Thermodynamics of indirect water injection in internal combustion engines: theoretical assessment of the fresh mixture cooling effect

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Water injection is a well-known efficient way to improve the performance of internal combustion engines. Amazingly, most of previous studies have yet only assess this process in an experimental manner, depriving us of an understanding of its specific influence on different operating phases of the engine — density of the aspirated fresh mixture, work required by the compression stroke, and so on — but also of the possibility to predict its effects if set up on an existing engine. Thanks to a theoretical framework specifically developed, and similar to the one commonly used for the analysis of air conditioning systems, we start in this paper to untangle in a theoretical manner the different consequences of water injection on internal combustion engines. This first study is specifically focused on the fresh mixture density increase, due to the vaporisation of liquid water in the intake manifold. Results show that, in the best scenarios, we cannot expect to increase the amount of fuel finally aspirated into the cylinders by more than 10%. The methodology presented here, as well as the python software specifically developed, can be of a precious help for the optimisation of such process if applied to existing or future engines.

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1. Introduction

1.1. Context

Internal Combustion Engine (ICE) is by far the most widespread of all prime movers, i.e. primary sources of mechanical power. According to several authors [1, 2], this situation is probably not about to change, at least in the short run. However, in order to keep up being so helpful to mankind, ICE has to overcome at least two formidable challenges.

The first one is related to the fuels such engines consume, predominantly of a fossil and then non sustainable nature¹. The second challenge ICE has to overcome is the control of its polluting emissions, including *green house gases*, *nitrogen oxides* and *fine particles*, whom daily consequences are already of a concrete and hazardous nature for a lot of us. ICE has thus to evolve in order to do its usual duty more cleanly and from a wider range of different fuels [2, 4, 5].

Because it helps to improve its performance and to decrease some of its polluting emissions, *water injection* (WI) is an old, well-known, but still promising solution that can help ICE in its quest for improvement [6, 7]. Most of previous studies of the effects of WI on ICE have been yet almost only based on experimental results, and are then partially ineffective to untangle its actual effects on different phases of the engine. For example, what is fresh mixture density increase we can expect from such process? By how much does it decreases the mechanical work consumed by the compression stroke? What are its consequences on the efficiency of the whole engine cycle? What is the optimal amount of water to inject in the intake manifold?

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¹In 2015 for example, about 92% of the fuels supplying the US transportation sector were petroleum by-products, according to the United States Energy Information Administration [3].

In order to start to answer these questions, we present in this paper a first theoretical analysis of the consequences of WI on the performance of engine cycles, and more specifically on the density increase of the aspirated fresh mixture.

After a presentation of the different hypotheses considered in the rest, and a short reminder of the good and bad points of WI, its thermodynamic analysis, based on mass and energy balances, will be detailed. An algebraic model specifically developed will be presented, which links the fresh mixture density at the aspiration point with the main parameters of both the intake system and the surroundings (pressure, temperature and so on). Considering some practical applications of this approach, we will show that the interest of such improvement process is dependent on the ambient temperature and humidity ; that the nature of the fuel, the Air-Fuel equivalent ratio λ value, and the presence of the fuel itself are almost no influential ; and that in the best situations, so with a very dry air and an intake manifold at relatively high temperature, we cannot expect to increase the amount of aspirated fuel by more than 10%.

The framework presented in this paper will be of a precious help to anyone interested in applying or optimising such improvement strategy on given or future engines.

1.2. Hypotheses

This paper is focused on the sole *fresh charge cooling effect* and on its consequences on the density of the fresh mixture aspirated by the engine. For so, we will consider the followed hypotheses:

- H1. The *indirect injection*, i.e. done in the engine intake manifold and not directly into the engine cylinders, is the only one considered in the present paper.
- H2. *Neat water* is the sole fluid to be injected, although injecting different blends, composed for instance of *hydrous ethanol* as presented in [8, 9], could be modelled with the very same approach.
- H3. Water is the *sole component to be vaporised* in the intake manifold. This hypothesis implies that the liquid fuels potentially consumed are already in a gaseous state when mixed with fresh air, and that we neglect their respective enthalpies of vaporisation, at least in front of the liquid water one.
- H4. Water is injected as a liquid *downstream of the fuel*, as represented in Figure 1, although the opposite situation sometimes occurs, as presented in references [10] or [11] for instance. This hypothesis is justified by the low proportion of fuel within the fresh mixture, that induces an insignificant effect of the presence of the fuel on the whole cooling process.
- H5. The whole water injection process is supposed as *adiabatic*.
- H6. The temperature of the injected liquid water is supposed to be the one of the inner intake manifold, at point 1 in Figure 1.
- H7. The *pressure drop* through the water injection process is supposed as negligible and we assume that $p_1 = p_2$ at points 1 and 2 in Figure 1.
- H8. All the gases considered in this study are supposed as *perfect*, i.e. whom behaviours can be described by the *ideal gas law*, and whom physical properties are supposed to be independent from temperature.

Figure 1: Internal Combustion Engine aspirating fresh air, so a blend of dry air and water vapour due to ambient humidity, from the surroundings at point 0. Fuel is supposed to be injected between points 0 and 1, so upstream of water which is supposed to be injected between points 1 and 2. In traditional compression ignited engines, the fuel is injected directly into the cylinder and only after the compression stroke. Humid air is thus the sole component of the fresh mixture mixed with water between points 1 and 2.

2. Water injection in internal combustion engines

2.1. Effects of water injection

When injected as liquid into an intake manifold, water evaporates and blends with the engine fresh mixture. We then talk about *indirect/port injection of water*, *inlet manifold water injection* or sometimes about *intake fumigation*. The *enthalpy of vaporisation* consumed by such phase change process is removed from the enthalpy of the whole intake flow and thus cools down the latter. The resulting temperature drop:

1. *increases the fresh mixture density* and so the amount of the latter actually aspirated within the engine *swept volume*. Consequently the mechanical power the engine delivers increases as well. This good point is the one considered in this paper.
2. leads to a *lower mechanical work required by the compression stroke*, the gas to compress being initially at a lower temperature, and some remaining liquid water being able to be vaporised during the compression in order to “internally” refrigerate the latter [12].
3. decreases the combustion *flame temperature*, the fresh mixture being at lower temperature at the end of the compression stroke and the heat of combustion being released in a larger amount of gas. The probability of *knock* phenomena is thus reduced, and the fresh mixture *octane number*² is increased [15, 16]. Higher *compression ratios* can then be reached.
4. increases the *amount of gas involved in the expansion stroke* and then the mechanical work provided by the latter.

