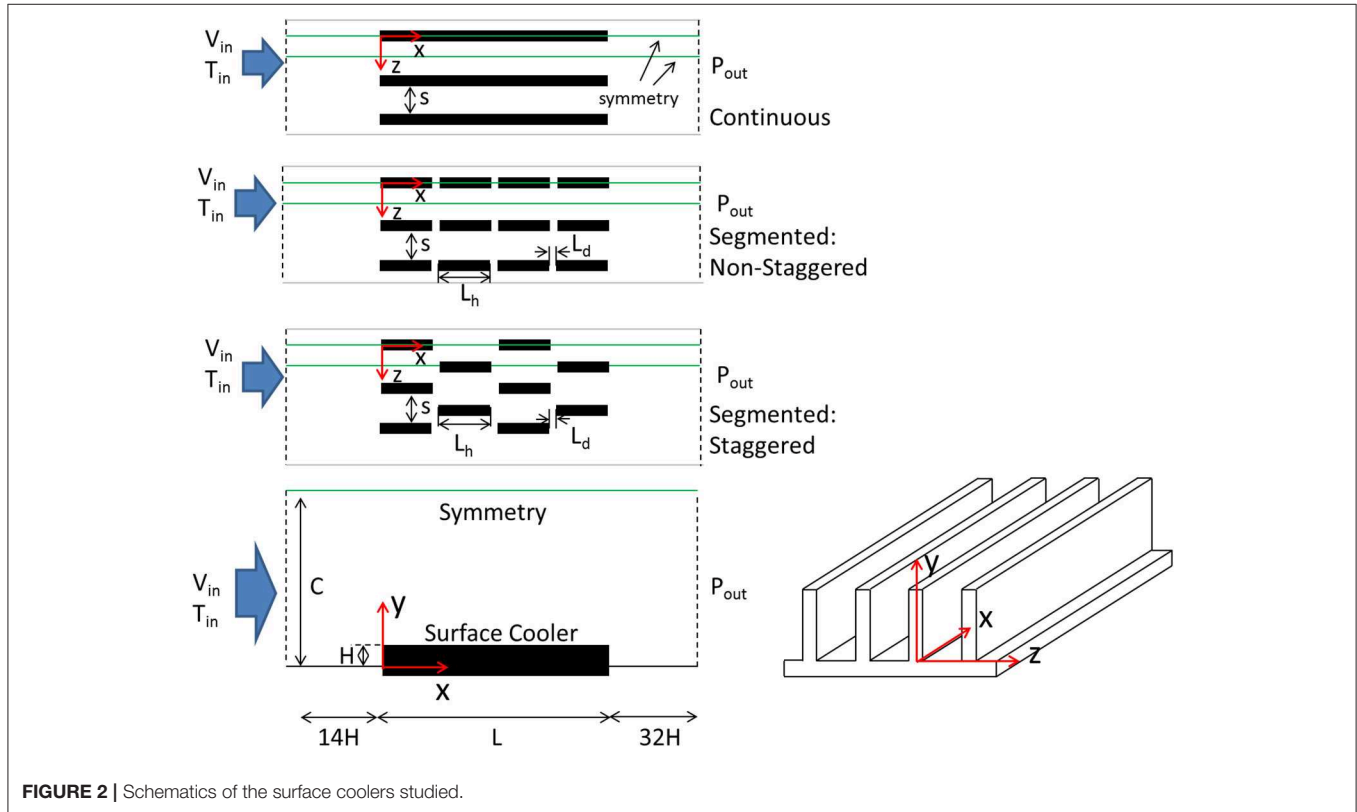


FIGURE 2 | Schematics of the surface coolers studied.

TABLE 1 | Summary of cases studied⁺.

Case(s)	S/H	t/H	T _{in}	ΔT (K)	V _{in} (m/s)	L/H	C/H
1, 2, 3, 4*, 5, 6	0.4	0.2	473	20	135	10	2.5, 5, 10, 20, 30, 40
7*, 8*, 9*	0.2, 0.6, 0.8	0.2	473	20	135	10	20
10, 11	0.2	0.1, 0.4	473	20	135	10	20
12, 13	0.2	0.4	473	20	135	1, 5	20
14, 15, 16, 17	0.2	0.2	300	20	32.5, 65, 97.5, 135	10	20
18, 19, 20, 21	0.2	0.2	300	0, 50, 75, 100	32.5	10	20
22, 23, 24, 25	0.4	0.2	300	20	32.5, 65, 97.5, 135	10	20
26, 27, 28, 29	0.4	0.2	300	0, 50, 75, 100	32.5	10	20
30, 31	0.6	0.2	300	20, 100	32.5	10	20
32, 33	0.8	0.2	300	20, 100	32.5	10	20
34	0.6	0.2	300	20	135	10	20
35	0.8	0.2	300	20	135	10	20
Non-staggered	0.4	0.2	473	20	135	10	20
Staggered	0.4	0.2	473	20	135	10	20

⁺H = 1 cm, P_{out} = 2 atm, *isothermal wall and conjugate heat transfer.

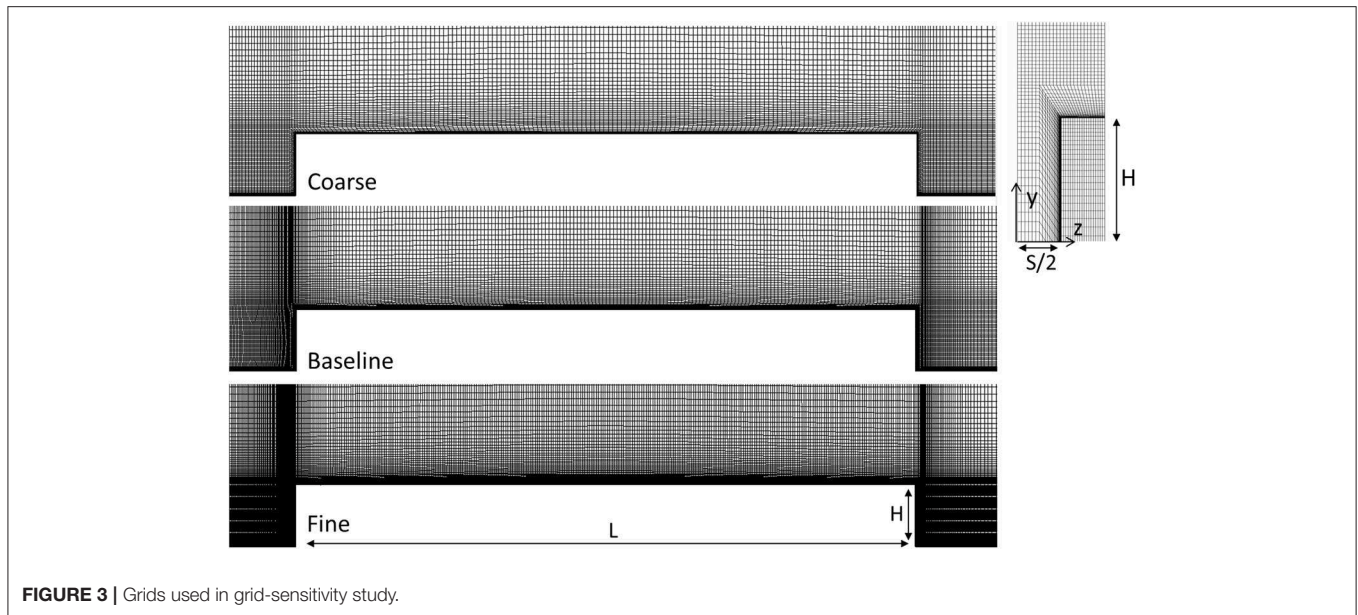


FIGURE 3 | Grids used in grid-sensitivity study.

