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THE IMPACT OF TUBE CORRUGATION WITHIN THE MULTI-DISCIPLINARY DESIGN OPTIMIZATION OF A CHARGE AIR COOLER.

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Key words: Multi-disciplinary Optimization, Charge-air Coolers, Heat-transfer, Heat-exchangers, corrugated tubes.

Abstract. The following paper explores the impact of corrugated tubes within a charge air cooler (CAC) on overall cooler performance, cost and size, for the first time. Corrugated tubes have been demonstrated to perform better in terms of heat transfer, when compared to a smooth tube [2], however they have not been optimized in the context of a CAC. In this study, a CAC with corrugated tubes is compared against a similar system comprising of smooth tubes as a baseline design. Both CACs have common design parameters, such as number of tubes per rows, number of rows, number of passes, fins per meter, fin material, and tube material, while two additional design parameters exist i.e., groove depth, and pitch for the CAC with corrugated tubes, that characterizes the helical corrugation. These two systems are optimized to minimize manufacturing cost where cost is a function of cooler dimensions and material selection. Feasible designs are then obtained by satisfying dimension, pressure, weight, performance and vibrations based constraints. A vibration constraint introduced here is an addition to the current state of the art [3], making this approach, a multi-disciplinary one and the first of its kind. Finally, the optimum is compared which signifies the importance of a multi-disciplinary analysis for both cooler configurations.

Nomenclature

Abbreviations

CAC Charge air cooler

$FTHE$ Fin and tube Heat Exchanger

KPI Key performance indicator

LHS Latin Hypercube Sampling

MDO Multi-disciplinary optimization

$PFHE$ Plate and Fin Heat exchanger

$SHTE$ Shell and Tube Heat Exchangers

Math symbols

δ Thickness

γ_e Fins per meter

ρ Density

ff	Friction factor
k	Thermal conductivity
L	Length
Nu	Nusselt number
f	Frequency
Q	Power dissipation
Re	Reynolds
T	Temperature
V	Volume
v	Velocity
W	Weight

Subscripts

a	air, air side
f	fins
i	inner, inside
o	outer, outside
p	plate
r	row
t	tube, tube side

1 INTRODUCTION

An internal combustion engine, emits exhaust gases at a very high temperature. Such exhaust gasses, have a high kinetic as well as thermal energy, sufficient enough to drive a turbocharger. However, the gas exiting the turbocharger has not lost all of its thermal energy, as it is still at higher temperature. This makes it unsuitable to resupply to the engine to derive work out of it. A charge air cooler (CAC) is one such system, which is placed after a turbocharger and before an engine intake, so that the temperature of this charge air could be brought down. This is beneficial because low-temperature, high density charged-air is better suited for combustion compared to a high temperature, low density air, as would be the case if a CAC weren't used. This helps the engine intake to get a resupply of the charge air with higher density, which in turn helps for an efficient combustion process within the engine. The usual temperature difference achieved within a CAC ranges anywhere between 180-280°C from a turbocharger exit to 30-50°C at the CAC exit. Thus, it is an important piece of equipment, required for an efficient functioning of, not only for the supercharged engines of the automotive sector, but also the larger output engines used on ships, and industrial power plants (see figure 1). Optimization of such important engine sub-systems therefore becomes very important.

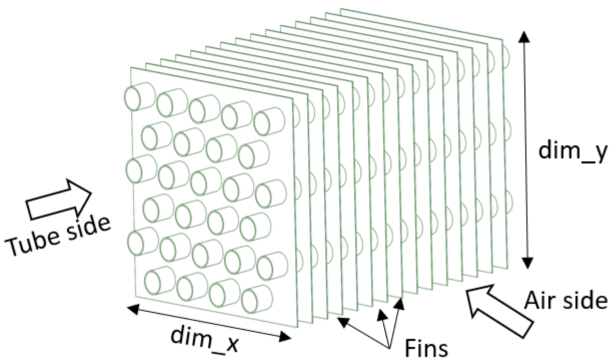

Figure 1: Schematic of a simple Charge air cooler (Not to scale)