Otherwise, considering the engine polluting emissions, water injection:

1. decreases the amount of *nitrogen oxides* (NO_x) finally rejected by the combustion, in decreasing the flame temperature [8, 10, 17–20]. The main source of NO_x is indeed the oxidation process of atmospheric nitrogen contained in the air composing the fresh mixture, due to a too high combustion peak temperature [21]. In doing so, water injection is an adversary of the exhaust gas recirculation (EGR) technology [22].

²Incidentally, water injection is also known as *Anti-Detonant Injection* (ADI), see [13, 14].

2. increases the production of unburnt hydrocarbons (UHC), purportedly because of an increase of the quenching layer of combustion, due to the cooling process of the cylinder inner surface, see [23, 24].
3. sometimes decreases or increases the production of *carbon monoxide* (CO), depending on the specific experimented configuration, as explained in [25].

2.2. Practical uses of water injection

The most famous application of such performance improvement strategy is probably the one concerning some supercharged World War II aircraft engines [26], although it has been later applied for the very same reason by Renault on some 1980s F1 supercharged engines [27]. For the sole purpose of NO_x control emissions, water can also be injected directly into the engine cylinder and vaporised, absorbing a fraction of the chemical energy hence released and then decreasing the peak temperature [7, 28, 29]. The latter process has been successfully applied to industrial engines, for example to some *marine Diesel engines* manufactured by Wärtsilä [30, page 73]. More recently, BMW reintroduced the use of water injection on its M4 GTS model, with an announced will to spread this process on other commercialised vehicles in the future [31].

3. Thermodynamics of humid fresh mixtures

When injected as liquid or spray into an engine intake manifold, water, in vaporising, decreases the temperature of the fresh mixture. Incidentally, the very same cooling effect is sometimes used in heating, ventilation and air conditioning (HVAC) systems, in order to humidify and cool down the air flow rate injected into buildings [32]. For such a reason, a specific formalism has been developed in order to treat the humidification process of intake fresh mixtures, similar to the one used in *psychrometrics*, the field of engineering concerned with humid air related processes.

3.1. Mass balance of the water injection process

3.1.1. Fresh mixture flow rate

While for HVAC systems, the gas stream to cool down is only composed of dry air and water, for internal combustion engines (ICE) we often have to add the fuel, as for spark ignited or dual fuel compression ignited engines [33]. After fuel injection, so at point 1 or 2 in Figure 1, the whole humid gas mass flow rate under concern can then be expressed as the sum:

$$\dot{m}_i = \dot{m}_{\text{fuel}} + \dot{m}_{\text{air}} + \dot{m}_{w,i} = \dot{m}_{\text{dfm}} + \dot{m}_{w,i} \quad (1)$$

With $i = \{1, 2\}$ and the subscripts “air”, “w” and “dfm” meaning dry air, water and dry fresh mixture (so air and fuel), respectively. It is worth noting that the existence of the water flow rate $\dot{m}_w$ is due for a part to the water injection process, but also to the surrounding air humidity, corresponding to point 0 in Figure 1. Equation (1) must now be improved by introducing several important and useful dimensionless parameters.

3.1.2. Air-Fuel Ratio

The *Air-Fuel Ratio*, henceforth noted AFR, is defined as the ratio of the combustive dry air mass flow rate consumed by the engine, over the fuel one:

$$\text{AFR} = \frac{\dot{m}_{\text{air}}}{\dot{m}_{\text{fuel}}} \Rightarrow \dot{m}_{\text{air}} = \dot{m}_{\text{fuel}} \cdot \text{AFR} \quad (2)$$

From definition (2), we can then express the dry fresh mixture flow rate $\dot{m}_{\text{dfm}}$ of Equation (1) as:

$$\dot{m}_{\text{dfm}} = \varphi \cdot \dot{m}_{\text{fuel}} + \dot{m}_{\text{air}} = \dot{m}_{\text{fuel}} \cdot (\varphi + \text{AFR}) \quad (3)$$

φ is a parameter equal to unity if the fuel is actually mixed with air in the fresh mixture before the intake valve, or equal to zero otherwise:

- Fuel is indeed injected into “traditional” compression ignited (CI) engines directly into the cylinder and only near the end of the compression stroke [21]. In Equation (3), it means $\varphi = 0$.
- On the contrary, in spark ignited (SI) or dual fuel CI engines, fuel is mixed with air during the intake process and we have $\varphi = 1$ in Equation (3).

The AFR value depends on the fuel chemical composition and of the effective *Air-Fuel equivalence Ratio*, often noted λ and defined as:

$$\lambda = \frac{\text{AFR}}{\text{AFR}_{\text{st}}} \quad (4)$$

with subscript “st” that means *stoichiometric*. Thus, $\lambda = 1$ for a stoichiometric combustion, while $\lambda > 1$ for a lean combustion, i.e. with excess air, and $\lambda < 1$ for a rich combustion [21].

3.1.3. Specific water content

The parameter sometimes called *specific humidity*, *moisture content*, *humidity ratio* or *mixing ratio*, noted ω , is traditionally defined, for a given amount of humid air, as the ratio of the water vapour mass, over the dry air one [32]. For ventilating systems, this way to quantify the humidity of air is usually preferred to a more common mass fraction. Indeed, during an humidification or drying process, whereas the amount of water vapour actually contained in a given amount of humid gas may vary, the corresponding amount of the sole dry gas remains the same. Using ω instead of a mass fraction helps then to simplify the resulting calculations. The dry gas under concern in this study being the dry fresh mixture of Equation (3), we have at point i :

$$\omega_i = \frac{\dot{m}_{w,i}}{\dot{m}_{\text{dfm}}} = \frac{\dot{m}_{w,i}}{\dot{m}_{\text{fuel}} \cdot (\varphi + \text{AFR})} \quad (5)$$

A further difference with usual HVAC systems is the potential presence of liquid water mixed, as *fog* for instance, with the fresh mixture stream. As we shall explain in the rest, it is possible to inject more water than the maximal amount which is possible to evaporate. As long as no “cold spot” exists within the intake system, which can imply some premature liquefaction phenomena, the water droplets can be transported until the inside of the cylinder. The latter process can be beneficial for the engine compression stroke for instance, because of the cooling effect of water droplets evaporation inside the swept volume, as it occurs with direct injection of water presented in references [13, 29, 34–36]. As a consequence and for clarity purposes, ω will be henceforth called *specific water content*.