TABLE 2 | Grid-sensitivity study.

# of grid points across	Coarse	Baseline	Fine
S/2	31	42	85
H	75	75	120
L	150	200	300
# Nodes (Million)	1.731	2.695	11.586
Bypass	0.487	0.485	0.485
Loss	0.449	0.454	0.458
Q (W)	8.9	7.87	7.81
Avg q'' (W/m ²)	7,040	7,020	6,970
$P_{in}-P_{out}$	883	883	883

pins), and the other has a series of fins along the surface cooler arranged in either a staggered or non-staggered fashion (referred to as segmented fins). Also, both surface coolers have length L and are located $14H$ from the inlet and $32H$ from the outlet. For both surface coolers, they are taken to be planar (no curvature). Note that the surface at $y = C$ is a symmetry plane so that the distance between surface coolers in y direction (clearance) is $2(C-H)$.

For surface coolers with continuous fins, the following parameters were examined: spacing between fins ($S/H = 0.2, 0.4, 0.6, 0.8$), thickness of fins ($t/H = 0.1, 0.2, 0.4$), length of fins ($L/H = 1, 5, 10$), and clearance ($C/H = 2.5, 5, 10, 20, 30, 40$). For surface coolers with segmented fins, the length of the fins (L_h/H) is 1, and the distance between the segmented fins in the x -direction (L_d/H) is 0.5.

The flow that approaches the surface cooler at $x = -14H$ is air and has uniform velocity of V_{in} in the x -direction ($V_{in} = 32.5, 64, 96, \text{ and } 135 \text{ m/s}$) and uniform temperature of T_{in} ($T_{in} = 300, 473 \text{ K}$). The pressure at the outlet ($x = 42H$) is $P_b = 2 \text{ atm}$. The surface at $y = 0$ is adiabatic, whereas all fin surfaces are

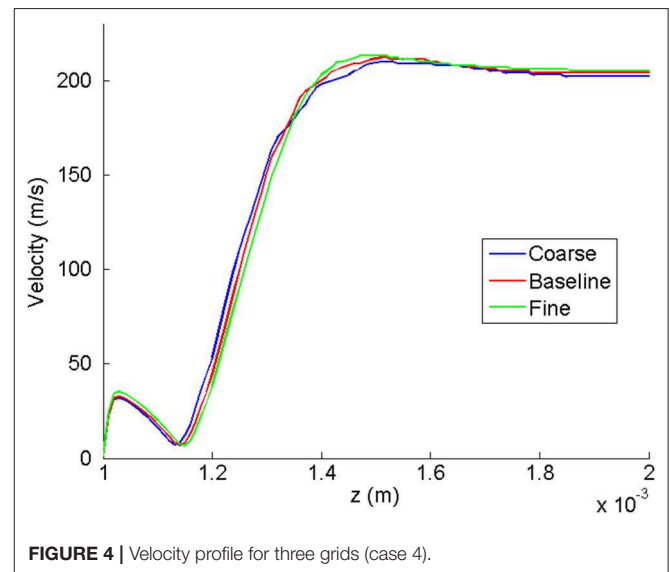


FIGURE 4 | Velocity profile for three grids (case 4).

maintained at a constant temperature, T_w , given by:

$$T_w = T_{in} + \Delta T \quad (3)$$

where ΔT is either 0, 20, 50, 75, or 100K.

To examine if bypass and loss could be affected by conjugate heat transfer, a variation of the problem was studied, where conjugate heat transfer was accounted for, where the surface cooler is made of aluminum. For this variation, the following surface are adiabatic: $y = 0$ and $x \in [-14H, 0]$ and $[10H, 42H]$; $x = 0$ and $y \in [-0.2H, 0]$; and $x = L$ and $y \in [-0.2H, 0]$. The surface at $y = -0.2H$ and $x \in [0, 10H]$ is maintained at a constant temperature, T_w .

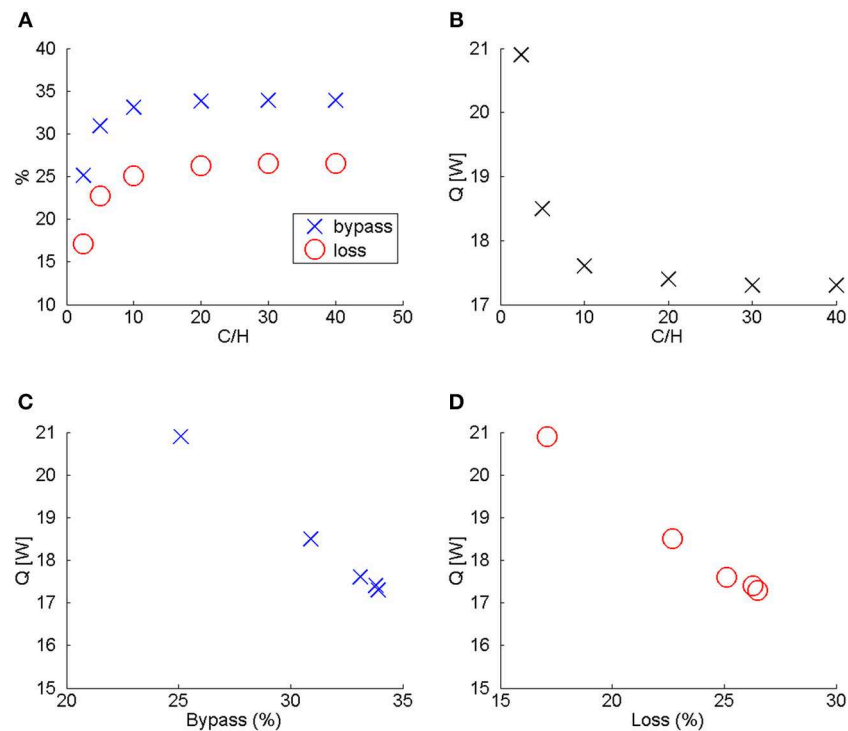


FIGURE 5 | Effects of C/H on bypass, loss, and heat-transfer rate (Cases 1-6). **(A)** Bypass and loss as a function of C/H. **(B)** Heat-transfer rate (Q) as a function of C/H. **(C)** Heat-transfer rate (Q) as a function of bypass. **(D)** Heat-transfer rate (Q) as a function of loss.

