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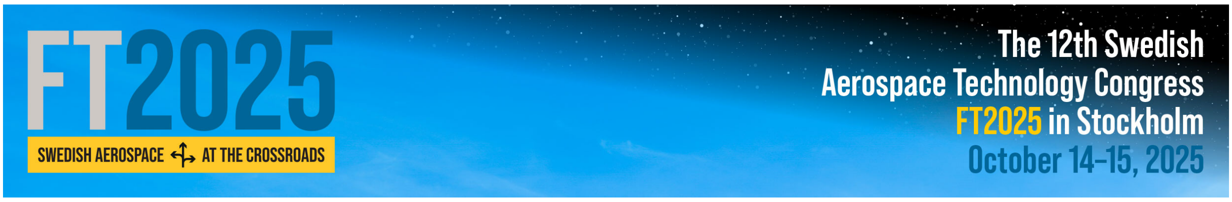


# GenHEX: A new heat exchanger design framework

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## Abstract

Thermal management is considered a main challenge when attempting to further increase the efficiency of gas-turbine engines or during the development of future aircraft engine concept including fuel cell, hybrid electric or unconventional cycle engines such as the composite cycle engine. Heat exchanger selection is notoriously difficult, especially in aviation when weight and volume are of great importance and should be included already during the early design stages. This paper aim to further convince the usefulness of the GenHEX method and present how it can be used to estimate heat exchanger performance without having to select a heat exchanger family of configuration. This is achieved through the generalization of all/any heat exchanger matrix geometry to the lowest possible resolution which still allow for estimation of the aerothermal performance and design guidance when attempting to translate the generalized geometrical parameters to actual matrix geometries. Although the GenHEX method has been validated against state-of-the-art heat exchangers and has already been implemented into engine cycle simulations for the development of future aircraft propulsion It is of great interest to broaden the usage to further benchmark the method against existing and installed heat exchangers to further improve the performance estimation metrics while providing a easy-to-use means for selecting application suitable heat exchangers.

**Keywords:** Heat Exchangers, Thermal management, Heat transfer, Cooling

## 1 Introduction

Thermal management is a hot topic, and the task of heat exchanger selection is notoriously difficult. Heat exchangers come in a vast range of designs and so far none have proven to be universally superior. Instead, each comes with different trade-offs for weight, volume, aerothermal performance, manufacturability, and cost. The heat exchanger design task can be reduced to two steps, rating and sizing [1]. The sizing step is, in comparison, fairly straightforward and uses scaling parameters to reach the target thermal load. In the rating step, one first has to decide which family of heat exchangers to consider - tubes, plates, etc and whether they should have fins or not - and then which of the huge number of different configurations - tube shapes, plate spacings, fin types - that suits best. The normal design process involves bouncing between databases, handbooks, publications and simulation tools, where the number of evaluated configurations greatly correlates to the likelihood of finding the most suitable heat exchanger for their application. One could buy access to a heat exchanger design tool - no names mentioned to avoid commercialism - which can aid the process, but the problem still remains since the designer has to provide design inputs and know what trade-off is best for their application. Sadly, design guides mainly mentions intuition or experience as means for streamlining the design process [1]. Sadly, intuition is difficult to achieve and therefore many are now trying to circumvent that by utilizing the rapidly increasing power and availability of machine learning and artificial intelligence (AI). So far, the most common usecase for AI has been to attempt inter- and extrapolation of configurations based on specific design parameters - such as tube shapes, fin spacings, corrugation angles, pin diameters, plate curvatures, channel widths, void fractions, etc - which are often unique for each family of heat exchangers. The large amount of different design parameters has, so far, restricted the interpolation between heat exchanger families, meaning that intuition, design guidelines, or brute force are still required.

During cycle innovation, among other cases, the detailed design of the heat exchanger is less important than the expected performance. For land-based applications, the aerothermal performance - heat transfer and fluid friction - is the main concern, and it is common practice to design based on the coefficient of performance (thermal/pumping power) at a given effectiveness. However, for airborne applications the overall performance - aerothermal, specific, and volumetric - should be considered since weight and volume have a large impact on the system performance. A more wholesome estimation strategy for the heat exchangers are therefore required to reduce the risk of achieving overoptimistic performance, which could encourage development of concepts which at a later stage might prove unfeasible due to the huge or heave heat exchangers.

To solve these issues, a new framework called GenHEX is promoted which can be used to estimate the overall performance and also provides guidelines for the main features of the detailed heat exchanger design. It uses a reduced-order geometrical expression which spans the entire design space, overarching multiple heat exchanger families, using only six parameters for rating and another four for scaling. The designer is now able to include this framework into their cycle and scan the entire design space for overall performance which suits their application. It is also advised to predetermine some of the design parameters to reduce the computational cost and increase the feasibility of the outcome. The generalized parameters which govern the matrix design of the low-resolution geometry can later be used as pointers when deciding detailed design features of the most suitable heat exchanger.

In this work, GenHEX and the related design parameters will be thoroughly introduced, followed by guidelines on how to interoperate the generalized geometrical parameters and translate them to actual heat exchanger geometries.

## 2 The geometrical description

At first glance, it may seem impossible to predict the aerothermal performance of a heat exchanger without knowing its exact geometry. After all, heat transfer and pressure losses are strongly influenced by the detailed arrangement of walls, fins, passages, and flow paths. However, much of this complexity can be abstracted — at least in the early design phase — by focusing on the underlying physics that govern performance. After all, the purpose is not to estimate the performance of a certain heat exchanger geometry, it is instead to estimate the best possible heat transfer for a certain pressure loss, weight, and volume. As if a state-of-the-art heat exchanger perfectly suited for the application is used.

To demonstrate the low-resolution geometrical description, consider an idealized heat exchanger: a compact, cross-flow configuration where the two fluids are separated by thin walls, flow uniformly across the surface, and engage the full internal volume. There are no entrance effects, wakes, or dead zones. In such a setup, the thermal and hydraulic behavior can be approximated analytically or using classical flat plate correlations.