3.1.4. Water-Fuel Ratio

The *Water-Fuel Ratio*, noted WFR in the rest, is the fundamental parameter of water injection systems. It is defined as the ratio of the injected liquid water mass flow rate $\dot{m}_{\text{inj}}$, over the fuel one $\dot{m}_{\text{fuel}}$ (see Figure 1):

$$\text{WFR} = \frac{\dot{m}_{\text{inj}}}{\dot{m}_{\text{fuel}}} \Rightarrow \dot{m}_{\text{inj}} = \dot{m}_{\text{fuel}} \cdot \text{WFR} \quad (6)$$

Depending on the concerned application, the Water-Fuel Ratio can be typically such as $0.2 \leq \text{WFR} \leq 1.5$, as presented for instance in references [6, 24, 28, 29, 37]. Anyway, its value has

Figure 2: Simplified representation of the fresh mixture composition, with points 0 to 2 corresponding to the ones of Figure 1.

been so far chosen from experimental approaches only, usually in order to optimise the power finally produced by the engine. We will use it in the rest for the same purpose, but in a quite different way.

Using AFR and ω , respectively defined by Equations (2) and (5), the fresh mixture mass flow rate at point i , previously expressed by Equation (1), can now be written as the product:

$$\dot{m}_i = \dot{m}_{\text{fuel}} \cdot (\varphi + \text{AFR}) \cdot (1 + \omega_i) \quad (7)$$

The difference between the mass flow rate of Equation (7), when considered at point 2 and 1, being equal to the liquid water injected one $\dot{m}_{\text{inj}}$ defined by (6) (see in Figure 1), we obtain a new expression of the Water-Fuel Ratio:

$$\text{WFR} = (\varphi + \text{AFR}) \cdot (\omega_2 - \omega_1) \quad (8)$$

With the specific water content at point 1 calculated from the surrounding one at point 0 according to:

$$\omega_1 = \frac{\text{AFR}}{\varphi + \text{AFR}} \cdot \omega_0 \quad (9)$$

Fuel being possibly mixed with fresh air between point 0 and 1 in Figure 1, the dry gas flow rate thanks to whom specific water content ω is defined has changed in the meanwhile, so the value $\varphi = 1$ in Equation (9). If not, we have $\omega_1 = \omega_0$ for $\varphi = 0$, as represented for instance in Figure 2. Practically, most hydrocarbon fuels have and $\text{AFR} \simeq 15$ and so $\omega_1/\omega_0 = 15/16 \simeq 0.94$.

3.1.5. Gaseous and liquid phases of water

The cooling effect of the fresh mixture being created by the vaporisation of the injected liquid water, at first glance, the larger the amount of the latter, the larger the resulting mixture temperature drop. The amount of liquid water which can be actually vaporised is yet limited by

the *equilibrium state* between liquid water and vapour, often clumsily called “saturated” state of air³.

Indeed, the water vapour concentration within the fresh mixture will never exceed the one represented by the *equilibrium vapour pressure*, noted p_{eq} in the rest, so the partial pressure of water vapour when in equilibrium with its liquid phase. Practically, p_{eq} depends on relative temperature θ only, and can be calculated thanks to experimental correlations as for example the Hyland and Wexler [39] one. The *molar fraction* of water vapour into the fresh mixture being equal to its *partial pressure*, according to the Dalton’s law, we always have:

$$\frac{n_{\text{vap}}}{n} \leq \frac{p_{\text{eq}}(\theta)}{p} \quad (10)$$

with n the molar amount of matter, in mol, p the “total” pressure of the fresh mixture in the intake manifold and the subscript “vap” meaning water vapour. If combined with the definition of Equation (5) of the specific vapour content ω , Equation (10) leads to a maximum value of the same parameter, considered for example at point 2, such as:

$$\omega_{\text{vap,eq}}(p_2, \theta_2) = \alpha_w \cdot \underbrace{\frac{\varphi + \alpha_{\text{fuel}} \cdot \text{AFR}}{\alpha_{\text{fuel}} \cdot (\varphi + \text{AFR})}}_{\textcircled{1}} \cdot \frac{p_{\text{eq}}(\theta_2)}{p_2 - p_{\text{eq}}(\theta_2)} \quad (11)$$

With p_2 and θ_2 the fresh mixture pressure and temperature at point 2, respectively, and the constants α ’s defined as:

$$\alpha_w = \frac{M_w}{M_{\text{air}}} \simeq 0.621\,98 \quad \alpha_{\text{fuel}} = \frac{M_{\text{fuel}}}{M_{\text{air}}} \quad (12)$$

M is the molar mass of each component. It is important to note that θ_2 is here the fresh mixture temperature at the intake valve point, so **after** the cooling effect due to water injection. Thus, it will be the result of a further calculation explained in the rest. Once again, without any fuel mixed with dry air, so with $\varphi = 0$, fraction $\textcircled{1} = 1$ and Equation (11) is the one usually used for air conditioning systems [32].

3.1.6. Equilibrium Water-Fuel Ratio

To the maximal amount of water vapour contained in the fresh mixture after the water injection process, represented by $\omega_{\text{vap,eq}}$ of Equation (11), corresponds a specific value of the Water-Fuel Ratio noted WFR_{eq} . The latter is by definition the largest value of the WFR that ensure a complete vaporisation of the injected liquid mass flow rate $\dot{m}_{\text{inj}}$. On the contrary, if $\text{WFR} > \text{WFR}_{\text{eq}}$, a fraction of $\dot{m}_{\text{inj}}$ cannot be vaporised and is blend as liquid with the fresh mixture. According to (8) and (11), we have then:

$$\text{WFR}_{\text{eq}} = (\varphi + \text{AFR}) \cdot (\omega_{\text{vap,eq}} - \omega_1) \quad (13)$$

In the rest of this paper, we will suppose that a liquid fraction exists at the intake point only if $\text{WFR} > \text{WFR}_{\text{eq}}$. Such hypothesis means that *any amount of liquid which can be evaporated, actually is*. Furthermore, as previously mentioned, the intake valve temperature θ_2 , and so the equilibrium Water-Fuel Ratio WFR_{eq} , are the results of another calculation presented in the rest. In practical applications, it is important to keep in mind that pressure p_2 , and thus temperature θ_2 , can widely vary regarding to the possible use of ancillary systems not drawn on Figure 1, as e.g. a *supercharger*, a *turbocharger*, an *intercooler*, and so on.

³As clearly reminded in reference [38, chapter 5] for instance, this so-called “saturation” state of air has actually nothing to do with air itself, or with any other dry gas humidified by water. The use of the word “saturation” in such context is then clearly misleading regarding to the actual physical processes involved. This is why we will avoid it in the rest.