Heat exchangers have been a subject of wide variety of optimization studies, as there are plethora of variables that affect the efficiency of the system. For instance Shell and Tube Heat Exchangers (SHTX) are a very general class of heat exchanger optimized for cost and exergy [1]. Compact heat exchangers, or Fin and Tube Heat exchangers (FTHE) are another class of heat exchangers, optimized through various approaches such as Genetic Algorithms[2] and [3]. The reader is directed for an exhaustive review on various such approaches and methods on heat exchangers in Rao et al. [4].

Within heat exchangers too, there has always been an interest in methods to augment heat transfer efficiency of the system, either by active methods, in which there is an external excitation to the system in order to increase heat transfer, or, through passive methods, where surface or topological improvements are combined together. We limit ourselves to passive methods in the current discussion. The reason to do this has been documented exhaustively in various studies trying to improve the Nusselt number and decrease the friction factor ([5, 6, 7]) of the heat exchanger. It is also well documented that, the corrugated tube, increases the heat transfer efficiency, at the cost of increased pressure drop [8].

Within the literature, one of the assumptions that is commonly observed, is that the fin and tube materials are the same, or there is no choice between them. This is not the case in practice, as a variety of different material choices exist for both fins and tubes. The current paper introduces material selection as a design parameter which has, so far, been neglected in similar heat exchanger optimizations within the literature.

Another gap observed within the literature, to the best of the author's knowledge, is the complete lack of multi-disciplinary approaches within heat exchanger design studies. As the charge air cooler is subject to various sources of vibrations, this makes it prone to cracks and other vibration failures. Attempts have been made to perform multi-objective optimization studies [9] on the heat exchangers however, unlike the current approach, these have neglected any vibration analysis.

The novel contributions of the current paper are summarized as follows,

1. Materials for tubes and fins have been factored into the optimization routine as design variables. (See table 2).

2. The optimization is defined using a mixed integer variable problem, which means, that some of the design parameters are integer variables ($N_t, N_r, N_{plates}, \gamma_e, k_f, k_t, N_{pass}$), while some are defined continuous range (L, e, p), which is an improvement to the existing literature.

3. Corrugated tubes have been optimized individually before [10, 6], however there are limited studies for specific type of heat exchangers, such as a double pipe heat exchangers [7], and multi-tube heat exchangers [11] which uses corrugated tubes instead of smooth tubes for the whole system.

4. The current study also obtains the first natural frequency of the tube bundles present in the charge-air cooler arrangement and, for the first time, uses these as a constraint within a multi-disciplinary optimization.

2 Mathematical models

2.1 Thermal model

The overall heat transfer coefficient is obtained as,

$$U = \left(\frac{1}{\eta_0 h_a} + \frac{\ln\left(\frac{Dt_o}{Dt_i}\right) A}{2\pi k_t N_t L_t} + \frac{1}{h_t} \right)^{-1}, \tag{1}$$

where η_0 is the fin surface effectiveness, k_t is tube conductivity, h_a is the air side heat transfer coefficient, h_t is the tube side heat transfer coefficient, and A is the area. The overall heat exchanger performance is examined using cross-flow effectiveness given as, $NTU = UA/C_{min}$. The air side and tube side outlet temperature (T_{t_out}), is iteratively calculated as,

$$T_{t_o} = T_{t_i} - (NTU)\frac{C_{\min}}{C_t}\left(T_{t_i} - T_{a_i}\right), \tag{2}$$

and,

$$T_{a_o} = T_{a_i} + (NTU)\frac{C_{\min}}{C_a}\left(T_{t_i} - T_{a_i}\right), \tag{3}$$

where C_{min}, C_{max} are minimum and maximum heat capacity rates respectively.