As complexity is gradually added - non-uniform flow, asymmetric surfaces, varying wall thicknesses — the challenge becomes to preserve predictive capability without reverting to full CFD or geometry-specific models. The key question is: what is the minimum set of parameters that still captures the essential performance trends across very different HEX types? This is the motivation behind the GenHEX framework, which in essence builds on three pillars; First, a geometrical description of the heat exchanger at the lowest possible resolution at which it can still be translated into actual designs. Any heat exchanger should be uniquely represented and it should span the entire design space continuously, even between heat exchanger families. Second, a method for estimating the aerothermal performance of the generalized heat exchanger. The estimated aerothermal performance should be "as if" using a state-of-the-art heat exchanger suited for the application. Third, an approach for grading the low-resolution design which takes the user specific application into consideration. Although the second and third pillars are very interesting and important, they will not be the main focus of this paper, but are previously explained in other publications [2, 3].

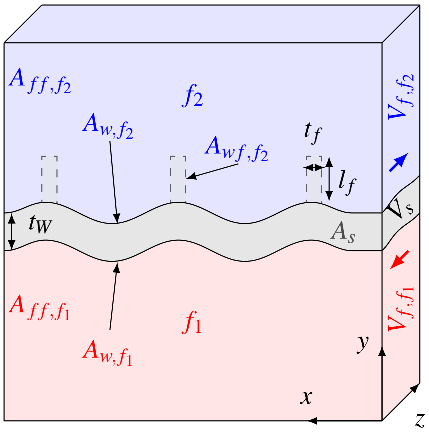
### 2.1 The design parameters

The geometry is described and fully defined by 10 parameters and two geometrical equations. The novelty of the geometrical description used in GenHEX mainly lies in the geometrical generalization parameters (GGPs) - the *void fraction ratio* ($\sigma_r$), *surface area density ratio* ($\alpha_r$) and *solid volume fraction* ($\chi$) - and the equations used to couple these to the *heat transfer surface area* ($A_{w,f}$) and *free flow area* ($A_{ff,f}$) for each fluid stream. Although a specific heat exchanger matrix corresponds to a unique set of GGPs, the reverse is not true as a given set of GGPs can map to many different heat exchanger configurations. Hence, it still falls on the designer to find a high-performance heat exchanger for their application, but knowing these design parameters greatly reduces the number of configurations which should be evaluated. The design parameters are listed and illustrated in Figure 1 then presented in turn. Note that even though the illustration shows a wavy plate heat exchanger with fins, the generalized parameters could represent just any heat exchanger.

A small note on the subscripts used, a total quantity such as the *total heat transfer surface* $A_w$ is the sum of heat transfer surfaces in each stream $A_{w,f}$ such that $A_w = A_{w,f_1} + A_{w,f_2}$. The subscripts $f_1$ or $f_2$ refer to a certain stream, while the subscript $f$ could be either of the streams and mainly indicate that it is not the total quantity.

#### The outer dimensions: $L_x, L_y, L_z$

The outer dimensions of the heat exchanger are used to scale the total *heat transfer surface area* ($A_w$) of the heat exchanger. Scaling along any axis ($L_x, L_y$ or $L_z$) has a directly proportional relationship to $A_w$ ($(aL_x)(bL_y)(cL_z) = (abc)A$). However, along with the outer dimensions, the flow directions of each fluid need to be provided, which determine whether the heat exchanger operates in cross-, parallel-, or counterflow. The outer dimension along either flow direction is denoted $L_f$. Increasing the heat transfer area by scaling along the flow




**Scaling inputs**

- Outer dimension : $L_x, L_y, L_z$
- Overall structure thickness : $t$

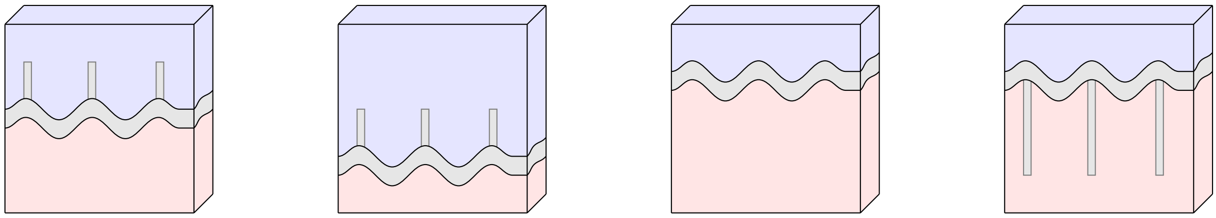
**Matrix inputs**

- Undisturbed flow lengths : $\ell_{f_1}, \ell_{f_2}$
- Fin characteristic dimension : $l_f/\sqrt{t_f}$
- Void fraction ratio* : $\sigma_r$
- Surface area density ratio* : $\alpha_r$
- Solid volume fraction* : $\chi$

**Products**

- Total volume : $V_t = L_x L_y L_z$
- Solid volume : $V_s = \chi V_t$
- Frontal area : $A_{fr} = L_x L_y$
- Free-flow area : $A_{ff} = \sigma A_{fr}$
- Solid cross section area : $A_s = \chi A_{fr}$
- Wetted surface area : $A_w = \alpha V_t$
- Wetted fin surface area : $A_{wf}$
- Fluid volume : $V_f = \sigma V_t$

*Figure 1: General representation of a counterflow heat exchanger that transfers heat between two fluids ($f_1, f_2$), including lists of inputs required to determine which matrix to use and how to scale it for the thermal load. Also important products for heat exchanger design. *Geometrical generalization parameters (GGPs)*



*(a)* $\sigma_r \approx 1$, $\alpha_r < 1$

*(b)* $\sigma_r < 1$, $\alpha_r < 1$

*(c)* $\sigma_r > 1$, $\alpha_r = 1$, $\chi$ *decreased*

*(d)* $\sigma_r > 1$, $\alpha_r > 1$, $\chi$ *increased*

*Figure 2: Changes in the geometry of the heat exchanger matrix from varying the geometrical generalization parameters.*

direction of either fluid is less efficient than scaling any of the transverse directions, since the fluid velocity remains constant, but the flow length increases, which results in higher pressure losses. The outer dimensions of the heat exchanger are mainly used to calculate the *total volume*:

$$V_t = L_x L_y L_z \tag{1}$$

**The solid volume fraction: $\chi$**

The ratio of *solid volume* ($V_s$) to $V_t$ is denoted by the *solid volume fraction*

$$\chi = V_s/V_t \tag{2}$$

This governs how much of the total volume is occupied by the solid volume and is strictly related to the porosity (1-$\chi$) used by others. A higher value of $\chi$ results in a heavier heat exchanger which has more heat transfer surface area ($A_w$) and thus transfers more heat, but also has more narrow flow passages, and thereby greater pressure losses.