Table 1 summarizes all simulations performed to study the effects of geometry and operating conditions on bypass, loss, pressure drop, and heat transfer.

PROBLEM FORMULATION AND NUMERICAL METHOD OF SOLUTION

The air that flows over and through the surface-cooler described in the previous section was modeled by the time-averaged continuity, compressible Navier-Stokes, and total energy equations (RANS) for thermally perfect gas with temperature dependent properties. The Reynolds stresses that result from the time averaging was closed by the shear-stress transport (SST) model of Menter (1994). In this study, the turbulence is assumed to occur from the leading edge of the fin because the laminar region is $<10\%$ of L for the worst-case scenario, where by-pass transition is assumed to take place.

At the inflow boundary ($x = -14H$), uniform velocity and temperature were specified. At the outflow boundary ($x = L + 32H$), the pressure P_b was specified. All walls are no-slip. Also, all walls leading up to and after the surface cooler are adiabatic. For studies where conjugate heat transfer is not accounted for, all walls at $y = 0$ are adiabatic, and all fin surfaces are maintained at T_w . For simulations that account for conjugate heat transfer, the temperature at $y = -0.2H$ is maintained constant at T_w so the wall at $y = 0$ is no longer adiabatic from $x = 0$ to $x = L$. Symmetry was invoked wherever possible as shown in **Figure 2**, namely, at

$y = C$, $z = 0$, and $z = (S + t)/2$. At symmetry planes, symmetry boundary conditions were imposed.

For the conjugate studies, the solid aluminum that make up the fins and the plate of the surface cooler was modeled by the thermal energy equation and the Fourier law of conduction with temperature-dependent thermal conductivity. The boundary conditions imposed were isothermal wall at $y = -0.2H$ and $x \in [0, 10H]$ and adiabatic wall at $x = 0$ and $y \in [-0.2H, 0]$ and at $x = L$ and $y \in [-0.2H, 0]$. At all interfaces where the solid and the air meet, temperature and heat flux are the same for the solid and the air.

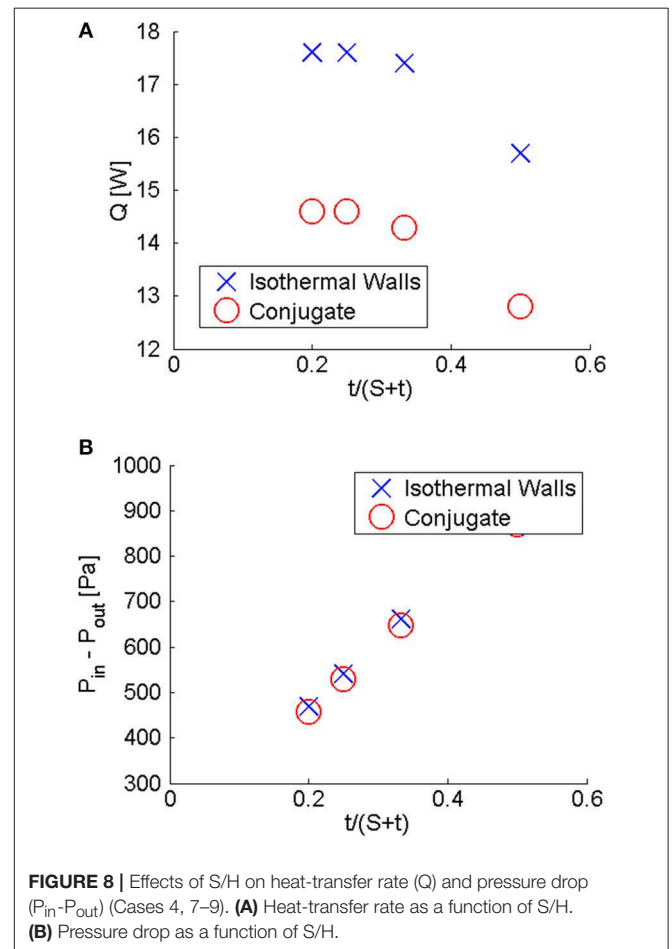
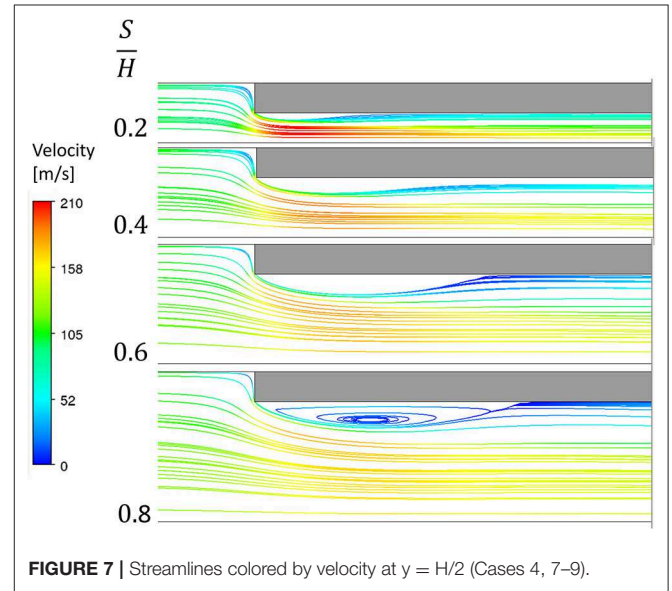
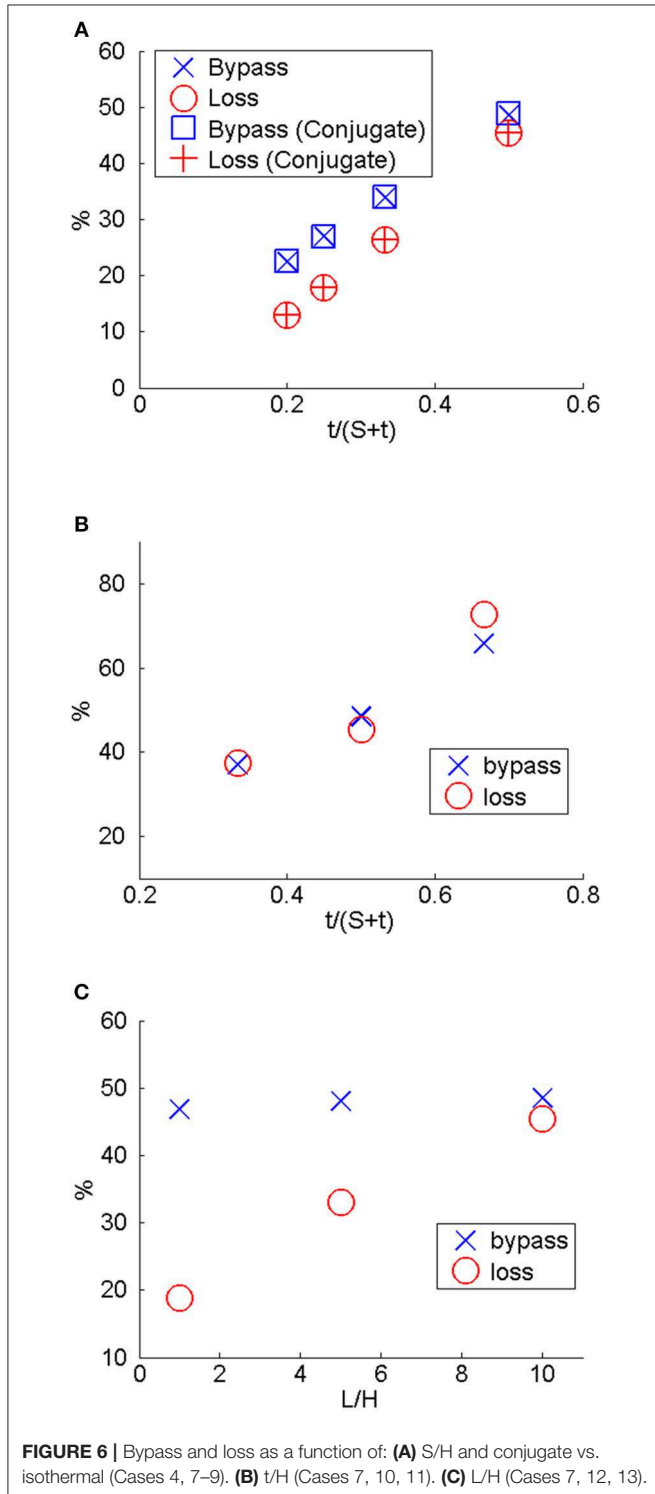
Solutions were obtained by using Version 17.1 of the ANSYS CFX software¹. For the equations governing the air phase, fourth-order Rhie-Chow interpolation was used to convert staggered to non-staggered for pressure-velocity coupling. Second-order upwind differencing was used for advection terms, and second-order central differencing was used for all diffusion terms. For the equations that govern the solid phase, second-order central differencing was used. Since only steady-state solutions were of interest, iterations were continued until the residuals plateaued. At convergence, residuals are $<10^{-5}$, 10^{-5} , 10^{-6} , and 10^{-5} for continuity, momentum, energy, and turbulence, respectively.