2.1.1 Tube side

The equations to calculate various thermal parameters necessary for the tube side of the CAC in the current study, have been outlined below, for both smooth and corrugated tubes.

$$ff_{smooth} = 0.001375\left(1 + \left(\frac{\kappa}{Dt_i}20 + \frac{10^6}{\mathrm{Re_t}}\right)^{\frac{1}{3}}\right),$$

$$ff_{corrugated} = 1.15Re^{-0.239}(\frac{he}{D_H})^{0.179}(\frac{hp}{D_H})^{0.164},$$

$$Nu_{smooth} = \frac{\frac{ff_t}{2}(Re_t - 1000)Pr}{1 + 12.7\left(\frac{ff_t}{2}\right)^{0.5}\left(Pr^{\frac{2}{3}} - 1\right)},$$

$$Nu_{corrugated} = 1.579Re^{0.639}(\frac{he}{D_H})^{0.46}(\frac{hp}{D_H})^{0.35}Pr^{0.3}.$$

Heat transfer capability of corrugated tubes is based on the Nusselt number and friction factor correlations as a function of of groove depth (he) and groove pitch (hp)[5], valid for $5500 \leq Re \leq 60000$, $0.18 < hp/D_H < 0.27$ and $0.02 < he/D_H < 0.06$. κ is the surface roughness, $\mathrm{Re_t}$ is the Reynolds number of the tube side and Pr is the Prandtl number. Whereas, the $\mathrm{ff_{smooth}}$ and $\mathrm{Nu_{smooth}}$ are valid between the range of $4000 \leq \mathrm{Re} \leq 50^6$.

The pressure drop, which are observed as an output metric, is given as,

$$\Delta P_t = \frac{G_t^2}{2\rho_{ti}}\left(\frac{4(ff_t)L_t N_{pass}}{D_{ti}} + 1.4N_{pass} + 2(N_{pass} - 1)\right), \tag{4}$$

where G is the mass flux through the tubes, first term is the pressure due to the core friction which is the most dominating term, while the second and third term accounts for entrance and exit effects[12].

2.1.2 Air side

The surface effectiveness of the fins can be written as,

$$\eta_{overall} = 1 - \frac{A_f}{A}\left(1 - \eta_f\right), \tag{5}$$

where A_f is the fin surface area, A is the total surface area and η_f is the fin efficiency. This is obtained as, $\eta_f = \tanh(Bi)/Bi$, with Bi as the Biot's number.

The air side pressure drop is calculated as,

$$\Delta P_a = \frac{G_a^2}{2\rho_a}\left(\frac{4(ff_t)X_r N_r}{D_{ha}}\right), \tag{6}$$

where X_r is the row pitch for the tubes, and G_a as mass flux of air side flow, while Dh is the hydraulic diameter.

2.2 Vibration model

In order to assess the multi-disciplinary aspects of a fin and tube arrangement of a charge air cooler, the formula to calculate frequency of this system is presented in this section. From a vibrations perspective, the same decision variables of material, row size, column size, fins per meter, etc are used to obtain the frequency of the system. We are interested in the first natural mode of the bending frequency, because, engine manufacturers require this to be above a certain range, in this case, above 100 Hz for a four-stroke engine, and 50 Hz for a two-stroke. The following formulation is made under the assumption that the bending modes can be approximated using Euler-Bernoulli theory, with an added weight due to the fins, however, the weight of the fluid has not been included.

Regarding the support plates as variables, addition of one support plates, divides the tubes into two tubes of equal length, and so on, and obtain the frequency accordingly. The position of the support plate is not considered as a variable, and the addition of support plates from the optimizer will simply split the tubes of equal lengths. Although the vibration model, considers the thickness of the support plate as negligible, it is however, important for the cost based objective function, which relies on material, for which the thickness is not ignored. The frequency is given by,

$$\text{frequency}_{\text{bending}} = \left(\frac{1}{2\pi}\right)\sqrt{\frac{E_{tube} I_{tube} B}{\mu_{\text{tube}} A + \mu_{\text{fin}} \frac{b}{\text{L}} A}}, \tag{7}$$

where A & B are constants, b is tube pitch, E_{tube} are the Young's modulus of the tube materials, I_{tube} are the the second moment of area of the tube, while μ_{tube}, μ_{fin} are the area densities, L is the section of tube length.

