Air to Air Heat Exchanger Mimicked from the Directional Passive Water Transport Featured by the Skin Morphology of Phrynosoma Cornutum

SÁNCHEZ Federico¹ and ZHU Shifan¹

¹College of Mechanical and Electrical Engineering, Harbin Engineering University, China

Abstract. A biomimetic approach is proposed to improve the performance of an intercooler using only the passive water transport featured by the lizard species Phrynosoma Cornutum for racing applications. The lizard is able to transport in a preferred direction, water over its integument whilst consuming no energy to do so. That feature is used in the external surface of the intercooler and an evaporation rate is desired. The model started with the air conditions after the turbo charger stage, then the convection of air in the heat exchanger is modelled by the boundary layer theory which is applicable for the scale used justified by the Péclet number value, for both the heat and mass transfer. The convection heat transfer was satisfactorily modelled and the values reached inside the expected values for the optimal 2 mm wall thickness selected to demonstrate the enhancement due to the mass transfer and a surface temperature of 62.02 °C was stabilized for the conditions selected under a few iterations. According to the mass convection theory a mathematical model of the heat exchange was made and for the heat lost there exist a drawback due to a deviation from real values due to the highly turbulent flow under the conditions selected and the very small value of the Schmidt number obtained in the literature and pooled to the lack of a proper equation to compute the Sherwood number, which was not found and modelled outside the Schmidt number range. To solve this, a correction factor of the water evaporation is introduced which assumes that not all the water from the channels evaporates and some part is taken by the turbulent air as water droplets, but it needs further validation by real experiments, which is expected to be the next stage of the project.

1. Introduction

Natural systems have been existing over 3.8 billion years [1] of evolution, natural selection and constant improvement, when compared with human knowledge, the latter becomes insignificant. Life forms a technology in every proper sense, with a diversity of design, materials, engines, and mechanical contrivances of every degree of complexity [2]. Back in 1997, a woman spread and popularized the term Biomimicry [3] when she wrote her book and established a completely new field of research, by which many professionals including a wide range of experts were able to solve their problems while working interdisciplinary. The main idea is bring not only biologists but nature’s designs to the design table, while seeking sustainability and innovation [4].

According to Gleich et al. [5] Biomimetics have been developed in three main strands: the first and oldest one is functional morphology, form and function by which it’s favored the structure rather the
Biomimetics is the attempt to learn from nature; it deals with the development of innovations on the basis of investigation of natural, evolutionary optimized biological structures, functions processes and systems [5].

Research and development in biomimicry and biomimetism is measured by the Da Vinci Index 2.0, created in 2013 by the Fermanian Business & Economic Institute located in Point Loma Nazarene University in San Diego, California [6]. It is an improvement from the first Da Vinci Index created two years before by the same institution, based in number of scholarly articles, patents, grants and the dollar value of those grants. Innovation and sustainability and the main drives of these research areas [1]. In short, biomimicry can be defined as the innovation inspired by nature. [3, 4, 7] Some websites offer online solutions such as Biomimicry Institute [4], AskNature.org and the Sustainability Workshop of AutoDesk® [7] based on past user experience or designs.

Since the invention of the turbo engine systems, its functions have been always the same: to inject more fresh and compressed air into the engine in the same time lapse and to improve the peak performance of both power and torque output.

While the air is compressed its density and temperature raise accordingly to levels not really safe for the engine block because it will increase the temperature of the components of the engine, thus the cooling system will have to work harder in order to maintain a safe operation and to obtain the output values for which it was designed for.

Heat exchangers, called intercoolers have been used to chill the air temperature before entering the intake manifold preventing the phenomenon of engine knocking or pre-detonation of the fuel-air mixture inside the combustion chamber. This means that the fuel will ignite instants before the spark plug detonates it at the precise moment it was intended for. This precise moment is determined by the setup and tuning of the ECU (Engine Control Unit) which is a programmable chip that controls the timings and values of the engine. This knocking will damage the engine almost immediately because it counteracts the angular momentum of the crankshaft by pushing the piston against its angular velocity trying to stop it suddenly, acting as an engine brake stroke.

Intercoolers are radiators (more generally heat exchangers) that cool down the air that is going straight to the intake manifold of the engine the same way the radiator of the cooling system maintains the engine within its proper, normal and safe operating range of temperatures. The most common types of intercoolers are, air to air, air to water and the merge of those to types for this kind of application. It should be emphasized that what is cooled down inside the intercooler at any moment is the intake air before being injected to the engine, opposite as the radiator, where what is cooled is the coolant circulating the engine to maintain certain operating range of temperature.