All grids used were structured and consisted of hexahedral elements. The total number of grid points ranged from 1.8 to 3

¹CFX Version 17.1 ANSYS CFX-Solver Theory Guide, Ansys, Inc. (2016).

million, depending on the geometry. Also, y^+ of all grid points next to walls is less than unity. A grid-sensitivity study was performed to ensure that the grid employed to generate solutions has sufficient resolution to produce meaningful results. Case 7 in **Table 1** was chosen for this study because this case has the highest

maximum velocity. **Figure 3** shows the three grids employed. **Table 2** gives the number of grid points between fins (S), across the fin height (H), and over the fin length (L). This table also gives



the total number of grid points employed, the computed bypass, loss, heat-transfer rate, average heat flux (q''), and difference between inlet and outlet pressure. **Figure 4** shows the velocity profile as a function of z at ($x = 0.01L$, $y = H/2$), generated by using the three grids. From **Table 2** and **Figure 4**, it can be seen that the baseline grid with 2.695 million nodes and fine grid with 11.586 million nodes yielded results that differ by $<1\%$. Thus, the baseline grid was used for the rest of the study.

RESULTS

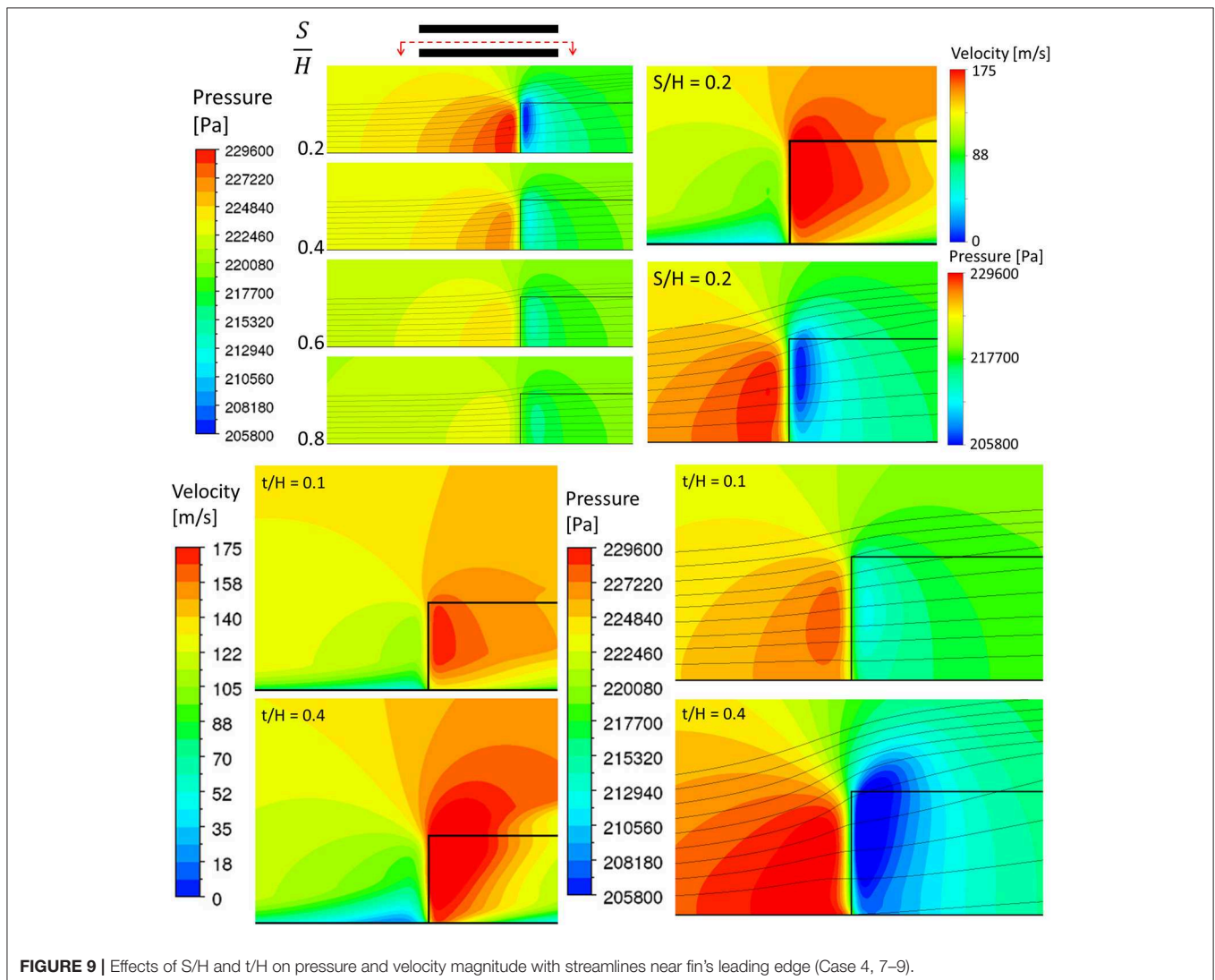
In this section, the results obtained to show how geometry and operating conditions affect bypass, loss, pressure drop, and heat transfer on surface coolers with continuous and segmented fins are described.