3 Problem statement

3.1 Design parameters

To define the problem, we first discuss the design variables used in the present study. These are, number of tubes per rows (N_t), number of rows (N_r), number of passes (N_{pass}), fins per meter (γ_e), fin material (k_f), tube material (k_t), number of plates (N_{plates}), length of the tube (L_t). Two other parameters of groove depth (he) and groove pitch (hp) are defined as parameters for corrugated tube. The various details of these parameters are summarised in table 1. These are treated with a mixed variables LHS sampling, i.e. they comprise of both integer and continuous variables, which are commonplace in engineering applications.

Design Parameter Shorthand	Parameter name	Lower bound	Upper bound	Additional note
N_t	Number of tubes per rows	26	32	Integer variable
N_r	Number of rows	17	34	Integer variable
N_{pass}	Number of passes	2	8	[2,4,6,8], Integer variable
γ_e	Fin frequency	373	561	Integer variable
k_f	Fin material	1	3	1=Aluminium, 2=Copper, 3=Stainless Steel, Integer variable
k_t	Tube material	1	3	1 = CuNi10, 2 = Copper, 3 = Stainless Steel, Integer variable
N_{plates}	Number of plates	0	4	Integer variable
L_t	Length of tube	1	2.5	Continuous variable
he	Groove depth	0.02	0.06	Continuous variable
hp	Groove pitch	0.18	0.27	Continuous variable

Table 1: Design parameters with their bounds and variable type, generated with a mixed-integer, Latin hyper cube sampling. The last two design parameters i.e. e and p are only used in the corrugated tube CAC optimization, while the rest of the design parameters are used as design parameters in both, corrugated as well as smooth tube CAC.

3.2 Constraints

The optimization problem is defined with a single objective, which is to minimise the materials based cost. In addition to this, several constraints are imposed. The constraints are defined such that, the horizontal dimensions of the bounding box of the charge air cooler shouldn't exceed 1.054m (dim_x), the vertical dimension shouldn't exceed 0.8m (dim_y), the pressure inside tube bundles must not exceed 80kPa (ΔP_t), the pressure across tube bundle should not exceed 2.5kPa (ΔP_a), weight of the system shouldn't exceed 600kg (W), coolant velocity inside tubes shouldn't exceed 1.5m/s (v_t), and Reynolds number should be limited under 60000 (Re_t). The power dissipation of the cooler should exceed 4120 kW (Q), while the first natural bending frequency (f) as defined in section 2.2 should be higher than 100 Hz.

Constraints dim_x and dim_y are size based constraints, responsible for meeting the limitations on the engine dimensions. ΔP_t and ΔP_a are pressure based constraints, that are minimum allowable pressure drop for a set of input conditions. This KPI is an important metric for design approval from both manufacturer's and customer's perspective. A limit on Re_t is imposed in order to make an even comparison with the corrugated tube, as the correlations employed for the Nusselt number and Reynolds number from [5] are valid only under 60,000.