To improve the effectiveness of cooling the air temperature within the intercooler using the same device model and the same time for the process, some car manufacturers like Subaru® [8] take advantage of a fine pulverized cold water stream, sprayed all over the front outer surface of the intercooler where a fresh stream of laminar flow air is passing through the space between the intercooler ducts at high speed. This water will extract some of the energy in form of heat from the hot and compressed air, raising its temperature until it reaches the saturation temperature of water at that specific pressure (depending on the atmospheric conditions where the intercooler is located) to start the evaporation process as a saturated mixture of water and water vapor.

Intercoolers work by means of forced external convection heat transfer occurring at the outer face of the heat exchanger. The stream of air is renewed constantly due to the movement of the car, taking away heat faster. The evaporation of water will take even more heat which will be sent far away from the intercooler by the same fast air stream, producing forced convection.

The injection of the pulverized water spray needs a high pressure pump, pipes and nozzles to achieve the proper pulverization and not just to leak water (which will be useless). This water conditions require to extract some power from the engine to drive the pump into the designed pressure of the system, causing a tiny loss of power in the engine from that extra power generated thanks to the system. At the end, the overall gain of power is more than that used to move the water injection system.
2. Methodology

2.1. Lizard Skin

The water over the Texas Horned Lizard (Phrynosoma Cornutum) flows at almost doubled velocity to the snout than to the tail in comparison. It was reported by Comanns et. al. in [9] that in the first 333 ms, after the water was in contact with the skin, the velocity towards the snout was $3.15 \pm 0.94 \, \text{mm/s}$ while the velocity towards the tail was $1.61 \pm 0.45 \, \text{mm/s}$. The passive water transport is possible over the lizard’s integument because of the morphology of the integument itself or skin, where there are periodic and asymmetric changes in the shape leading to abrupt widening’s and gradual narrowing’s of the capillary canals, this is one on the essential features of the skin. The second essential feature is: there are some interconnections between the channels in order to continue the halted stream of water when it reaches an abrupt widening [10], as can be clearly seen in Figure 3.

In the real integument of the Phrynosoma Cornutum, the dimensions of the capillary channels ranges from 30 µm to 300 µm. This phenomenon is possible due to the super hydrophilic surface the scales of the lizard has. It was said in the last section that the practical angle to separate hydrophobicity from hydrophilicity is 65°. The angle of contact between the water and the scales is smaller than 10° which makes the skin of the lizard super hydrophilic [10]. This feature improves the movement of water because it doesn’t let the skin to get wet while minimizing the loss of water due to evaporation.

2.2. Fluid as Ideal Gases

The saturated water vapor expected to be produced and the air, should be modelled, and the ideal gas is preferred over the real gases for obvious reasons. Nevertheless, one of the simplest methods available in the literature is the compressibility factor $Z$ that is a measure of the deviation from the ideal gas behavior suffered by all real gases. Real gases deviate severely from the ideal behavior near the critical point and also close to the saturation state in the specific volume vs. temperature diagram or thermodynamic v-T phase diagram. For the case of real gases, “gases behave differently at a given temperature and pressure, but they behave very much the same at temperatures and pressures normalized with respect to their critical temperatures and pressures. The normalization is done as” [11].

At low pressures regardless of the temperature of the gas, it behaves as if it were an ideal gas. At high temperatures, the ideal gas can be assumed to be accurate enough to be used as a model. Obviously the closer the state of a gas is to its critical point, greater will be the deviation of the ideal gas behavior due to the liquefaction of the gas at high pressure.

A simple explanation of the temperature of the critical point can be given by the liquefaction of gases, which accounts for the most of the deviation of real gases from the ideal gas behavior and “if a gas is below its critical temperature, it can be liquefied simply by compression. However, the critical temperature for most gases is very low” [12].

2.3. Turbo Compressor Stage

Now a way to obtain the temperature of the air after the compressor stage of the turbo should be implemented and the mathematical model used for the air of the air after the turbo compressor stage are given by [13]. Here is where we make use of the specific characteristic curve of an actual turbo given by a manufacturer [14], because data is required to input in the equation of temperature, pressure and mass flow of air after being compressed.

$$T_{out,turbo} = T_{in} + \frac{T_{in} \cdot [-1 + (P_{out}/P_{in})^{0.263}]}{Efficiency} \quad (1)$$

2.4. Heat Transfer for the Case of a Dry External Intercooler Surface

It is well known that intercoolers are air to air heat exchangers that work by means of internal and external convection under cross flow with no stream mixture at any time. This makes it very difficult to model accurately without the use of the empirical models given by [16, 17], and [15]. Let us remember the Nusselt number that compares the heat transfer enhancement of convection with respect to the
conduction phenomenon for the same temperature conditions of the wall and the fluid. \( L_c \) is the characteristic length of the problem.