3.1.7. Intake valve fresh mixture composition

From the combination of Equation (7) considered at point 2 and of Equations (8) and (9), it is possible to express the whole fresh mixture mass flow rate at the same point as:

$$\dot{m}_2 = \dot{m}_{\text{fuel}} \cdot (\varphi + \text{AFR} \cdot (1 + \omega_0) + \text{WFR}) \quad (14)$$

And the respective fractions of fuel, air and water can be expressed at the same location in dividing the corresponding mass flow rates by the one of Equation (14), so for fuel:

$$x_{\text{fuel},2} = \frac{\dot{m}_{\text{fuel}}}{\dot{m}_2} = \frac{\varphi}{\varphi + \text{AFR} \cdot (1 + \omega_0) + \text{WFR}} \quad (15)$$

For air:

$$x_{\text{air},2} = \frac{\dot{m}_{\text{air}}}{\dot{m}_2} = \frac{\text{AFR}}{\varphi + \text{AFR} \cdot (1 + \omega_0) + \text{WFR}} \quad (16)$$

And for water in both phases:

$$x_{w,2} = \frac{\dot{m}_{w,2}}{\dot{m}_2} = \frac{\text{AFR} \cdot \omega_0 + \text{WFR}}{\varphi + \text{AFR} \cdot (1 + \omega_0) + \text{WFR}} \quad (17)$$

It is easy to check that the sum of Equations (15), (16) and (17) is indeed always equal to unity. The distinction between the water vapour fraction and the liquid water one is made thanks to the equilibrium value of the Water-Fuel Ratio, as defined by Equation (13). Hence, while the water vapour fraction is the one of Equation (17) if $\text{WFR} \leq \text{WFR}_{\text{eq}}$, we have in the opposite situation (when $\text{WFR} > \text{WFR}_{\text{eq}}$):

$$x_{\text{vap},2} = \frac{\dot{m}_{\text{vap},2}}{\dot{m}_2} = \frac{\text{AFR} \cdot \omega_0 + \text{WFR}_{\text{eq}}}{\varphi + \text{AFR} \cdot (1 + \omega_0) + \text{WFR}} \quad (18)$$

and:

$$x_{\text{liq},2} = \frac{\dot{m}_{\text{liq},2}}{\dot{m}_2} = \frac{\text{WFR} - \text{WFR}_{\text{eq}}}{\varphi + \text{AFR} \cdot (1 + \omega_0) + \text{WFR}} \quad (19)$$

Thus, we always have $x_{w,2} = x_{\text{vap},2} + x_{\text{liq},2}$.

3.2. Energy balance of the cooling process

As previously explained, the liquid water injection process induces the fresh mixture cooling thanks to an exchange of the whole gaseous mixture enthalpy for the liquid water enthalpy of vaporisation. It is worth noting that the very same phenomenon also exists for other injected liquids, as e.g. the liquid fuels commonly consumed by such engines, but with a weaker cooling effect due to a comparatively greater value of the water enthalpy of vaporisation ($\geq 6 \times$ the *octane* and *cetane* ones and $\geq 4 \times$ the *ethanol*'s one, see in Table 1 page 12), so the hypothesis H3 of page 3.

Anyhow, the energy balance of such an *open process* can be established in considering the *enthalpy rate* of each flow under concern. The one of the humid fresh mixture can be expressed at point i using AFR and ω of Equations (2) and (5), as:

$$\begin{aligned} (\dot{m} \cdot h)_i &= \varphi \cdot \dot{m}_{\text{fuel}} \cdot h_{\text{fuel},i} + \dot{m}_{\text{air}} \cdot h_{\text{air},i} + \dot{m}_{w,i} \cdot h_{w,i} \\ &= \dot{m}_{\text{fuel}} \cdot (\varphi \cdot h_{\text{fuel},i} + \text{AFR} \cdot h_{\text{air},i} + (\varphi + \text{AFR}) \cdot \omega_i \cdot h_{w,i}) \end{aligned} \quad (20)$$

The h 's are the usual specific enthalpies of fuel, air and water, respectively, both expressed in kJ/kg, and depending on the local temperature (so the subscripts " i "). As for water content of Equation (5) for example, it is common when studying heat, ventilation and air conditioning (HVAC) systems, to divide all *extensive parameters* by the constant flow rate, here the one of

air and fuel, so $\dot{m}_{\text{dfm}}$ of Equation (3). In doing so to the enthalpy rate of Equation (20), we obtain a first expression of the specific enthalpy of the humid fresh mixture at point i :

$$\underline{h}_i = \frac{(\dot{m} \cdot h)_i}{\dot{m}_{\text{dfm}}} = \frac{\varphi \cdot h_{\text{fuel},i} + \text{AFR} \cdot h_{\text{air},i}}{\varphi + \text{AFR}} + \omega_i \cdot h_{w,i} = \underline{h}_{\text{dfm},i} + \omega_i \cdot h_{w,i} \quad (21)$$

Once again, this expression is the one used for HVAC applications, so with dry air only, as long as no fuel is involved and then as $\varphi = 0$.

Considering now the energy balance of the water injection system of Figure 1, supposed as adiabatic according to hypothesis H5, we have, in using the specific enthalpy of Equation (21):

$$\dot{m}_{\text{dfm}} \cdot (\underline{h}_{\text{dfm},1} + \omega_1 \cdot h_{w,1}) + \dot{m}_{\text{inj}} \cdot h_{\text{inj}} = \dot{m}_{\text{dfm}} \cdot (\underline{h}_{\text{dfm},2} + \omega_2 \cdot h_{w,2}) \quad (22)$$

According to previous assumptions, the water blended with fresh mixture can be composed at intake point 2 of vapour only, or on liquid and vapour, depending on the value of the actual Water-Fuel Ratio. As long as $\text{WFR} \leq \text{WFR}_{\text{eq}}$, the concerned water is thus only in a vapour phase at intake valve point 2, whereas liquid and vapour can be both presents otherwise.

3.3. Complete vaporisation

In the first situation, so if $\text{WFR} \leq \text{WFR}_{\text{eq}}$, water is only present within the fresh mixture as a vapour, and the specific enthalpy of water is the one of its vapour phase only. At intake valve point 2, we have:

$$\omega_2 \cdot h_{w,2} = \omega_2 \cdot h_{\text{vap},2} \quad (23)$$

h_{vap} is here the specific enthalpy of water vapour alone. We also have, according to Equations (6) and (8), the specific enthalpy of the injected liquid water:

$$\dot{m}_{\text{inj}} \cdot h_{\text{inj}} = \dot{m}_{\text{fuel}} \cdot \text{WFR} \cdot (-\ell_{w,1}) \quad (24)$$

$\ell_{w,1}$ is the specific *enthalpy of vaporisation of water* at temperature θ_1 — according to hypothesis H6 — which corresponds to the amount of heat required to vaporise, at constant pressure, one unit of mass of liquid water taken at this temperature. Equation (24) also includes a minus sign “−” because of the fundamental endothermic nature of vaporisation processes.