3.3 Objective Function

Both the tube configurations, i.e. smooth and corrugated tubes have been assumed of the same cost, due to the lack of data to build a cost model in order to capture the additional expense of grooving. However, the results discussed are given relative to the smooth tube, which can augment the costing data, when available. Finally, other material related constants are defined in table 2, cost is then, defined as a function of materials that comprise of the tube bundles, fins and support plates.

$$cost_{tubes} = \rho_t V_t N_t (cost/kg)_{tubes} \tag{8}$$

$$cost_{fins} = \rho_f V_f \delta_e L_t (cost/kg)_{fins} \tag{9}$$

$$cost_{plates} = \rho_p V_p N_{plates} (cost/kg)_{plates} \tag{10}$$

$$cost_{CAC} = cost_{tubes} + cost_{fins} + cost_{plates} + misc \tag{11}$$

The value of $misc$ has been added as an offset to incorporate miscellaneous expenses. It is important to note, that the objective function in this study, differs from various approaches available in the literature ([1], [2]), where optimal costs are obtained by adding manufacturing costs and operating costs, that appreciate with an interest rate over some life span of the system. The reason to do this is to simply conduct a design analysis before a design-freeze is agreed between manufacturers and customers. As this is not a trivial task, both parties require the answers to a few questions such as, what is the optimal number of columns, rows, and passes of tubes that should be arranged within a specific dimensions allocated for a cooler around an engine configuration, while satisfying all the predefined KPI (in this case, the constraints). The operation cost over the life span is not a desired quantity during the design phase for the manufacturer. This formulation is hence, intended to improve the design decision making abilities for the manufacturers, which is why, an operating cost has not been considered.

Lastly, the minimization of cost makes the current approach a single-objective, multi-disciplinary, mixed-integer, constrained optimization problem, which is solved using an NSGA-II [13] algorithm ([14]), with a population size of 1000, ran over 800 generations. It should be noted that the reason to apply NSGA-II on a single objective problem, boils down to the fact that NSGA-II also handles the constraints in a non-dominated fashion. i.e. when two designs are contested for feasibility, there are 3 possibilities that arise. (1) both designs are feasible, (2) one of the designs is feasible and the other is not, or (3) neither of the designs are feasible. This is especially

Material property	Aluminium (Al)	Copper (Cu)	Stainless Steel (SS)	Copper Nickel (CuNi10)
fin or tube conductivity (W/mK)	200	384	16	40
Density(kg/m^3)	2700	7940	8000	8940
Cost/kg ($/kg)	7.168	11.37	17.24	15.79
Young's modulus (N/m^2)	69e9	110e9	200e9	140e9
Used as fin material	Yes	Yes	Yes	No
Used as tube material	No	Yes	Yes	Yes
Used as plate material	Yes	Yes	Yes	Yes

Table 2: Material properties used for sampling as well as for objective function evaluation.

important when two solutions are ranked according to their constraint violation. i.e the smaller the constraint violation, the better will be the rank of the contested design [13].

4 Results and Discussion

The following results are based on certain initial conditions, such as charge air inlet pressure and mass flow rate of 365 kPa and 27.11 kg/s respectively. The charge air inlet temperature is at 189.9 °C, while the coolant flow rate and inlet temperature are 3682.3 lit/min and 25 °C. Two types of charge air coolers are then compared where the first CAC comprises of smooth tubes outlined in Part 1, while the second, comprises of corrugated tubes is discussed in part 2. Both theses coolers are applied with a constraints for frequency (f), other than those which remain common for both the cases. This leads to four cases namely, smooth tube without vibration constraint, (baseline design), smooth tube with vibration constraint, corrugated tube without vibration constraint, and corrugated tube with the vibration constraint.

4.1 Part 1: Effects of performing MDO analysis

To begin with[15], the optimum for these four cases as seen from table 3, makes the design expensive by 7.56% for the smooth tube configuration and 5.52% for the corrugated tube configuration. (refer figure 2). In both these cases, it is observed that majority of the KPI remains the same, except for the mass, and the frequency, as seen in table 4, where each of the four columns represent the optimum of our cases.

The increase in weight, after imposing the vibration constraint, is due to the addition of the support (N_{plates} variable in table 3), required to achieve a minimum frequency of 100 Hz. When this constraint is not imposed, the optimum is cheaper, but without any support plate, which would make the design prone to vibration failure.