Once the Nusselt number is obtained and the variables of state of the air, both inside and outside the intercooler duct are known, the temperature values of the air at the outlets of the external and internal air streams are computed by means of the Logarithmic Mean Temperature, and the equivalent thermal resistance method, using the modified Newton law of convection.

\[
\dot{Q} = UA_s \Delta T_{im}
\]  

(2)

Where \( \Delta T_{im} \) the usual logarithmic mean temperature difference and \( U \) is the heat transfer coefficient of convection obtained by equivalent thermal resistance method.

\[
\Delta T_{im} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1/\Delta T_2)}
\]  

(3)

The correction factor \( F \) for cross flow heat exchangers (inlet and outlet values of temperature of both hot and cold streams). It basically computes a transformation of the equivalent temperature difference of the cross flow to compare with the parallel flow, like a scale or conversion factor.

\[
\Delta T_{im} = F \Delta T_{im,cr}
\]  

(4)

The limiting value for \( F \) is 1, this means \( F \leq 1 \). So it is a normalization of the deviation from the counter flow case, similar to the ideal and real gas models difference. Two temperature ratios must be calculated to use \( F \), \( P \) and \( R \):

\[
R = \frac{T_1 - T_2}{t_2 - t_1} = \frac{(\dot{m}c_p)_{\text{tube side}}}{(\dot{m}c_p)_{\text{shell side}}}
\]  

(5)

\[
P = \frac{t_2 - t_1}{T_1 - t_1}
\]  

(6)

Where the subscript 1 represents the inlet pipe and the 2 the outlet for both hot and cold air streams, \( T \) represents the shell side (or the outside air stream) and \( t \) represents the tube (or inside air stream) temperatures in equation (5) and equation (6). Relating those expressions the correction factor \( F \) can be found using any correlation factor graphic obtained easily in [16, 17], and [15].

2.5. External Forced Convection

For the case of a square of edge length \( D \) duct with forced flow under external convection and when the work fluid is a gas, in our case air, the Nusselt number is:

\[
Nu = 0.094Re^{0.675}Pr^{1/3}
\]  

(7)

For equation (7), its validity is conditioned of the case when \( 3,900 < Re < 79,000 \). \( Re \) is the Reynolds number, which is a ratio of the inertial forces of the fluid over the viscous forces and \( Pr \) is the Prandtl number, which is a ratio of the dependence of the molecular diffusivity of momentum or kinematic viscosity \( (\nu = \mu/\rho) \) over the molecular diffusivity of heat \( (\alpha = k/\rho c_p) \). The diameter used for is the hydraulic diameter for a rectangular shape.

2.6. Internal Forced Convection

To obtain the Nusselt number for the case of turbulent flow under internal forced convection is a modification of the second Petukhov equation improved for better accurate calculations and also is able
to model the behavior for lower Re values. The range of application of the modified Petukhov equation is $3,000 < Re < 5,000,000$ and $0.5 \leq Pr \leq 2,000$.

\[
N_u = \frac{(f/8)(Re - 1000)Pr}{1 + 12.7(f/8)^{0.5}(Pr^{2/3} - 1)}
\]

This are all that is needed to obtain all the possible cases of the convection of air concerning the air to air intercooler. It should be emphasized that so far we have only calculated the convection of air, but we also need to use the water evaporation over the external surface of the heat exchanger in the next section. Again in equation (8), Re is the Reynolds number, Pr is the Prandtl number and f is the friction factor obtained by the usual Blasius equation available in any piping textbook.

2.7. Evaporative Heat Transfer and Energy taken from the Water Mass Transfer

Once the model of the dry surface intercooler is set up with the well known boundary layer theory, let us consider the case at which the channel diameter transporting compressed water over a “flat surface” may be small enough in dimension to not let the macroscopic boundary layer theory to be valid due to the size, because the boundary layer will be of the same order of magnitude as the hydraulic diameter of the channel.

As the channel get smaller in dimension, the mass diffusion forces will gain importance and may overcome the boundary layer forces. This means that a ratio of the momentum (and convection) diffusivity of the mass transfer and the diffusion of the mass must be quantified using the Péclet number [16]

\[
P_e = \frac{vL_c}{D_{\text{water-air}}}
\]

where $v$ is the velocity of flow, $L_c$ the characteristic length and $D_{\text{water-air}}$ is the diffusion coefficient called Marrero and Mason formula, widely spread over the literature.