Continuous Fins

This section describes results obtained for surface coolers with continuous fins.

Figure 5 shows bypass, loss, and heat-transfer rate as a function of C/H . From **Figure 5A**, it can be seen that bypass and loss increase as the height of the channel increases. However, beyond $C/H \approx 20$, further increases in C has minimal effect on bypass, loss ($<3\%$). The behavior of the heat-transfer rate correlates with bypass and loss. When bypass and loss are low, heat-transfer rate is high, and when bypass and loss are high, heat-transfer rate is low (**Figures 5C,D**). Also, increasing C/H beyond 20 results in no additional decrease in the heat-transfer rate. Since bypass, loss, and heat-transfer rate are essentially independent of C/H when $C/H \geq 20$, C/H was henceforth set equal to 20 for the rest of the studies. Note that since $y = C$ is a symmetry plane, $C/H = 20$ implies the clearance must be $2(C-H) = 2H(C/H-1) = 2H(19) = 38H$ to have no effects from the flow on the opposite wall. This condition is typically satisfied in aircraft applications.

To understand the effects of C/H , start with $C/H = 1$. When $C/H = 1$, there is no bypass or loss since the surface coolers on the two opposite walls will be touching each other. If $C/H > 1$, then

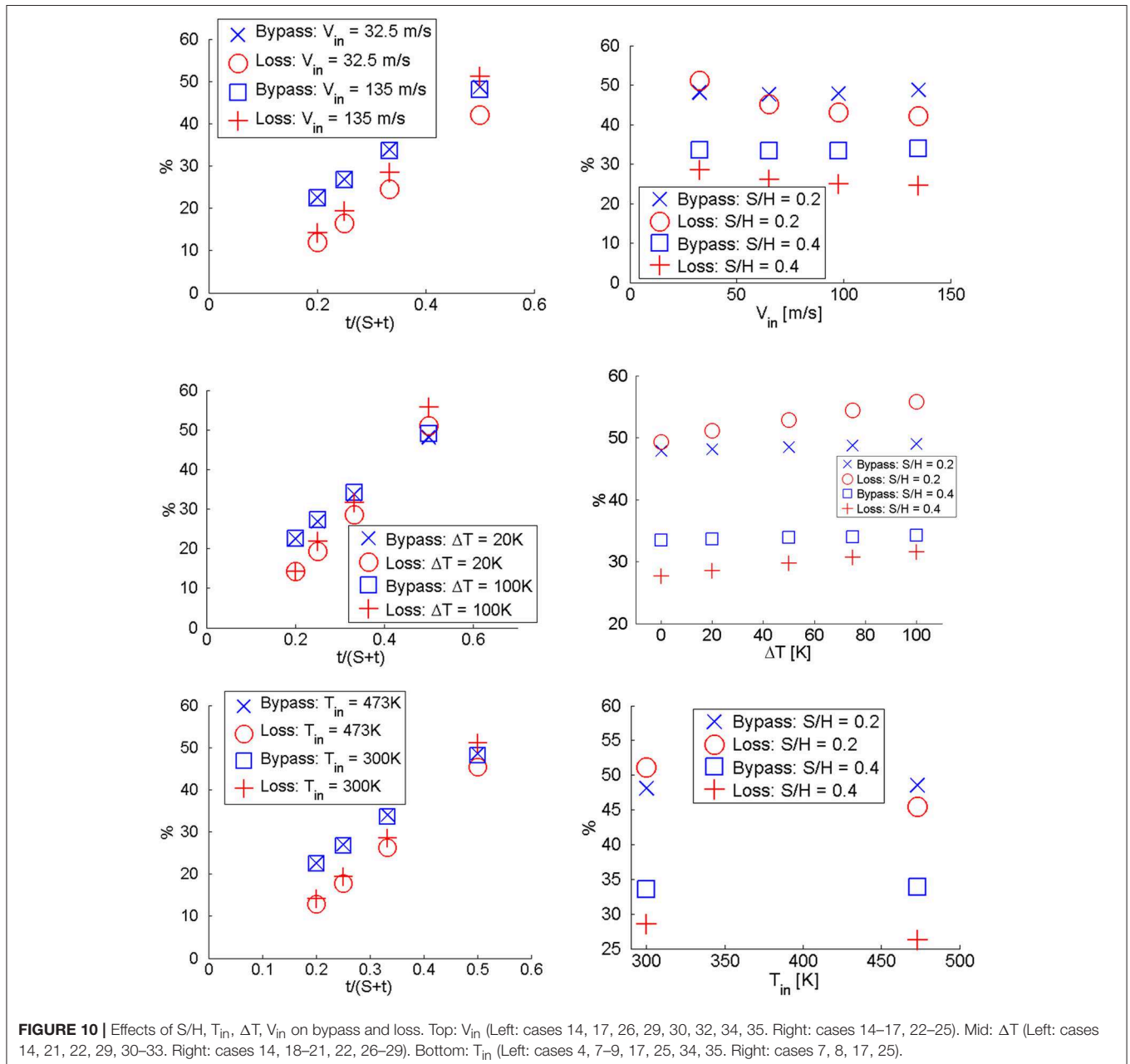


there will be space between the two surface coolers for bypass and loss to take place. When C/H is greater than unity but less than some critical value (≈ 20 for the configurations studied), the two surface coolers still affect each other and impedes bypass and loss. However, once C/H exceeds about 20, the effects of the displacement thicknesses created about each surface cooler no longer affect the bypass and loss on the other surface cooler.

Figure 6A shows that bypass and loss are relatively unaffected by conjugate heat transfer, since conjugate and isothermal-wall simulations yielded essentially the same results for a range of air and wall temperatures ($< < 1\%$ relative difference for cases 4, 7–9). Therefore, remaining simulations were performed with

isothermal walls when examining bypass and loss. Heat-transfer rate from the fins is, however, strongly affected by conduction in the solids since the temperature on the fin surfaces decreases along y , and this will be discussed later.