**The overall structure thickness: $t$**

The *overall structure thickness* ($t$) is used to calculate the total amount of surface area ($A_w$) within the heat exchanger, given the solid volume fraction.

$$A_w = \frac{V_s}{t/2} \tag{3}$$

In the case of an unfinned heat exchanger, it is simply the average thickness of the walls separating the fluids. These walls are usually, but not necessarily, constant in thickness. In the case of a finned heat exchanger where the *thickness of walls* ($t_W$) and *thickness of fins* ($t_f$) are different, then $t$ is the surface average of these thicknesses. Hence, instead of $t$ as input, one could provide $t_W$ and $t_f$. This would increase the number of parameters required, but in the case where either or both is predetermined, it might be a good idea.

**The fin characteristic dimension: $l_f/\sqrt{t_f}$**

The ratio of *fin length* ($l_f$) to the square root of $t_f$ is denoted *fin characteristic dimension* ($l_f/\sqrt{t_f}$) and has an impact on the *fin surface efficiency* $\eta_f$, as shown in Figure 6, along with the *thermal conductivity* ($k$) of the solid material. The fin characteristic dimension also govern the structural integrity of the fin as it somewhat resembles an aspect ratio of length to thickness, meaning that a sufficiently low value could ensure structural integrity even as the fins are extended. However, both structural integrity and fin efficiency increase with increased $t_f$, at the cost of reduced $A_w$ for constant $V_s$. In the case where $t_f$ is given, then $l_f/\sqrt{t_f}$ implicitly sets the length of the fin, which can result in designs featuring infeasible long and slender fins which could bend or break once built.

**The surface area density ratio: $\alpha_r$**

The total heat transfer surface area from Equation 3 should be distributed between the two fluid streams. The *surface area density* ($\alpha_f = A_{w,f}/V_t$) of each fluid is a measure of how much surface area per total volume there is. The distribution of heat transfer surface areas is governed by the *surface area density ratio*

$$\alpha_r = \alpha_{f_1}/\alpha_{f_2} \tag{4}$$

which practically becomes a ratio of heat transfer surface areas ($\alpha_r = A_{w,f_1}/A_{w,f_2}$). Currently, the practice is to exclude designs that feature fins protruding into both fluids, since it is believed that adding more walls - tubes or plates - is a more efficient method than adding fins, simply due to the temperature gradients along fins which reduce the surface efficiency. Hence, bare (non-finned) designs will have $\alpha_r \approx 1$, while values far above, or below, unity tells the designer which side should be finned along with the *wetted fin area* ($A_{wf}$) and the *finned to total surface area ratio* ($\sigma_{ftt} = A_{wf}/A_w = (1-\alpha_r)/(\alpha_r+1)$.

**The void fraction ratio: $\sigma_r$**

The pressure losses in a heat exchanger are strongly related to the flow velocities and thereby the *free flow area* ($A_{ff,f}$) provided for each fluid. These free flow areas would preferably be very large but are limited by the cross-sectional *frontal area* ($A_{fr}$) of the heat exchanger and the *cross-sectional area of the solid material* ($A_s$)

$$A_{fr} = A_{ff,f_1} + A_{ff,f_2} + A_s \tag{5}$$

The *void fraction* ($\sigma_f = A_{ff,f}/Afr$) is the ratio of the free flow area to the frontal area for each fluid. The *void fraction ratio*

$$\sigma_r = \sigma_{f_1}/\sigma_{f_2} = (A_{ff,f_1}L_{f_1})/(A_{ff,f_2}L_{f_2}) \tag{6}$$

governs the distribution of the available free flow area between the two fluids. However, for counterflow or parallel flow configurations, the frontal area and the length of the heat exchanger along the flow direction are equal for each fluid, and thus $\sigma_r = A_{ff,f_1}/A_{ff,f_2}$. It is therefore worth repeating that extending the heat exchanger volume in flow transversal directions is a good idea since it allows greater values of $A_{ff}$, for counterflow or parallel flow configurations, there are two transversal directions while for cross flow configurations there is only one transversal direction.

**The undisturbed flow lengths: $\ell_{f_1}, \ell_{f_2}$**

In order to estimate the aerothermal performance of these generalized heat exchanger geometries, the correlations for the Colburn factor and the Fanning friction factor were derived from the heat exchanger library published by Kays and London [4]. The correlations are based on the *Reynolds number* (*Re*) and the *undisturbed flow length* ($\ell$) divided by the *hydraulic diameter* ($D_{h,f} = 4\sigma_f/\alpha_f$). $\ell$ was introduced by LaHaye et. al. [5] and practically resemble the length along the flow direction for which boundary layers are developed on structures before the structure is ended and the boundary layer development restarted. The general trend is that shorter flow structures, low values of $\ell$, increase the absolute values of heat transfer and pressure losses, at the cost of reduced heat transfer to pressure loss efficiency.