The frequency obtained as a result of keeping the vibration constraint active, signifies the importance of a multi-disciplinary approach, as this enabled the optimizer to factor in the requirement, not immediately discernible in a non-multi-disciplinary approach. If the constraint were inactive, the optimum achieved, would miss a real world scenario design requirement, and a separate analysis would have been required to find the appropriate number of plates in order to

	Smooth tube	Smooth tube	Corrugated tube	Corrugated tube
Optimum value of	with frequency constraint	without frequency constraint	with frequency constriant	without frequency constraint
N_t	26	26	27	27
N_r	25	25	22	22
N_{pass}	2	2	2	2
γ_e	475	475	474	474
k_f	Al	Al	Al	Al
k_t	Cu	Cu	Cu	Cu
N_{plates}	2	0	1	0
L_t	1.623	1.624	1.44	1.44
he	-	-	0.06	0.06
hp	-	-	0.269	0.269
Cost ($)	5871.43	5458.74	3805.61	3617.11

Table 3: Optimum parameters and the objective of the smooth tube and corrugated tube configurations with and without the frequency constraint.

	Smooth tube	Smooth tube	Corrugated tube	Corrugated tube	Constraint
KPI	with frequency constraint	without frequency constraint	with frequency constraint	without frequency constraint	Value
ΔP_a	2498.98	2498.99	2496.53	2496.27	2500
ΔP_t	9.69	9.69	16.53	16.53	80
Re_t	24006	24006	26270.14	26270	60000
W	487.16	446.61	380.77	362.28	600
γ_e	0.0019	0.0019	0.0019	0.0019	-
f	186.24	20.69	104.95	26.23	100
ff	0.0062	0.0062	0.0123	0.0123	-
Nu	150.9	150.9	496.52	496.4	-

Table 4: Comparison of some KPI for the smooth tube and corrugated tube configuration, while keeping the vibration constraint active and inactive, with constraint values.

find a design meeting a criteria of first natural frequency of 100Hz. Using the current approach, such an optimum design, best suited for both the disciplines, is achieved in one go. Table 3 highlights the optimum cost and parameters obtained for both the cases. One can find that the optimal material for the fins is aluminum, which can be explained from cost/kg table 2, that Aluminum being the cheapest, is the preferred fin material. The same is true for copper as tube material, as this is the cheapest material available for tubes.

4.2 Part 2: Effects of performing MDO analysis using corrugated tubes

In literature, there are various studies, that finds the optimal groove depth and pitch parameters to achieve the maximum heat transfer while trying to minimize the pressure drop ([10]), however these have only focused on the tubes rather than their performance in the context of a whole heat exchanger. Hence, in the current approach, the corrugated tube parameters have now been added to the already existing 8 design parameters of the smooth tube CAC configuration, to improve on this aspect.

The results are indicated in table 4, which lists out the KPI for a corrugated tube CAC with and without the vibration constraint imposed on the system. When compared against each other, it is observed that N_{plates} is a non-zero value when the constraint is active. This is similar to the result found for the smooth tube, that a support plate is required to cross the minimum frequency of 100Hz. However, no such plates are found in the optimum where the constraint is inactive. Consequently, the frequency of the system in this case is below 100Hz. However, when the results are compared with that of a smooth tube configurations, it is found that the number of tubes reduces in the corrugated tube configuration, while still satisfying the constraints imposed by the smooth tube CAC system. This implies that the corrugated tube CAC is able to achieve same heat dissipation (Q) and a better overall heat transfer coefficient (U) compared to the smooth tube CAC system, but with fewer number of tubes [1], of shorter length.