Figure 6A also shows bypass and loss for different spacing between fins (cases 4 and 7–9). In this figure, it can be seen that the further the fins are to one another, the lower is the bypass and loss. The decrease in bypass is expected because the area available for the air to flow between the fins increases. The reason for the decrease in loss is more complicated and will be explained when examining how pressure changes with fin spacing in connection with **Figure 9**. **Figure 6B** shows the effect of fin thickness on

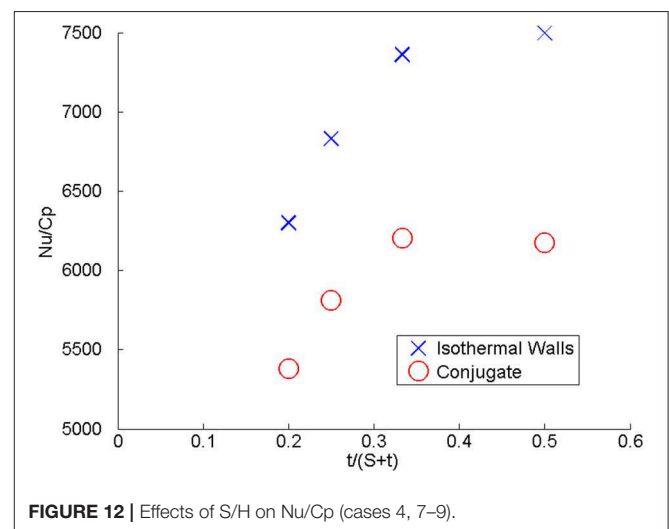
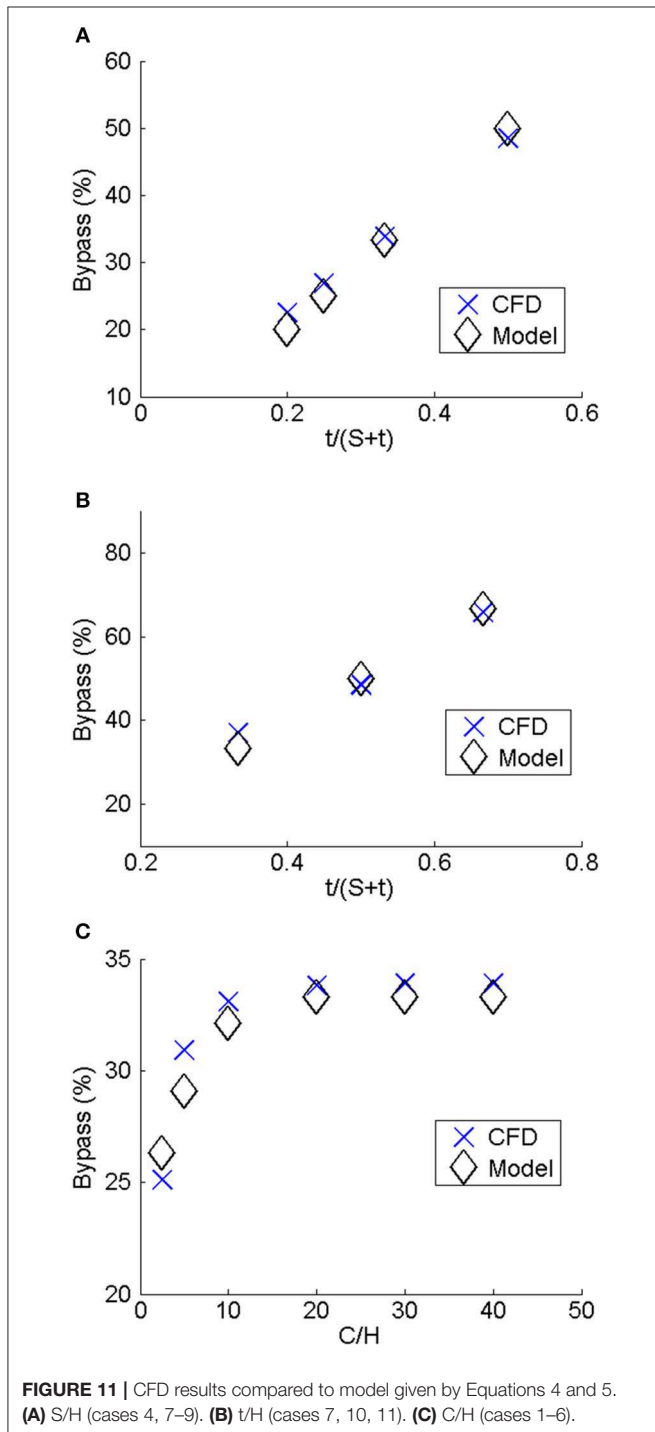


bypass and loss. From this figure, it can be seen that increasing the fin thickness increases bypass and loss. This is because increasing t for a given S increases the blockage for the oncoming flow. **Figure 6C** shows the effects of L/H on bypass and loss. From this figure, it can be seen that L/H has a small effect on bypass. This is because bypass is dominated by the stagnation region induced by the fin's leading edge. L/H , however, does affect loss. As length

increases, the loss also increases. This is because for sufficiently long L , all flow that entered the region between the fins will exit the fins at $y = H$ because of pressure rise from viscous forces. If bypass, loss, and pressure drop were plotted as a function of $t/(S + t)$, which is a measure of the blockage created by the fins of the surface cooler, then all increase monotonically with $t/(S + t)$.

Figure 7 shows the effect of S/H on the flow around the leading edge of the fins. This figure shows streamlines colored by velocity on a plane at $y = H/2$. From this figure, it can be seen that the flow separates around the leading edge. This separation bubble at the leading edge increases in size as S/H increases. Also for the range of parameters studied, as S/H increases, the maximum velocity between the fins decreases because bypass and loss are less and because the vena contracta effect diminishes.

Figure 8 shows the heat-transfer rate to the fins, Q , and the pressure drop across surface cooler from the inlet to the exit, $P_{in} - P_b$ as a function of fin spacing, S/H . As noted, although bypass and loss are relatively unaffected by conjugate heat transfer, heat transfer is strongly affected by whether conjugate heat transfer is accounted for, and this can be seen in **Figure 8A**. Also from this figure, it can be seen that Q increases as S/H increases until $S/H \approx 0.4$. For $S/H > 0.4$, further increases in spacing have minimal effect on the heat-transfer through the fins. As S/H increases from 0.2 to 0.4, Q per fin increased greatly from 12.8 to 14.3 W because bypass and loss greatly decreased from 48.5 to 26.3% for bypass and 54.4 to 26.3% for loss. The reason that Q did not increase more when S/H increased from 0.2 to 0.4 is because the size of the separation bubble that forms at the leading edge of the fin increases in size as S/H increases. As S/H exceed 0.4 for the operating conditions being studied, though bypass and loss continue to reduce—albeit at a much slower rate—the separation bubble also continues to increase in size. The net effect is essentially little increase in Q . This indicates that the fin's leading edge should be designed to eliminate or reduce the size of the separation bubble at its leading edge to fully benefit from the reduction in bypass and loss. By plotting Q as a function of fin blockage, $t/(S+t)$, Q reaches a maximum of 17.6 W when



$t/(S+1) = 0.25$ if isothermal wall and a maximum of 7.31 W when $t/(S+1) = 0.25$ if conjugate heat transfer is accounted for. This shows that there is an optimal blockage from the heat-transfer perspective.