The use of $Pe$ is justified and “based on the mean flow velocity through the narrowest opening between the tubes” [15], or in this case the narrowest part of the open channel carved over the flat surface of the heat exchanger, because of the periodic changes in the geometry allowing the passive water transport in the preferred direction.

For the dimensions of the channel with small values of length and of course $Pe$, there exists a dominant region where the effects are felt, and “In this region of small Péclet-numbers where diffusion and convection are of equal importance, few correlations are available for flat structures under laminar flow” [15].

In order to solve this, two cases of the Sherwood number are solved numerically and presented by Kachel et. al. resulting in one expression for the case of a macroscopic channel carved in a flat plate that is big enough, this means that is the region where the boundary layer is applicable, $Sh$ is for the case of laminar flow over a flat surface

\[
Sh = 0.664 \left( \frac{v}{D_{\text{water-air}}} \right)^{-1/6} Pe^{1/2}
\]

where $v$ is the dynamic viscosity and $D$ is the diffusion coefficient of water in air.

In the contrary case of turbulent flow over the flat surface and when the boundary layer theory is still valid, equation (10) must not be used and instead the equation (11) is needed to evaluate the value of $Sh$, for $Sc > 0.6$

\[
Sh = 0.037 Re_{L}^{0.8} Sc^{1/3}
\]
Where $Sc$ is the Schmidt number, the ratio of the kinematic viscosity and the diffusion of water vapor in air, from equation

$$Sc = \frac{v}{D_{\text{water-air}}}$$  \hspace{1cm} (12)

If the dimension of the channel is small enough in the order of magnitude of $10^{-5}$ m, that is the case of Kachel et. al. $Pe \rightarrow 0$ and then the macroscopic boundary layer theory is no longer applicable, because it neglects the fact that the diffusion forces are always present, but those forces are diminished by the fact that the convective forces are much stronger. Then the Sherwood number becomes numerically [16], for the case of laminar flow:

$$Sh = 0.8 \ P_e^{0.1}$$  \hspace{1cm} (13)

For the case of small Péclet number under turbulent conditions there are no expression available in the literature.

To find the coefficient of convection due to the mass transfer, $h_{\text{heat}}$ the mass convection $h_{\text{mass}}$ should be known first. From the definition of the Sherwood number which a measure of the convection heat mass transfer and the diffusive mass transfer.

$$h_{\text{heat}} = \rho c_p h_{\text{mass}} \left( \frac{\alpha}{D_{\text{water-air}}} \right)^{2/3}$$  \hspace{1cm} (14)

With this expression, and introducing it into equation (2) the model is completed for the case of the dry intercooler surface when the water transport and the evaporation are occurring simultaneously.

3. Final Design

The material selected of the final design is Al 6063-O, a heat treated alloy with good thermal conductivity and proper elasticity modulus suitable for this application. The final dimensions of the intercooler are 600 mm by 60 mm or all ducts and a height 15 mm with a separation of 15 mm with a total of 6 ducts. The thickness selected is 2 mm for all the walls. For the selected pressure boost of 2 bar absolute, the safety factor is 1.8, as can be seen in Figure 1 and Figure 2.

![Proposed design of the intercooler with the specified dimensions.](image-url)
Figure 2. Von Mises Stress pattern for the case of 2 mm thickness. Remember $E = 71.7$ [GPa].

Figure 3. Mimicked morphology from the Texas Horned Lizard. Dimensions in [mm] and actual look of the improved model (not calculated in this paper) but the morphology is periodically as well.

Results

With the model of the intercooler in mind, an external temperature of air of 20 °C and a pressure of 1 bar were selected, with a HR of 70%. For all the cases used, the air and the water vapor were treated as ideal gases because of the criteria stated above as the $Z$ factor in all cases was very close to the unity. A volumetric flow of 0.15m³/s was selected to be compressed by the turbo, and from equation (1) a discharge temperature of 109 °C was found, which is the temperature of the inlet of the intercooler. Then the mass flow rate is 0.283 kg/s and the volumetric flow is 0.0279 m³/s. Under these conditions, the viscosity and diffusivity are $1.93 \times 10^{-5}$ m²/s and $2.27 \times 10^{-5}$ m²/s respectively. For the internal convection then, from the dimensions given above and the air state, the velocity inside the duct is 7.55 m/s, for which corresponds a Re of 7215 which was practically treated as the turbulent flow was fully developed near the entrance. The Pr is 0.71 and the friction factor is found to be 0.034.