When applied to this water injection process with a complete vaporisation, the energy balance of Equation (22), combined with Equations (23) and (24) leads to the follow equality:

$$\underline{h}_{\text{dfm},1} + \omega_1 \cdot (h_{\text{vap},1} + \ell_{w,1}) = \underline{h}_{\text{dfm},2} + \omega_2 \cdot (h_{\text{vap},2} + \ell_{w,2}) \quad (25)$$

and so to an equality between two specific enthalpies of humid fresh mixture:

$$\underline{h}_1 = \underline{h}_2 \Leftrightarrow \underline{h}_i = \underline{h}_{\text{dfm},i} + \omega_i \cdot (h_{\text{vap},i} + \ell_{w,i}) \quad (26)$$

It is quite usual now: definition (26) is exactly the one used in ventilation systems if the dry fresh mixture is composed by the sole dry air [32] and so if $\underline{h}_{\text{dfm}} = h_{\text{air}}$ and $\varphi = 0$. Moreover, as it is usual to in psychometrics, we will neglect the variation of the enthalpy of vaporisation between temperatures at point 1 and at point 2. The variation of ℓ_w being for instance lower than 8% between 0°C and 80°C.

3.4. Incomplete vaporisation

If the injected water flow rate $\dot{m}_{\text{inj}}$ is too large to be completely vaporised, i.e. if $\text{WFR} > \text{WFR}_{\text{eq}}$, liquid and vapour phases are both present after the water injection point, so at the intake valve point 2 in Figure 1. The corresponding enthalpy of water at the very same point is then:

$$\omega_2 \cdot h_{w,2} = \underbrace{\omega_{\text{vap,eq},2} \cdot h_{\text{vap},2}}_{\textcircled{1}} + \underbrace{(\omega_2 - \omega_{\text{vap,eq},2}) \cdot h_{\text{liq},2}}_{\textcircled{2}} \quad (27)$$

① corresponds to the fraction of the liquid water injected which is completely vaporised and ② is the one remaining as liquid. The enthalpy rate of the liquid water injected may also vary regarding to WFR and we have, according to (3), (8) and (13):

$$\dot{m}_{\text{inj}} \cdot h_{\text{inj}} = \underbrace{\dot{m}_{\text{fuel}} \cdot \text{WFR}_{\text{eq}} \cdot (-\ell_{w,1})}_{\text{①}} + \underbrace{\dot{m}_{\text{fuel}} \cdot (\text{WFR} - \text{WFR}_{\text{eq}}) \cdot h_{\text{liq},1}}_{\text{②}} \quad (28)$$

With ① and ② which have the same physical meaning as in Equation (27). When applied to a water injection process with $\text{WFR} > \text{WFR}_{\text{eq}}$, the same energy balance, based on the combination of Equations (22), (27) and (28), leads to:

$$\underline{h}_1 - \underline{h}_2 = (\omega_2 - \omega_{\text{vap,eq}}) \cdot (h_{\text{liq},2} - h_{\text{liq},1}) \quad (29)$$

The difference between the humid fresh mixture specific enthalpy when considered at point 1 and 2 is now equal to the enthalpy absorbed by the non evaporated water. Hence, to this non vaporised water corresponds a further specific enthalpy, which increases the one of the whole stream cooled down by the vaporisation effect. We shall see in the rest that such phenomenon leads to a smaller temperature drop of the fresh mixture.

It is important to note that Equation (29) does not mean that the enthalpy rate of the whole process is not conserved: the injection process is still supposed as adiabatic (hypothesis H5). It means that the specific enthalpy of the humid fresh mixture, as defined by Equation (26) in order to be conserved through a complete evaporation process, is not conserved anymore when the injected flow rate cannot be entirely vaporised.

Using the conventional formalism of psychrometrics, we can express each specific enthalpy h thanks to a temperature difference — all gases are supposed to be perfect, according to hypothesis H8 — such as:

$$\Delta h = h - h_{\text{ref}} = \int_{T_{\text{ref}}}^T c_p(\vartheta) \cdot d\vartheta \simeq \bar{c}_p \cdot (T - T_{\text{ref}}) \quad (30)$$

with $\bar{c}_p$ the average specific heat at constant pressure within the considered temperature range $T_{\text{ref}} \leq \vartheta \leq T$, and h_{ref} the reference enthalpy value taken at reference temperature T_{ref} . Conventionally, in psychrometrics, we consider that $h_{\text{ref}} = 0 \text{ J/kg}$ at $T_{\text{ref}} = 0^\circ\text{C}$ and so for each component we have:

$$h = \bar{c}_p \cdot \theta \quad (31)$$

with θ the relative temperature. Combining Equations (26) and (31) with a constant water enthalpy of vaporisation, we obtain a new expression of the fresh mixture specific enthalpy at point i :

$$\underline{h}_i = \bar{c}_{p,\text{dfm}} \cdot \theta_i + \omega_i \cdot (\bar{c}_{p,\text{vap}} \cdot \theta_i + \ell_w) = (\bar{c}_{p,\text{dfm}} + \omega_i \cdot \bar{c}_{p,\text{vap}}) \cdot \theta_i + \omega_i \cdot \ell_w \quad (32)$$

With the specific heat at constant pressure of the dry fresh mixture:

$$\bar{c}_{p,\text{dfm}} = \frac{\varphi \cdot \bar{c}_{p,\text{fuel}} + \text{AFR} \cdot \bar{c}_{p,\text{air}}}{\varphi + \text{AFR}} \quad (33)$$

In psychrometrics, the values of $\ell_w = 2501 \text{ kJ/kg}$ and $\bar{c}_{p,\text{liq}} \simeq 4.18 \text{ kJ/(kg} \cdot \text{K)}$ are conventionally considered at a reference temperature of 0°C , as reminded in [32]. The other physical properties considered in this paper are presented in Table 1. We will use the same convention in the rest, although the temperature ranges considered here are slightly more away from this reference temperature of 0°C than the typical ones for air conditioning systems.