### 2.2 The two geometrical equations

Two geometrical equations are used for the reduction of input parameters and could be seen as the foundation of the reduced order geometrical description. The first equation is the assembly of the total volume from the *fluid volumes* ($V_f$) and solid volume.

$$\begin{aligned} V_t &= V_{f_1} + V_{f_2} + V_s && (7)\\ &= A_{ff,f_1}L_{f_1} + A_{ff,f_2}L_{f_2} + V_s && (8) \end{aligned}$$

Dividing by the total volume and some slight rearranged result in the first geometrical equation.

$$\begin{aligned} 1 &= \frac{A_{ff,f_1}L_{f_1} + A_{ff,f_2}L_{f_2} + V_s}{V_t} && (9)\\ &= \frac{A_{ff,f_1}}{A_{fr,f_1}} + \frac{A_{ff,f_2}}{A_{fr,f_2}} + \frac{V_s}{V_t} && (10)\\ &= \sigma_{f_1} + \sigma_{f_2} + \chi && (11)\\ \Rightarrow \sigma_{f,2} &= \frac{1-\chi}{\sigma_r+1} && (12) \end{aligned}$$

The second equation originates from the distribution of the total heat transfer surface area

$$V_s = \left(A_{w,f_1} + A_{w,f_2}\right)\frac{t}{2} \tag{13}$$

Again, it is divided by the total volume and rearranged.

$$\begin{aligned} \chi &= \left(\alpha_{w,f_1} + \alpha_{w,f_2}\right)\frac{t}{2} && (14)\\ \Rightarrow \alpha_{f_2} &= \frac{2\chi}{t\,(\alpha_r+1)} && (15) \end{aligned}$$

# 3 Selecting the heat exchanger matrix

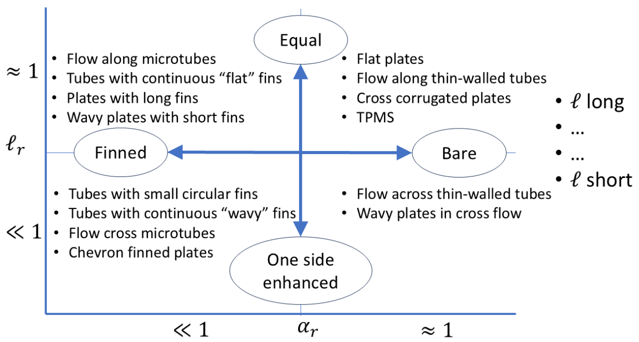



*Figure 3: Matrix showing suitable geometries depending on the surface area density ratio ($\alpha_r$) and undisturbed flow lengths ($\ell$)*

The main parameters governing what type of heat exchanger matrix to use are $\alpha_r$ and the respective values of $\ell_f$. In short, the values of $\alpha_r$ govern whether the matrix should have protruding surfaces, namely fins, or not, while the values of $\ell_f$ show whether the surfaces should be long and straight or short/curved. In general, it can be observed that a low thermal conductivity penalizes the fins and, therefore, $\sigma_r \to 1$, having a high penalty for weight or volume will promote high heat transfer and thereby short values of $\ell_f$, having a configuration where one fluid is thermally superior to the other will lead to $\ell_r \neq 1$ where the thermally inferior fluid would have a short value of $\ell$. In Figure 3 certain heat exchanger matrices are sorted according to their values of $\alpha_r$, $\ell_r = \ell_{f_1}/\ell_{f_2}$ and magnitudes of $\ell$, and the different regions of this figure are discussed below.

**The top left corner:** $\ell_r \approx 1$, $\alpha_r \ll 1$

The top left corner features matrices with equal undisturbed flow lengths for both fluids but a large difference in the heat transfer surface areas. Suitable heat exchanger matrices would then be continuously finned tubes where the tube spacing is large enough for boundary layers to be fully developed on the external side. Other options would be plate heat exchangers, where one fluid side has long fins while the other has none, or microtube bundles in counter or parallel configuration. These heat exchangers are probably used when one fluid is thermally superior and adding more surface area to the thermally inferior stream is more worth it than enhancing heat transfer by increasing fluid friction. One case could be if a material of high thermal conductivity can be used.

**The top right corner:** $\ell_r \approx 1$, $\alpha_r \approx 1$

These heat exchangers feature an equal heat transfer surface area in both streams and also equal values for the undisturbed flow length. Examples with high values of $\ell$ would be flat plates or bare, thin-walled, tube bundles in counter- or parallel flow. These feature low absolute values for heat transfer and friction but a good heat transfer to friction ratio. Hence, they might be suitable for applications with similar fluids on both sides where pumping power is the main concern and heat exchangers are allowed to have a large volume. For short values of $\ell$, you could have a cross-corrugated plate or a *triply periodic minimal surface* (TPMS) heat exchanger. Note that achieving $\sigma_r \neq 1$ might be difficult for these!

**The bottom left corner:** $\ell_r \ll 1$, $\alpha_r \ll 1$

The bottom left corner has a large difference in both the heat transfer surface area and the undisturbed flow lengths. These types of heat exchangers would be suitable when one stream is thermally greatly superior to the other and a compact design is needed. One example could be an air-oil heat exchanger where the oil is thermally superior, and thereby the air stream benefits from greater surface area along with heat transfer enhancements. Suitable heat exchangers can be tube bundles with wavy fins, flow across microtube bundles, or plate heat exchangers featuring chevron or other fins.

**The bottom right corner:** $\ell_r \ll 1$, $\alpha_r \approx 1$

The bottom right corner features configurations where the heat transfer surface area is equal in both streams but one stream has a much shorter undisturbed flow length than the other. These heat exchangers might be used when one stream benefits from increased heat transfer and sacrificing some pressure loss is more beneficial than adding more weight in terms of surface area. An example would be if the designer is limited to a material of poor thermal conductivity. Thin-walled tube bundles in cross-flow or wavy plates in cross-flow are the main candidates here.

### 3.1 Translating the generalized description to actual matrices

After deciding on a heat exchanger family, one has to translate the generalized geometry to actual design features such as tube or fin spacings. However, no such guidelines have been presented before now! Finned or bare tubular matrices as well as bare or offset-strip finned plate heat exchangers are very common heat exchanger geometries and, therefore, chosen for the derivation. In the following examples, the hot fluid (orange) is denoted as fluid 1, such that the ratios are defined as hot/cold ($\sigma_r = \sigma_{f_1}/\sigma_{f_2} = \sigma_h/\sigma_c$ and $\alpha_r = \alpha_{f_1}/\alpha_{f_2} = \alpha_h/\alpha_c$).

#### 3.1.1 Continuous plates

The most basic heat exchanger design would consist of continuous plates where the hot and cold fluids alternate in each passage, such a design is illustrated in Figure 4a. A bare (unfinned) design should be used when $\alpha_r \approx 1$ and should be set up with values of $\ell_f$ that resemble the length of the heat exchanger along the flow directions of each fluid.