KPI	% difference between Smooth CAC and Corrugated CAC with the vibration constraint.	% difference between Smooth CAC and Corrugated CAC without the vibration constraint.
cost	-35.1%	-33.73
ΔP_a	-0.1	-0.1088
ΔP_t	70.5	70.58
Re_t	9.43	9.42
γ_e	0	0
f	-43.64	26.77
ff	98.38	98.3
Nu	229.03	228.95

Table 5: Comparison between some important KPI between smooth and corrugated tube CAC with and without the vibration constraint. (all KPI are in SI units)

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5 Conclusions

The current paper represents the first time, that a CAC has been optimized to reduce cost using a multi-disciplinary optimization approach. It is the first time that a corrugated tube has been optimized within the context of a heat exchanger in which the tube is optimized factoring in the design variables of the cooler as well. The multi-disciplinary approach is also novel, with materials as design variables also finding its way into an optimization study for a heat exchanger for the first time. The results presented are briefly summarized in the figure 2. When a vibration constraint is included, the designs become more expensive by 7.5% and

[1] $N_t \cdot Nr$ = Number of tubes
650 for smooth tube CAC, 594 for corrugated tube CAC, as seen from Table 3.

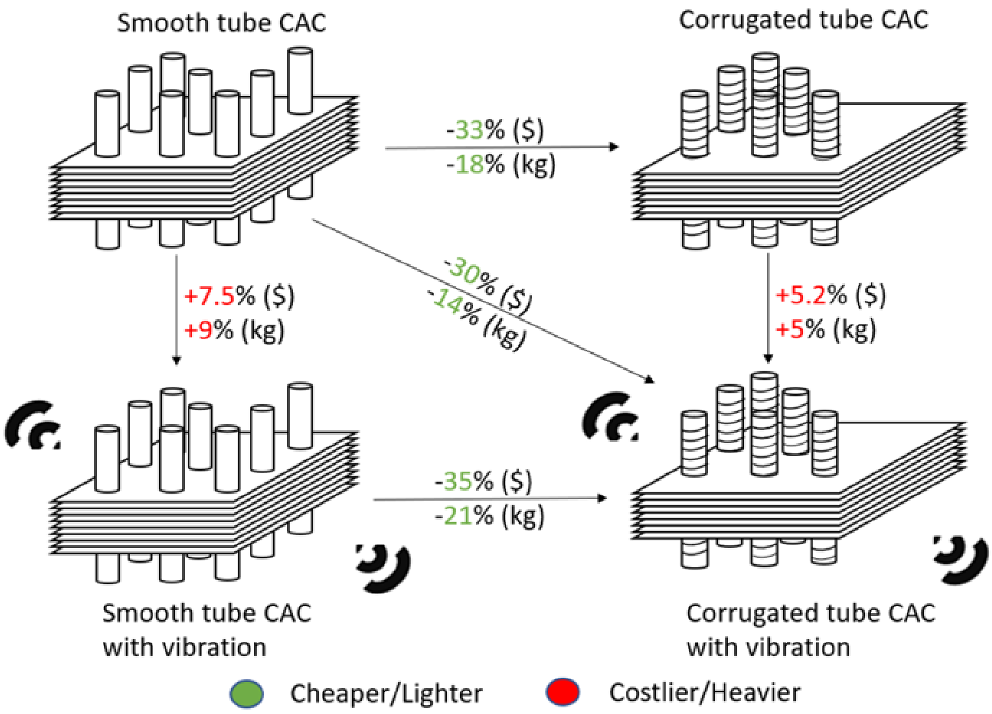

Figure 2: Optimum between all the configurations compared with each other. Left to right, swaps smooth tube with corrugated tubes, while top to bottom switches on the frequency constraint.

heavier by 9%. Switching to corrugated tubes for CAC from the baseline design, makes the design 33% cheaper and 18% lighter, while meeting the vibration constraint makes the design 5.2% more expensive and 5% heavier. Overall, a 30% cheaper and 14% lighter design is achieved if, corrugation and multi-disciplinary optimization is performed on the baseline design. These results indicate there are considerable advantages to employing a multi-disciplinary approach to a constrained, single objective, multi-variable, mixed-integer optimization problem.

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