With all those numbers, the Nu is 22.728 per tube and we have six tubes. Finally using the Petukhov equation, the value of $h_{\text{in}}$ was found to be 230 W/m²·K.

For the case of external convection, the velocity of air was assumed to be 25 m/s. Then, the viscosity and diffusion are $1.52 \times 10^{-5}$ m²/s and $2.07 \times 10^{-5}$ m²/s. Therefore, Re is 39577 (turbulent) and Pr is 0.731. Nu was calculated with equation (7) and it was 107. Finally, $h_{\text{out}}$ is 114 W/m²·K.

After a few iterations, the model converged to an external surface temperature of the intercooler of 62 °C and a discharge temperature of 65 °C, dissipating 2625 W, for dry conditions.

Table 1. Temperatures for both air streams obtained after several iteration of the model ran into an Excel® spreadsheet with the model of the heat exchanger.

| Temperature (°C)       | $T_{\text{in}}$ | $T_{\text{out}}$ |
|------------------------|-----------------|-----------------|
| Hot air inside intercooler | 109             | 65              |
| Cold air outside intercooler | 20              | 24.2            |
For the case when water is being transported passively over the surface, $Re$ is now 48 265, $Le$ is 0.649, $Sh$ is 50 and $Pe$ is 1565 which means that the boundary layer theory is applicable. From the Marrero and Mason equation, the water diffusion on air is $3.193 \times 10^{-5}$ m²/s.

$Sc$ for those conditions is 0.0138, which is well below the range of applicability of equation (11). For that, an efficiency factor of evaporation was introduced which model how much of the water over the surface evaporates, and the rest is taken by the turbulent air in form of droplets, which requires further validation and simulation in the next stage of the project. An efficiency of 50% was used, and the lead to a mass convection of 0.054 m/s calculated from the definition of $Sh$. With this, the mass flow is 0.0033 kg/s of water over the surface and the convection coefficient of heat is 49.2 W/m²·K. This makes the surface temperature converge at 45.3 °C, dissipating almost 150 W extra. This obviously is well below the expected temperature drop to be around 10 °C, so results are taken with care and by no means are lead to a final decision.

**Table 2.** Final results from the calculation of the heat transfer and water evaporation model with the iteration stopped at $T_{h,\text{out}} = 65$ °C.

| Case                  | $\dot{Q}$ [kW] | $T_s$ [°C] |
|-----------------------|-----------------|------------|
| Convection only       | 2.62            | 62.05      |
| Water evaporation and convection | 0.15          | 45.31      |

**Conclusions**

According to the mass convection theory, for the heat lost there exist a drawback due to a deviation from real values due to the highly turbulent flow under the conditions selected and the very small value of the Schmidt number obtained in the literature and pooled to the lack of a proper expression to compute the Sherwood number under macroscopic turbulent flow, which was not found and it was modelled outside the Schmidt number range. This means that the model of water evaporation and mass transfer should be improved.

However, the model proposed was satisfactory, it needs further validation and extensive simulations due to the lack of literature concerning the evaporation of water over the lizard skin. Also this is only the model and the first stage of subsequent works.

The water evaporation allowed to increase the convection heat exchange by adding a convection of mass to the convection of heat. However, from results and real data the values obtained were inside the range expected.

Also, due to dynamic condition in the functioning of an engine, real data are to be collected from experiments in the next stage of the project.

**Summary**

Péclet and Schmidt numbers are given, which separate the boundary layer theory and the diffusion theory and establishes clear limits for both cases and when they overlap. Basically the boundary layer theory is not valid if Péclet number is smaller than unity. Then the Sherwood number is introduced to find the mass convection for turbulent and laminar flows, and is analogous to the Nusselt number for mass convection and diffusion.

The aluminum is chosen because of the mechanical properties such as weight and weight and the thermal properties, such as the conduction makes it ideal for this application. It is justified also why there is no calculation of the stress, because the projects focused on the heat transfer and not in the mechanical design, but a simulation was made with the optional wall thicknesses just to be sure everything is under safe functioning.

Then is found the Schmidt and Sherwood numbers and it is found that for our conditions of very small Schmidt number and highly turbulent flow, there exist no expression in the literature for the Sherwood number to find the exact mass convection, even though many scientific papers were consulted on the topic and not used in the references. Then a correction factor is used because from experience there can’t exist a very big value of temperature decrease and the evaporation efficiency is introduced.
as a normalized percentage of the water that actually evaporates and the rest is taken away from the channels by the fast and turbulent air.

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