**Without fins**

In the case of bare plates where the *plate thickness* ($t_w$) and suitable values for $\sigma_r$ and $\chi$ are determined, the remaining parameters to calculate are the *number of plates* ($n$) and their *spacing* in the hot ($w_h$) and cold ($w_c$) streams. Obviously, the plates need to be aligned with the flow of both fluid streams, meaning that the plate-normal direction is optional for parallel or counterflow configurations but fixed in cross-flow configuration. Denoting the plate-normal width of the heat exchanger by $w$, the number of plates and the respective channel widths are calculated as

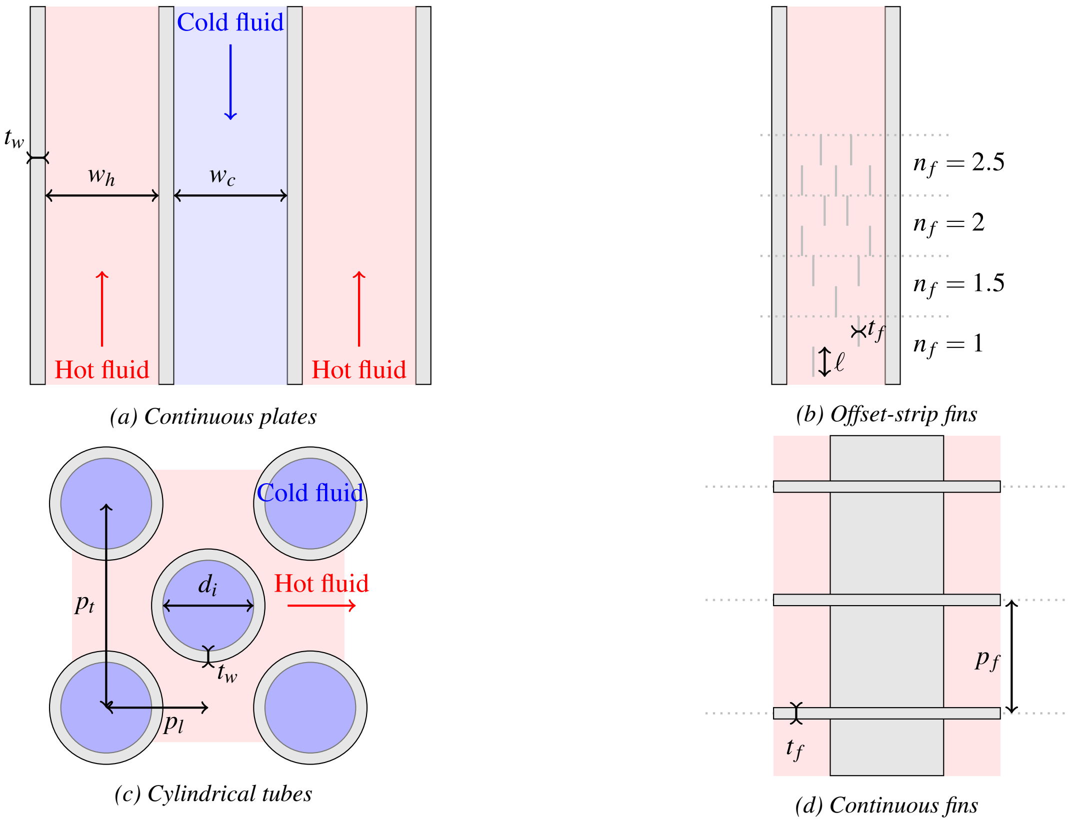
$$n = \frac{\chi w}{t_w} \tag{16}$$

$$w_c = 2\frac{w/n - t_w}{\sigma_r + 1}, n-1??? \tag{17}$$

$$w_h = w_c \sigma_r \tag{18}$$

**With offset-strip fins**

A configuration featuring $\alpha_r \neq 1$ could be achieved by adding offset-strip fins to one fluid stream, where adding fins to the hot stream in Figure 4a would result in $\alpha_r > 1$. Implementing a *number of fins* ($n_f$) parameter, which represents the number




*(a) Continuous plates*

*(b) Offset-strip fins*

*(c) Cylindrical tubes*

*(d) Continuous fins*

*Figure 4: Different heat exchanger configurations*

of parallel fins in the hot stream, as illustrated in Figure 4b, one can now calculate the design parameters as

$$n_f = \alpha_r - 1 \tag{19}$$

$$n = \frac{\chi w - (n_f t_f)/2}{t_w} \tag{20}$$

$$w_c = \frac{2\left(w/n - t_w - n_f t_f\right)}{1 + \sigma_r} \tag{21}$$

$$w_h = w_c \sigma_r + n_f t_f \tag{22}$$

### 3.1.2 Circular tubes

Although quite basic, heat exchangers that feature bundles of staggered circular tubes are quite common and a good choice of design if manufacturing cost should be reduced and/or the tube internal fluid is at a very high pressure. The main downsides are the lack of heat transfer enhancements on the tube internal side and the risk of additional losses in the tube internal fluid in the wake region of the tubes. The tube internal heat transfer can be enhanced by bending the tubes or by inserts that mix the fluid. The tube external losses can be reduced by correct staggering of the tubes or by elongating the tubes in the direction of the external flow to either elliptical or flat sided tubes. The tube external stream can be enhanced by the addition of fins to greatly increase the heat transfer surface area, the fins are either round and extruded from each individual tube or continuous and thermally bonded to the tubes.