3.5. Cooling effect and wet-bulb temperature

If we consider the water injection with a complete vaporisation, so with $\text{WFR} \leq \text{WFR}_{\text{eq}}$, we have, using Equations (26) and (32), an intake temperature drop $\theta_2 - \theta_1$ such as:

$$\theta_2 - \theta_1 = -\frac{\bar{c}_{p,\text{vap}} \cdot \theta_1 + \ell_w}{\bar{c}_{p,\text{dfm}} + \omega_2 \cdot \bar{c}_{p,\text{vap}}} \cdot (\omega_2 - \omega_1) \quad (34)$$

We can then express the specific water content ω as functions of the Water-Fuel Ratio, according to Equation (8), in order to obtain:

$$\theta_2 - \theta_1 = -\frac{(\bar{c}_{p,\text{vap}} \cdot \theta_1 + \ell_w) \cdot \text{WFR}}{(\varphi + \text{AFR}) \cdot (\bar{c}_{p,\text{dfm}} + \omega_1 \cdot \bar{c}_{p,\text{vap}}) + \bar{c}_{p,\text{vap}} \cdot \text{WFR}} \quad (35)$$

However, because we usually have $\ell_w \gg \bar{c}_{p,\text{vap}} \cdot \theta_1$ and $(\varphi + \text{AFR}) \cdot (\bar{c}_{p,\text{dfm}} + \omega_1 \cdot \bar{c}_{p,\text{vap}}) \gg \bar{c}_{p,\text{vap}} \cdot \text{WFR}$, the temperature difference $\theta_2 - \theta_1$ corresponding to the researched cooling effect can be expressed in an approximative but linear form regarding to Water-Fuel Ratio, as:

$$\theta_2 - \theta_1 \simeq -\frac{\ell_w \cdot \text{WFR}}{(\varphi + \text{AFR}) \cdot (\bar{c}_{p,\text{dfm}} + \omega_1 \cdot \bar{c}_{p,\text{vap}})} \quad (36)$$

Hence, as long as the equilibrium state between liquid water and vapour is not reached, the greater the WFR, the greater the cooling effect of the fresh mixture.

In the psychrometric or Carrier diagram (θ, ω) in Figure 3, when the vaporisation is complete and so when $\underline{h}_2 = \underline{h}_1$, the point (θ_2, ω_2) corresponding to intake valve point 2 is moving upward along the straight line representing the constant value of specific enthalpy $\underline{h}_2 = \underline{h}_1$. Thus, the maximum possible temperature drop is attained when the difference of specific water content is maximum as well, so when the line of $\underline{h}_2$ reach the curve corresponding to the equilibrium vapour pressure, with $\omega_2 = \omega_{\text{vap,eq}}$. In this case, the lowest intake temperature reached is by definition the *wet-bulb temperature*, noted θ_{wb} [32]. The value of θ_{wb} can be practically obtained in solving Equation (37), obtained from $\underline{h}_2 = \underline{h}_1$, and from Equations (11) and (32):

$$\bar{c}_{p,\text{dfm}} \cdot \theta_{\text{wb}} + \omega_{\text{vap,eq}}(p_2, \theta_{\text{wb}}) \cdot (\bar{c}_{p,\text{vap}} \cdot \theta_{\text{wb}} + \ell_w) = \underline{h}_1 \quad (37)$$

Whereas for dry air only, some correlations have been developed to compute directly θ_{wb} for θ_1 and ω_1 , Equation (37) is here practically solved thanks to a specific numerical Newton's algorithm, because of the potential presence of fuel in the fresh mixture with $\bar{c}_{p,\text{dfm}} \neq \bar{c}_{p,\text{air}}$; and because of pressure p_2 which can be clearly different from the atmospheric one.

In this way, the wet-bulb temperature is the lowest one of the intake valve point 2, because it corresponds to the maximum amount of vaporised liquid water. Thus, this temperature is

Table 1: Main physical properties of the chemical components considered as gases in this study. Datas extract from [32] and considered at $T = 273 \text{ K}$ for dry air and water vapour; and from the *National Institute of Standards and Technology* [40] at $T = 298 \text{ K}$ for the other components.

Component	M [g/mol]	r [J/(kg · K)]	c_p [J/(kg · K)]	ℓ [kJ/kg]
Dry air	28.964 5	287.055	1 006	—
Steam	18.015 3	461.520	1 805	2 501
Ethanol	46.068 4	180.480	1 415	547.89
Octane	114.232	72.785 3	1 644	363.25
Cetane	226.448	36.716 8	2 210	359.02

Figure 3: Psychrometric or Carrier diagram (θ, ω) commonly used for air conditioning applications. From the point 1 corresponding to the fresh mixture before the water injection process, the specific water content ω_1 increases until ω_2 and decreases the temperature from θ_1 to θ_2 . For a completely vaporised water injection, any attainable point 2 is on the straight line representing $\underline{h}_2 = \underline{h}_1$. The greatest cooling effect is thus obtained for $\omega_2 = \omega_{\text{vap,eq}}$. The corresponding temperature is by definition the *wet-bulb* one noted θ_{wb} . For incomplete vaporised water, once the wet-bulb temperature is reached, the intake temperature will increase until $\theta_{2'}$, because of the remaining liquid water specific enthalpy.

actually the one involved in the definition of the equilibrium Water-Fuel Ratio WFR_{eq} defined in Equation (13), and we have instead:

$$\text{WFR}_{\text{eq}} = (\varphi + \text{AFR}) \cdot (\omega_{\text{vap,eq}}(p_2, \theta_{\text{wb}}) - \omega_1) \quad (38)$$

The wet-bulb temperature will be of a great importance in the rest because it will be used to calculate the value of the equilibrium Water-Fuel Ratio WFR_{eq} according to (38), and then to demarcate the situations corresponding to water injection processes with complete vaporisations, so with $\text{WFR} \leq \text{WFR}_{\text{eq}}$ from the incompletely vaporised ones corresponding to $\text{WFR} > \text{WFR}_{\text{eq}}$.

According to energy balance of Equation (29), the specific enthalpy $\underline{h}$ of the fresh mixture is not conserved anymore when the injected water is not completely vaporised. On the contrary, we have, using Equation (31) in the expressions of the liquid water specific enthalpy, the difference:

$$\underline{h}_1 - \underline{h}_2 = (\omega_2 - \omega_{\text{vap,eq}}) \cdot \bar{c}_{p,\text{liq}} \cdot (\theta_2 - \theta_1) \leq 0 \quad (39)$$

Because $\theta_1 \geq \theta_2$ and $\omega_2 \geq \omega_{\text{vap,eq}}$, we have $\underline{h}_1 \leq \underline{h}_2$. When combined with Equation (32), Equation (39) leads to another expression of the temperature difference:

$$\theta_2 - \theta_1 = - \frac{\bar{c}_{p,\text{vap}} \cdot \theta_1 + \ell_w}{\bar{c}_{p,\text{dfm}} + (\omega_2 - \omega_{\text{vap,eq}}) \cdot \bar{c}_{p,\text{liq}} + \omega_{\text{vap,eq}} \cdot \bar{c}_{p,\text{vap}}} \cdot (\omega_{\text{vap,eq}} - \omega_1) \quad (40)$$

The difference of Equation (39) is represented by the line noted $\underline{h}_{2'} > \underline{h}_1$ in Figure 3. It is then clearly visible on the same diagram that injecting more water than the amount possibly vaporised, so with $\text{WFR} > \text{WFR}_{\text{eq}}$, leads to a smaller temperature drop such as $|\theta_{2'} - \theta_1| < |\theta_{\text{wb}} - \theta_1|$.