On pressure drop, **Figure 8B** shows that, like bypass and loss, it is relatively unaffected by conjugate heat transfer. This figure also shows that pressure drop decreases as S/H increases, and the rate of decrease is less at higher S/H . Pressure drop decreases as S/H increases is primarily due to the number of fins per unit area and the interactions of the boundary layers on the fin surfaces that face each other. When S/H is small, the boundary-layer increases are strong which increases pressure drop. As S/H increases, those interactions reduce and the number of fins per unit area dominates in determining the pressure drop.

Figure 9 show the effects of S/H and t/H on pressure and velocity magnitude. From this figure, it can be seen that the fin creates a stagnation region upstream of its leading edge ($x = 0$), where the pressure reaches its highest value and the velocity magnitude approaches zero. This region of high pressure and low velocity magnitude extends considerably upstream of the fin's leading edge as well as spanwise to the region between the fins. The degree of the fin's upstream and spanwise influence is a strongly affected by S/H and t/H . The smaller the S/H , or greater the t/H , the stronger is the effect. As the flow approaches the fin,

part of it is diverted away from the fin by the stagnation region (bypass). For the part that enters into the region between the fins, it accelerates because of the reduced cross section area and because of the vena contracta effect from separation at the leading edge of the fin. At the vena contracta, the velocity magnitude is the maximum and the pressure is the minimum. Note that the maximum pressure created by the flow approaching the fin and the minimum pressure created by the flow accelerating in the region between the fins creates a net force that has a component in the y direction. Thus, once the flow enters the region between the fins, a portion exits the fins at $y = H$ (loss). The loss is highest near the fin's leading edge and decreases steadily.

Figure 10 shows bypass and loss as a function of inlet velocity, temperature difference between the inlet and the surface cooler walls, and inlet temperature. These parameters were found to have minimal effects on bypass. However, for the smallest spacing between fins, these three parameters do affect loss. This is because the temperature and velocity at the inlet affect the boundary layer thickness approaching the fin. When the boundary layer is thicker, the pressure at the base of the fin is higher. This increases the component of the pressure gradient in the Y -direction, which increases loss.

Based on results of this study, bypass was shown to be a function of the ratio of the frontal area of the fin to the total area

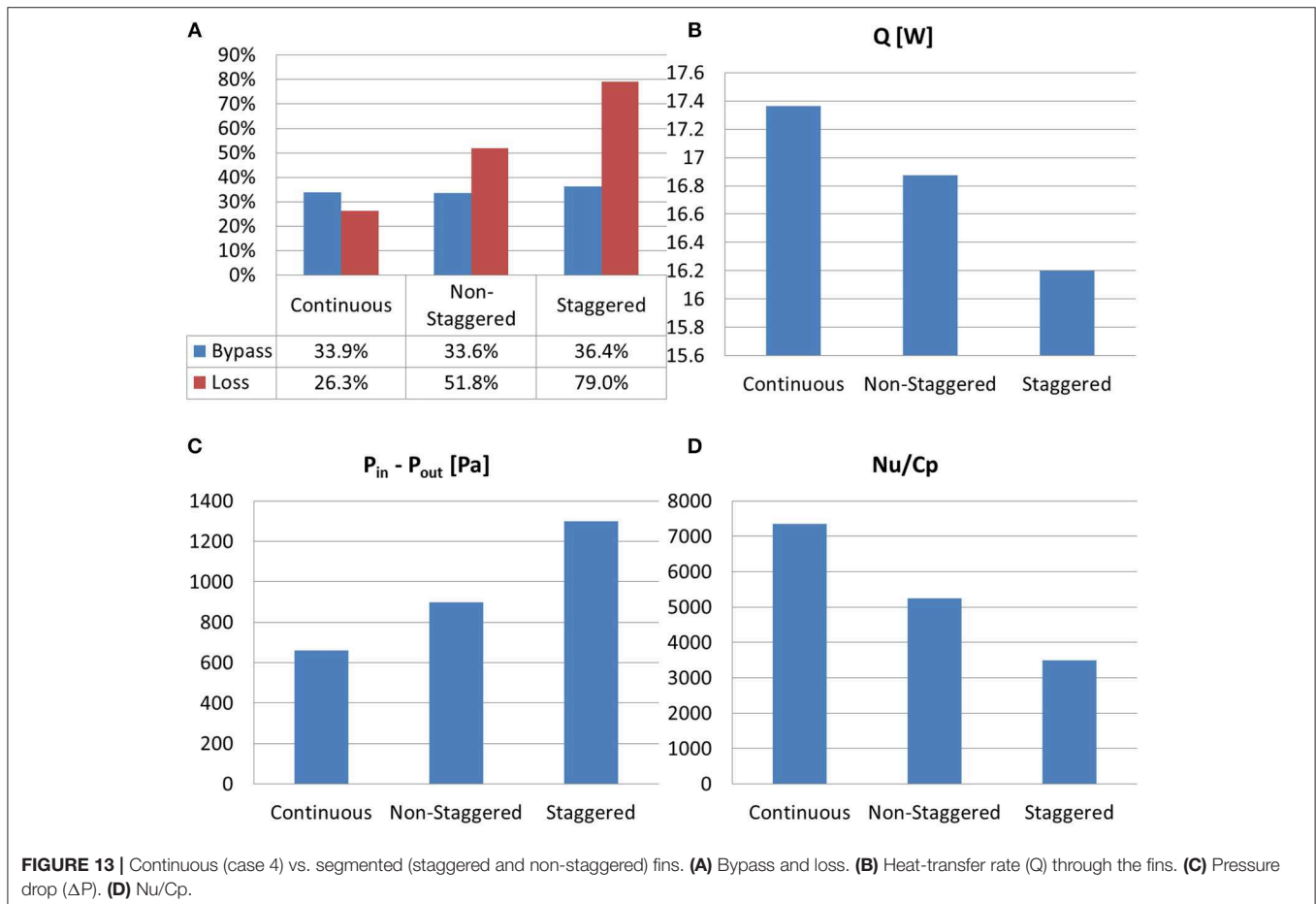


FIGURE 13 | Continuous (case 4) vs. segmented (staggered and non-staggered) fins. **(A)** Bypass and loss. **(B)** Heat-transfer rate (Q) through the fins. **(C)** Pressure drop (ΔP). **(D)** Nu/C_p .