**Without fins**

For configurations without fins, the tube inside diameter ($d_i$) and the longitudinal ($p_l$) and transversal ($p_t$) pitches need be determined. From the definition of the undisturbed flow length ($\ell$) by LaHaye et. al. [5] the longitudinal pitch should be the same as $\ell$. Then $p_t$ and $d_i$ can be calculated.

$$p_l = \ell_{f_1} \tag{23}$$

$$d_i = 2t_w \frac{(1/\chi - 1)(\alpha_r + 1)}{1 + \sigma_r} \tag{24}$$

$$p_t = \pi \frac{t_W d_i + t_W^2}{\chi p_l} \tag{25}$$

**With continuous fins**

In the case of tubes connected by continuous fins, we calculate the design parameters as

$$p_l = \ell_{f_1} \tag{26}$$

$$d_i = 2t_w \frac{(1/\chi - 1)(\alpha_r + 1)}{1 + \sigma_r} \tag{27}$$

$$p_t = \frac{\pi \left(\frac{d_i}{2}\right)^2 (\sigma_r + 1)}{p_l (1 - \chi)} \tag{28}$$

$$p_f = \frac{\frac{2 p_l p_t}{\pi} - \left(\frac{d_i}{2} + t_W\right)^2 - t_f (d_i + 2t_W)}{d_i (\alpha_r - 1) - 2t_W} \tag{29}$$

where it can be observed that the value of $p_t$ for the finned case goes towards the same value as for the case without fins when $t_f \to 0$ or $p_f \to \infty$.

# 4 Design workflow

During the early design stages, any heat exchanger should be simplified to the lowest possible resolution to allow scanning a large design space. Preferably, most design parameters should be part of the optimization routine, which could even include variations in the flow parameter such as the coolant mass flow rate. An optimization routine would then sample a set of heat exchanger configurations and grade them according to some user-provided criteria to determine the appropriate design features. Usually, a multi-objective optimization routine is required, since most design tasks target to minimize a cost function for a given thermal load. A suitable workflow is provided in Figure 5

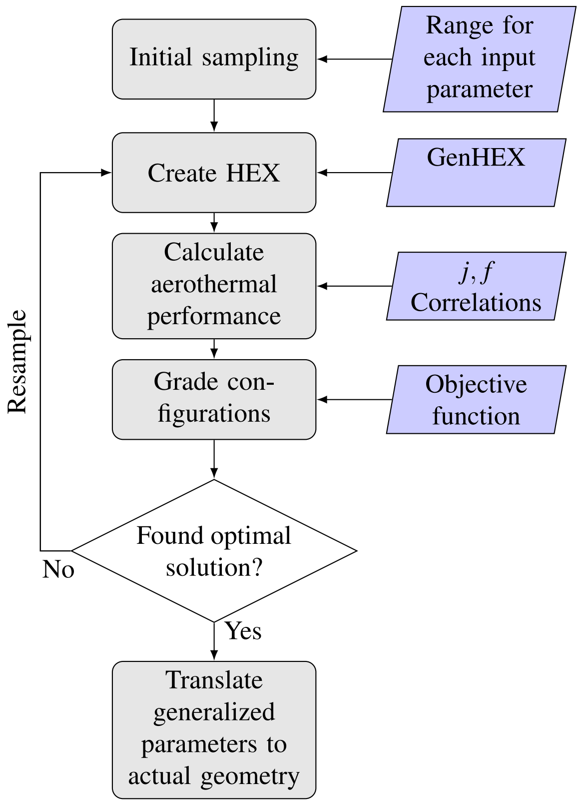
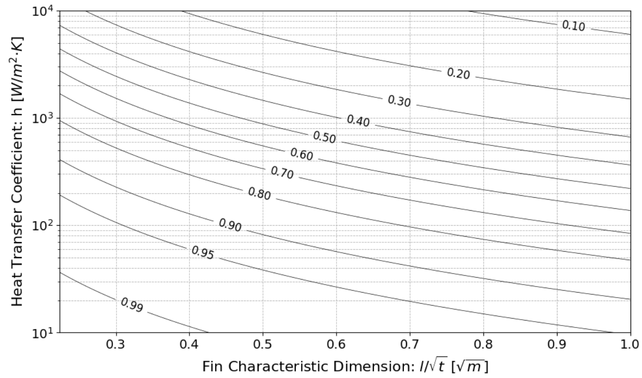


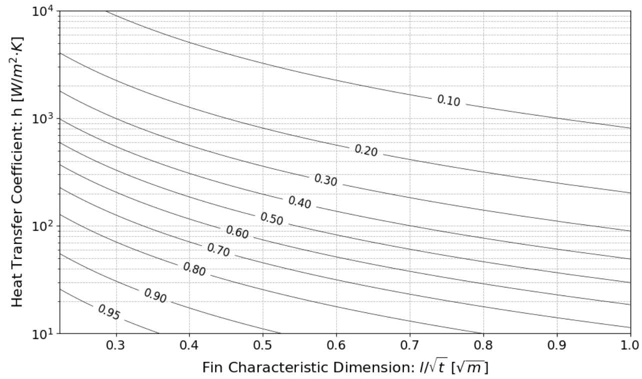

*Figure 5: Flowchart illustrating a recommended workflow when designing heat exchanger using GenHEX.*

## 4.1 Design study setup

Not all parameters should be left open for optimization during design progress. The most obvious one to exclude is the *structure thickness* ($t$), since any heat exchanger would weigh less and have lower thermal resistance for reduced thicknesses. The trade-off is that the structure becomes more fragile, which is currently not accounted for in GenHEX. Another parameter to exclude from optimization is the *fin characteristic dimension* ($l_f/\sqrt{t_f}$), which is a structural and performance parameter and should be chosen by the designer, taking into account the material, the risks of harsh conditions,




*(a) Aluminum 2219*
$k = 120$ *W/mK*




*(b) Stainless steel 304*
$k = 15.6$ *W/mK*

*Figure 6: Fin efficiency for varying heat transfer coefficient and fin characteristic dimension for Aluminium (2219) and Stainless steel (304).*

and the cost of maintenance. Reducing $l_f/\sqrt{t_f}$ by shortening the fins or increasing their thickness generally increases fin efficiency ($\eta_f$), but will reduce the total surface area for constant heat exchanger weight, so there is definitely a trade-off worth investigating as long as the structural integrity of the heat exchanger is not compromised. Figure 6 shows how the characteristic dimensions and the heat transfer coefficient influence $\eta_f$ for two different materials. The last of the parameters that should be influenced by the designer are the undisturbed flow lengths; these govern how the heat exchanger matrix is represented in the generalized description. An example would be the case where there is a large difference in pressure between the two fluids; in that case the high pressure fluid would likely have a long undisturbed flow length ($\ell$) since it would run inside near-circular tubes where the boundary layers are fully developed.