It is worth noting that Equation (40) must be understood in a slightly different way than previous Equation (34). Indeed, as long as WFR and then ω_2 increase, so the difference $\omega_2 - \omega_{\text{vap,eq}}$, that leads to a decrease of the temperature drop $|\theta_2 - \theta_1|$. At the same time yet, the intake temperature θ_2 thanks to whom the equilibrium specific water content $\omega_{\text{vap,eq}}$ is calculated varies as well, inducing a more complicated evolution of the temperature difference $\theta_2 - \theta_1 = f(\text{WFR})$. Anyway, the maximum drop always corresponds to a Water-Fuel Ratio equal to the equilibrium one:

$$\text{WFR}_{\text{eq}} = \arg \max_{\text{WFR}} (|\theta_2 - \theta_1|) \quad (41)$$

Practically though, the intake temperature θ_2 can not be calculated “directly” in solving Equation (40), but rather by solving Equation (42), similar to Equation (37) used to compute the wet-bulb temperature:

$$(\bar{c}_{p,\text{dfm}} + \omega_2 \cdot \bar{c}_{p,\text{liq}}) \cdot \theta_2 + \omega_{\text{vap,eq}}(p_2, \theta_2) \cdot ((\bar{c}_{p,\text{vap}} - \bar{c}_{p,\text{liq}}) \cdot \theta_2 + \bar{c}_{p,\text{liq}} \cdot \theta_1 + \ell_w) = \underline{h}_1 - \omega_2 \cdot \bar{c}_{p,\text{liq}} \cdot \theta_1 \quad (42)$$

The intake valve temperature θ_2 and all the other resulting parameters can now be calculated for any specific configuration of the water injection process.

3.6. Intake valve temperature

Let us suppose as known the “primary” parameters φ , AFR, as long as the specific heats $\bar{c}_p$ of both air and fuel. The actual value of the intake valve temperature θ_2 , and all the resulting parameters, can now be computed by the followed algorithm, also presented in Figure 4:

1. From the values of the temperature θ_1 and of the specific water content ω_1 before the point of water injection, we compute the corresponding specific enthalpy value $\underline{h}_1$, according to Equation (32).
2. From the values of the specific enthalpy $\underline{h}_1$ and of the intake pressure p_2 , we compute the wet-bulb temperature θ_{wb} , so the minimum fresh mixture temperature that can be reached, in solving Equation (37).
3. Using Equation (38), we compute the equilibrium Water-Fuel Ratio WFR_{eq} corresponding to θ_{wb} in Figure 3.
4. After a comparison between the actual Water-Fuel Ratio WFR and the one corresponding to a complete vaporisation of water WFR_{eq} , we compute the actual intake temperature θ_2 using Equation (34) if $\text{WFR} \leq \text{WFR}_{\text{eq}}$, or Equation (42) otherwise. The required value of ω_2 is obtained thanks to the actual WFR value used in Equation (8).

The same algorithm can also be used to optimise the fresh mixture temperature at the intake valve point, in ensuring at any time that $\text{WFR} = \text{WFR}_{\text{eq}}$.

3.7. Consequences on the amount of fresh mixture aspirated

It is now possible to calculate the most important parameter of this study, the *density* of the **gaseous** fresh mixture at the intake valve point. We emphasize here on the gaseous part of the mixture because it is the only one that will play the role of a working fluid in the next strokes of the engine cycle. Furthermore, even if some liquid water is actually aspirated within the engine, its specific volume is so small in front the gaseous one, that the effective swept volume of the

Figure 4: Graphical representation of the algorithm dedicated to the calculation of the fresh mixture temperature at the intake valve point θ_2 .

engine will not be affected. Thus, according to hypothesis H8 about the perfect nature of all the gases involved, the fresh mixture gaseous density at point 2 can be expressed as:

$$\rho_2 = \frac{m_2}{V_d} = \frac{p_2}{\underline{r} \cdot T_2} \quad (43)$$

$T_2 = \theta_2 + 273.15$ is the absolute fresh mixture temperature at the intake valve point, whom calculation process has just been detailed, and $\underline{r}$ is the fresh mixture specific constant given by:

$$\underline{r} = \bar{r}_{\text{dfm}} + \omega_2 \cdot r_w \Leftrightarrow \bar{r}_{\text{dfm}} = \frac{\varphi \cdot r_{\text{fuel}} + \text{AFR} \cdot r_{\text{air}}}{\varphi + \text{AFR}} \quad (44)$$

Density ρ_2 is the one of the whole fresh mixture, including water vapour. The power eventually produced by the engine being primarily related to the amount of fuel aspirated — the required AFR value being ensured by the fuel injection system — we will rather consider the “fuel density” inside the fresh mixture, noted $\rho_{\text{fuel},2}$ and defined by:

$$\rho_{\text{fuel},2} = \frac{m_{\text{fuel},2}}{V_d} = x_{\text{fuel},2} \cdot \rho_2 = \frac{x_{\text{fuel},2} \cdot p_2}{\underline{r} \cdot T_2} \quad (45)$$

With the mass fraction of fuel $x_{\text{fuel},2}$, from Equation (15), given by:

$$x_{\text{fuel},2} = \frac{1}{\varphi + \text{AFR} \cdot (1 + \omega_0) + \text{WFR}} \quad (46)$$

φ is replaced by 1, the fuel being always injected before the start of the combustion process. $\rho_{\text{fuel},2}$ of Equation (45) is then the amount of fuel aspirated per the swept volume V_d .

It is now possible to assess the influence of the main parameters of the intake and water injection systems on the temperature and density of the fresh mixture at the intake valve point.

4. Results and discussion

In order to illustrate the interest of such analysis, we will consider in this paragraph some practical, although quite idealised, applications to different types of engines and gaseous and liquid fuels.