$t/(t + S)$) and the clearance (C/H). When $C/H > 20$, bypass is very well modeled by simply considering the blockage area as shown by Equation (4). However, for $C/H < 20$, the effect of C must be accounted. Equation 5 is found from using power regression to the data generated to a curve and provides a method for estimating the bypass as a function of fin thickness, spacing, and channel height for rectangular fins. **Figure 11** shows the CFD results plotted against Equations 4, 5. From that figure, it can be seen that Equations 4, 5 match the CFD results well with relative error $<5\%$.

$$\text{Bypass} = \alpha \frac{t}{t+s} \text{ when } \frac{C}{H} > 20, \alpha = 1 \quad (4)$$

$$\text{Bypass} = \frac{t}{t+s} * 0.232 * \left(\frac{C}{H}\right)^{0.14} \text{ when } \frac{C}{H} < 20 \quad (5)$$

A dimensionless parameter to assess the performance of surface coolers is needed. Since the ideal surface cooler design is one with high heat-transfer rate and low pressure drop, the ratio of Nusselt number (Nu) to the pressure coefficient (C_p) could be useful in evaluating performance. Nu and C_p are defined by

$$Nu = \frac{hL}{k} \quad (6)$$

$$C_p = \frac{P_{in} - P_b}{\frac{1}{2} \rho V_{in}^2} \quad (7)$$

where

$$h = \frac{q''}{T_{wall} - T_{in}} \quad (8)$$

$$q'' = \frac{Q}{(2HL + 2Ht + Lt)} * \frac{t}{(S+t)} \quad (9)$$

Figure 12 shows how spacing between the fins affects Nu/C_p . From this figure, it can be seen that with conjugate heat transfer considered, Nu/C_p reaches a maximum at $S/H \approx 0.4$, which corresponds to $t/(S+t) \approx 0.33$ for a given H . When S/H exceeds 0.4 or correspondingly when $t/(S+t)$ is smaller than 0.33, Nu/C_p decreases. By plotting Nu/C_p as a function of fin blockage, $t/(S+t)$, Nu/C_p reaches a maximum of 7,500 when $t/(S+t) = 0.5$ if isothermal wall and a maximum of 6,210 when $t/(S+t) = 0.33$ if conjugate heat transfer is accounted for. This shows that there is an optimal blockage from the heat-transfer and pressure drop perspective.

Segmented Fins

Figure 13A shows the effects of segmenting the fin, both staggered and non-staggered arrangements, on bypass and loss. As can be seen in this figure, segmenting the fins has a small effect on bypass when compared to continuous fins. This is because the stagnation region induced by the fin's leading edge for segmented fins is nearly identical to that for continuous fins. However, for segmented fins, bypass occurs at the leading edge of each fin downstream of the first fin. Thus, the loss for segmented fins is substantially higher than those for continuous fins. For

segmented fins, staggering the fins produce even more loss than non-staggered fins. **Figure 13B** shows the heat-transfer rate of segmented fins when compared to continuous fins. From this figure, it can be seen that Q is lower when the fins are segmented. This is because segmented fins have significantly higher loss as well as lower surface area. On pressure drop, **Figure 13C** shows that it increases when the fins are segmented, and is substantially higher when the segmented fins are staggered. This is because when the fins are segmented, each fin's leading edge creates huge disturbances in the flow field. Thus, the spacing between fins in the streamwise direction is a design parameter that needs to be examined. In particular, the spacing between fins should be designed to minimize flow disturbance and still enable the restart of the boundary layer to enhance surface heat transfer at the leading edge of each segmented fin. **Figure 13D** shows the effect of segmenting the fins on Nu/C_p . From this figure, for the geometries studied, segmenting the fins, and especially staggering the segmented fins, resulted in the worst surface cooler.

SUMMARY

Steady RANS based on the SST model was used to study bypass and loss and their effects on heat transfer and pressure drop for two surface coolers, one with continuous fins and the other with segmented fins that are arranged in either staggered or non-staggered fashion. For continuous fins, this study found the clearance to have significant effects on bypass and loss up to a clearance of 38 times the height of the fin. The ratio of the frontal area of the fin (fin thickness) to the total area (fin thickness and the spacing between the fins) was also found to have a significant effect on bypass and loss. Bypass is dominated by the stagnation region in front of the fins' leading edge. Thus, the leading edge of fins should be designed to minimize the size of the stagnation region and reduce/minimize flow separation downstream of it. The temperature and velocity approaching the surface cooler has an effect on the boundary layer thickness and thereby affects the loss when the spacing between the fins is small. A model was developed for predicting bypass as a function of fin thickness, spacing between the fins, and the clearance. It was found that when the clearance is sufficiently high, bypass is very well modeled by considering the blockage area. Segmenting the fins was found to have little effect on the bypass, because the stagnation region induced by the fin's leading edge is nearly the same for segmented and continuous fins. However, segmenting the fins greatly increases loss because each of the downstream segmented fin's leading edge creates huge disturbances in the flow field. Thus, the spacing between fins should be designed to minimize flow disturbance and still enable the restart of the boundary layer to enhance surface heat transfer at the leading edge of each segmented fin. When bypass and loss are high, heat-transfer rate is low and pressure drop is high. Thus, decreasing bypass and loss is essential for improved heat transfer and reduced pressure drop. By using the parameter Nu/C_p , it was found that for the conditions studied, $t/(S + t) = 0.33$ gave the highest heat transfer per pressure drop when conjugate heat transfer is accounted for.

AUTHOR CONTRIBUTIONS

JL carried out all aspects of the research, including the simulations and analysis of the results. JP assisted in the analysis of the results in the early stage of the work. TF and KY assisted in the interpretation and analysis of the results. TS framed the problem and provided guidance on all aspects of the research.

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Conflict of Interest Statement: The authors declare that the research was conducted in the absence of any commercial or financial relationships that could be construed as a potential conflict of interest.

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NOMENCLATURE

α = Fitting parameter for bypass in Equation (4) ($\alpha = 1$ for cases studied)

C = Half of the channel height

C_p = Non-dimensional pressure drop, $\frac{P_{in} - P_b}{\frac{1}{2} \rho V_{in}^2}$

H = Height of the fin

h = Heat transfer coefficient

k = Thermal conductivity

L = Length of the surface cooler

L_h = Length of the segmented fin

L_d = Distance between segmented fins in the x direction

\dot{m}_{inlet} = Mass flow rate at the inlet for $y \in [0, H]$

\dot{m}_{finin} = Mass flow rate at the start of the surface cooler ($x = 0$) for $y \in [0, H]$

\dot{m}_{finout} = Mass flow rate at the end of the surface cooler ($x = L$) for $y \in [0, H]$

Nu = Nusselt number, $h[L_t/(S+t)]/k$

P = Pressure

P_{in} = Pressure at the inlet ($x = -14H$)

P_b = Pressure at the outlet ($x = 42H$)

q'' = Wall heat flux

Q = Heat-transfer rate from a single fin

S = Distance between two successive fins in the z direction

t = Thickness of the fin

T = Temperature

T_{in} = Temperature at the inlet ($x = -14H$)

T_{wall} = Temperature of the fin surface if isothermal; temperature at $y = -0.2H$ if conjugate

V_{in} = Velocity at the inlet ($x = -14H$)

x,y,z = Coordinate system (see **Figure 2**)