For the outer dimensions to be part of the optimization, the designer is required to have an objective function which includes the penalty from increased volume and size of ducting, headers or plenums. Otherwise, the flow transversal dimensions would likely increase to unreasonable values to reduce the flow velocities inside the heat exchanger, and thereby the pressure losses. The only trade-off is the heat transfer coefficient, which is also reduced, but the trade-off is most likely worth it.

That leaves the geometric generalization parameters ($\sigma_r$, $\alpha_r$, and $\chi$), the recommendation is to allow all of these to vary, and the only exclusion would be if it was predetermined that the heat exchanger would be an unfinned design in which case the heat transfer surface area on both sides would be equal ($\alpha_r \approx 1$).

### 4.1.1 One fluid side set

Consider a design case in which one fluid stream is predetermined. One example could be for a finned tube design where the number of tubes and their diameter are already set and the target is to determine the geometries of the fins on the external side. If the tube internal fluid is denoted $f_2$, then that would mean that $\sigma_{f_2}$ and $\alpha_{f_2}$ are set. The most intuitive would then be to choose $\chi$ as unknown, in which case we calculate $\sigma_r$ and $\alpha_r$ as:

$$\sigma_r = \frac{1-\chi}{\sigma_{f_2}} - 1 \tag{30}$$

$$\alpha_r = \frac{2\chi}{t\alpha_{f_2}} - 1 \tag{31}$$

## 5 Design case example

So far, the GenHEX framework has been used in four design studies. The first is presented in the same paper in which the GenHEX method was first introduced [2], it regarded the design of an air-air intercooler for an aircraft engine and includes a validation case in which the generalized parameters were translated into a cross-corrugated plate heat exchanger that was validated against experimental performance data. Then followed two design studies where GenHEX was used to estimate the aerothermal performance of heat exchangers implemented in larger system models, one aimed at designing a hydrogen fuel cell aircraft in SUAVE [6] and the other was concerned with the design of a hydrogen-fueled composite cycle engine [7]. In the latest design study [*submitted*] a hydrogen enhanced intercooling stack for the composite cycle engine was sized, where an air-air and an air-hydrogen heat exchanger are coupled in series to achieve far greater thermal load than would be feasible using only bypass air or hydrogen fuel alone as coolant.

In the first design study, a cross-corrugated plate design was chosen since relatively short values for $\ell$ were used in both fluid streams and in the optimization of generalized parameters, concluded that $\alpha_r \approx 1$ would minimize the objective function. Hence, equal heat transfer surface area in both fluid streams and rather disruptive geometry were sought. One could have had a plate heat exchanger, such as strip fin, with fins protruding into both fluids as well. However, fins on both sides would result in a reduced surface efficiency which could be quite severe if a low thermal conductivity solid material were used for the fins.

The latest design study [*submitted*] concerned a hydrogen-enhanced intercooler for a composite-cycle engine running on hydrogen [8, 7, 9, 10, 11, 12]. Two heat exchangers were mounted in series, where the first used air extracted from the bypass steam as coolant, and the second used hydrogen fuel as coolant. That publication presents the dimensionless performance for the combined heat exchangers depending on the amount of air extracted from the bypass and where in the core pressurization process the intercooler is positioned. For every set of engine design parameters, an optimization is performed to find the most suitable heat exchanger.

| Parameter | Air-air | Air-H2 |
|---|---|---|
| $L_x, L_y, L_z$ [m] | 0.2, 0.2, 3.46 | 0.2, 0.1, 2.98 |
| $t$ [mm] | 0.2 | 0.2 |
| $\ell_{f_1}, \ell_{f_2}$ [m] | 0.01, 0.01 | 0.01, 0.1 |
| $l_f/\sqrt{t_f}$ [m$^{.5}$] | 0.32 | 0.32 |
| $\sigma_r$ | 0.6 | 83 |
| $\alpha_r$ | 1 | 4 |
| $\chi$ | 0.09 | 0.12 |

*Table 1: Parameters used for sizing of heat exchangers in hydrogen-enhanced intercooler [submitted]*

The suggested GGPs for the air-air heat exchanger, along with the remaining setup parameters, are listed in Table 1 for a coolant flow ratio of 0.8 and a pressure split exponent of 0.5 [3]. The air-air heat exchanger has the core stream (denoted $f_1$) flowing along the y-axis and the internal bypass stream (denoted $f_2$) flowing along the x-axis. This results in a cross-flow heat exchanger, and it is set up with two passes in the overall counter-flow configuration. The air-$H_2$ heat exchanger is mounted downstream and has the core stream (still denoted $f_1$) flowing along the y-axis, the hydrogen fuel (denoted $f_2$) is routed in cross flow configuration, along the x-axis, with four passes.

Since the air-air heat exchanger is suggested with $\alpha_r = 1$, we design an unfinned configuration, and since $\ell_{f_1} = \ell_{f_2} = 0.01$ m there should be an equal and rather short undisturbed flow length in each fluid stream. Some wavy tubes, wavy plates, or a cross-corrugated plate heat exchanger then seems suitable. The air-$H_2$ heat exchanger instead has $\alpha_r = 4$, which means that it should be a finned design where the engine core stream should have four times the heat transfer surface area. Below are example calculations for the air-$H_2$ heat exchanger, translating the generalized expressions into a continuously finned tubes heat exchanger.

Using the values in Table 1 and Equations (26-29) the first iteration yields $d_i = 0.2$ mm. This tube size is not feasible due to the high risk of clogging, high cost, and manufacturing difficulties. Increasing the inner diameter to 5 mm we get the following.

$$\begin{aligned} d_i &= 5 \text{ mm} \\ p_l &= 10 \text{ mm} \\ p_t &= 187 \text{ mm} \\ p_f &= 81 \text{ mm} \end{aligned}$$

The wide spacing of these tubes means that the air flow will probably only "sense" one tube at a time, hence the value of undisturbed flow length should probably be more related to the long fins than the tubes. Changing $\ell_{f_1} = p_l$ from 10 mm to 30 mm results in

$$\begin{aligned} d_i =& 5 \text{ mm} \\ p_l =& 30 \text{ mm} \\ p_t =& 62 \text{ mm} \\ p_f =& 81 \text{ mm} \end{aligned}$$

where we note that the transverse tube pitch has reduced and the fin pitch remains unchanged. This makes sense since we still aim to pack the same number of tubes, but just in another arrangement the relative heat transfer surface areas stay the same.

For the sake of completeness, we continue to calculate some other features of the heat exchanger. The void fraction ($\sigma$) and the surface area density ($\alpha$) for each stream are calculated as

$$\begin{aligned} \sigma_{f_2} =& \frac{1-\chi}{\sigma_r+1} = 0.01 \\ \sigma_{f_1} =& 0.87 \\ \alpha_{f_2} =& \frac{2\chi}{t\,(\alpha_r+1)} = 240 \text{ m}^2/\text{m}^3 \\ \alpha_{f_1} =& 960 \text{ m}^2/\text{m}^3 \end{aligned}$$

Then, the frontal and free-flow areas for each stream can be calculated. Note the factor 1/4 for the frontal area of the $H_2$ stream, because it has 4 passes.

$$\begin{aligned} A_{fr,f_2} =& L_y L_z/4 = 0.075\text{m}^2 \\ A_{ff,f_2} =& \sigma_{f_2} A_{fr,f_2} = 0.00075\text{m}^2 \\ A_{fr,f_1} =& L_x L_z = 0.60\text{m}^2 \\ A_{ff,f_1} =& \sigma_{f_1} A_{fr,f_1} = 0.52\text{m}^2 \end{aligned}$$

We can then calculate the total number of tubes ($n$) of the secondary fluid as

$$\begin{aligned} n =& 4\frac{A_{ff,f_2}}{\pi\,(d_i/2)^2} = 152 \\ & \textbf{or} \\ n =& \frac{0.1}{p_l}\frac{2.98}{p_t} = 160 \end{aligned}$$

which can be considered to be within the margin of rounding errors.

In hindsight, one can conclude that this heat exchanger could probably be made smaller and more compact because of the rather wide spacing of the fins and tubes. However, during the study, there was no objective function to estimate the impact of the heat exchanger volume, and therefore the outer dimensions could not be part of the optimization.

## 6 Concluding remarks

Most easy-to-use methods for estimating heat exchanger performance account only for the aerothermal performance, i.e. heat transfer and pumping power. When sizing systems for aviation, it has proven insufficient due to the high impact of weight and volume on system performance. Therefore, a more wholesome design approach has been sought in which the total performance is estimated, including heat transfer, pumping power, weight, and volume. In essence, a large and heavy heat exchanger is believed to have a high aerothermal efficiency, whereas a more compact design sacrifices some additional pumping power for reduced volume and/or increased weight. The GenHEX framework is promoted because it is capable of estimating the total performance using a rather low number of parameters, while still retaining enough information about the geometry to allow guiding the designer towards what a suitable heat exchanger for their application should look like. The geometry of the matrix is governed by three dimensionless generalization parameters, which in essence determine the total heat transfer surface and distribute it and the fluid volume between the two fluid streams. This paper aims to further increase the general understanding and usefulness of GenHEX by thoroughly presenting the parameters used for the geometrical generalization, providing guidelines for the selection of heat exchanger families, and present equations for translating the generalized parameters to actual design features of a selection of common heat exchanger types.

## References


[1] Ramesh K Shah and Dusan P Sekulic. *Fundamentals of heat exchanger design*. John Wiley & Sons, 2003.

[2] Petter Miltén, Isak Johnsson, Anders Lundbladh, and Carlos Xisto. Generalized method for the conceptual design of compact heat exchangers. *Journal of Engineering for Gas Turbines and Power*, 146(11), 2024.

[3] Petter Miltén. Conceptual design of the hydrogen-enhanced intercooler. *Licentiate thesis Chalmers*, 2025.

[4] William Morrow Kays and Alexander Louis London. *Compact heat exchangers*. McGraw-Hill, New York, NY, 1984.

[5] R.K. Sakhuja P.G. LaHaye, F.j. Neugebauer. A generalized prediction of heat transfer surfaces. *Journal of Heat Transfer*, NOVEMBER 1974.

[6] Christian Svensson, Petter Miltén, and Tomas Grönstedt. Modelling hydrogen fuel cell aircraft in suave. In *Proceedings of the 34th Congress of the International Council of the Aeronautical Sciences*, 2024.

[7] Adam Johansson, Petter Miltén, Anders Lundbladh, and Carlos Xisto. Modelling a hydrogen fuelled composite cycle aeroengine. In *Proceedings of the 34th Congress of the International Council of the Aeronautical Sciences*, 2024.

[8] Sascha Kaiser, Hagen Kellermann, Markus Nickl, and Arne Seitz. A composite cycle engine concept for year

2050. In *Proceedings of the 31st Congress of the International Council of the Aeronautical Sciences, Belo Horizonte, Brazil*, pages 9–14, 2018.

[9] Sascha Kaiser, Arne Seitz, Stefan Donnerhack, and Anders Lundbladh. Composite cycle engine concept with hectopressure ratio. *Journal of Propulsion and Power*, 32(6):1413–1421, 2016.

[10] Sascha Kaiser, Arne Seitz, Patrick Vratny, and Mirko Hornung. Unified thermodynamic evaluation of radical aero engine cycles. In *Turbo Expo: Power for Land, Sea, and Air*, volume 49682, page V001T01A011. American Society of Mechanical Engineers, 2016.

[11] Sascha Kaiser, Oliver Schmitz, and Hermann Klingels. Aero engine concepts beyond 2030: Part 2—the free-piston composite cycle engine. In *Turbo Expo: Power for Land, Sea, and Air*, volume 84140, page V005T06A018. American Society of Mechanical Engineers, 2020.

[12] Sascha Kaiser. *Multidisciplinary Design of Aeronautical Composite Cycle Engines*. PhD thesis, Technische Universität München, 2020.