Figure 5: Equilibrium Water-Fuel Ratio WFR_{eq} (dashed line) and resulting temperature drop $|\theta_{\text{wb}} - \theta_1|$ (solid line) vs. intake manifold temperature θ_1 and specific water content ω_1 , for a stoichiometric combustion ($\lambda = 1$) of pure octane (C_8H_{18}) with an intake pressure $p_2 = 1$ bar.

4.1. Spark ignited engine fed by pure octane

Let us consider the case of a spark ignited internal combustion engine, so with $\varphi = 1$, fed by pure *octane* (C_8H_{18}), so with $\text{AFR}_{\text{st}} = 15.03$. The combustion process is supposed for now as *stoichiometric*, so with $\lambda = 1$, and the intake pressure is for the moment such as $p_2 = 1$ bar. As explained previously, the equilibrium Water-Fuel Ratio WFR_{eq} and so the resulting wet-bulb temperature θ_{wb} primary depend on “ambient related” parameters such as θ_1 and $\omega_1 \propto \omega_0$, according to Equation (9).

Such an influence is presented in Figure 5 with a range of intake manifold temperature given by $30^\circ\text{C} \leq \theta_1 \leq 60^\circ\text{C}$, and a range of specific water content at the same point given by an ambient temperature $5^\circ\text{C} \leq \theta_0 \leq 30^\circ\text{C}$ and an ambient relative humidity $0.2 \leq \phi_0 \leq 0.9$.

It is clearly visible in the same figure that WFR_{eq} is primarily influenced by the intake manifold temperature θ_1 : the higher θ_1 , the higher WFR_{eq} because of a higher value of the corresponding equilibrium pressure p_{eq} , and thus of the maximum specific water content $\omega_{\text{vap,eq}}$ (see Equation (11) or Figure 3). Hence, the hotter the surroundings, the larger the amount of liquid water to inject into the fresh mixture in order to reach WFR_{eq} . Although the specific water content ω_1 has an opposite influence on the same parameter: the higher ω_1 , the lower the amount of water required to reach equilibrium, as expressed by Equation (13).

The influence of θ_1 and ω_1 on the attainable temperature drop is quite the same as for WFR_{eq} , i.e. $|\theta_{\text{wb}} - \theta_1|$ evolves in the same way as θ_1 and in the opposite way as ω_1 . This evolution is due to the approximate proportionality between this temperature difference and the WFR value, already highlighted by Equation (36).

Such a result has two very practical consequences on the use of the water injection process. When the ambient humidity increases — as for instance if the concerned engine operates in a tropical country — the attainable temperature drop decreases, because of the decreasing amount of injected water required to reach equilibrium between liquid water and steam. The same effect is obtained when temperature θ_1 decreases — when the engine operates in cold country for instance — for the very same reason. In this last situation though, we can reasonably hope that

Figure 6: Equilibrium Water-Fuel Ratio WFR_{eq} (dashed line) and resulting temperature drop $|\theta_{\text{wb}} - \theta_1|$ (solid line) vs. temperature θ_1 and intake pressure p_2 , for a stoichiometric combustion ($\lambda = 1$) of pure octane (C_8H_{18}) with $\theta_0 = 20^\circ\text{C}$ and $\phi_0 = 0.5$.

this detrimental effect would be partly compensated by the increasing density of ambient air, due to this low surrounding temperature.

In order to assess the interest of water injection on atmospheric engines operating at high altitudes, but also on engines using forced induction processes, the influence of the intake valve pressure $p_2 = p_1$ — according to hypothesis H7 — on the same parameters is presented in Figure 6, with $0.5 \text{ bar} \leq p_i \leq 2.5 \text{ bar}$. As it was already visible in Equation (11), the intake pressure p_2 has a similar influence as the specific water content ω_1 : it decreases the temperature drop $|\theta_{\text{wb}} - \theta_1|$ when increasing.

The amount of fresh mixture, and then of fuel finally aspirated into the engine, depends on θ_2 , and then of $|\theta_2 - \theta_1|$, as well as p_2 , as highlighted by Equation (45). The relative increase of fuel density $\rho_{\text{fuel},2}$ regarding to an equivalent “dry” intake with $\text{WFR} = 0$, defined as:

$$\frac{\Delta\rho_{\text{fuel},2}}{\rho_{\text{fuel},2}} = \frac{\rho_{\text{fuel},2}(\text{WFR} = \text{WFR}_{\text{eq}}) - \rho_{\text{fuel},2}(\text{WFR} = 0)}{\rho_{\text{fuel},2}(\text{WFR} = 0)} \quad (47)$$

is presented vs. θ_1 and p_2 in Figure 7. The interest of water injection for fresh mixture density improvement clearly increases with the intake manifold temperature θ_1 but decreases with pressure p_2 . Even though θ_1 increases until 80°C , the relative increase of fuel aspirated is not higher than 10% for atmospheric engine, so with $p_2 \simeq 1 \text{ bar}$. For the sole purpose of fresh mixture density increase, and so for increasing the amount of fuel aspirated into an internal combustion engine, the indirect water injection seems, at first glance, of a moderated interest, at least when compared to usual forced induction strategies. As a matter of fact, increases of about 15% of brake torques experimentally observed by some authors [14] are then probably not only due to such increase of density, but also to the effect of water addition on the engine cycle itself.

The influence of the actual WFR value on the fuel density increase is presented in Figure 8 for three different values of the intake manifold temperature θ_1 and for $p_2 = 1 \text{ bar}$. The fresh mixture gaseous density, and so the amount of fuel eventually aspirated, is always maximum at equilibrium, so when $\text{WFR} = \text{WFR}_{\text{eq}}$.

Because of the low concentration of fuel in the fresh mixture ($\text{AFR} \gg 1$), the Air-Fuel equivalent Ratio λ has no practical influence of the relative increase of fuel density, as presented in

Figure 7: Relative fuel density increase regarding to a “dry” intake $\Delta\rho_{\text{fuel},2}/\rho_{\text{fuel},2}$, as defined by Equation (47), vs. θ_1 and p_2 , for a $\lambda = 1$ and pure octane (C_8H_{18}) with $\theta_0 = 20^\circ\text{C}$ and $\phi_0 = 0.5$.

Figure 8: Relative fuel density increase regarding to a “dry” intake $\Delta\rho_{\text{fuel},2}/\rho_{\text{fuel},2}$ vs. WFR for $p_2 = 1$ bar, $\lambda = 1$ and pure octane (C_8H_{18}), with $\theta_0 = 20^\circ\text{C}$ and $\phi_0 = 0.5$.

Figure 9: Relative fuel density increase regarding to a “dry” intake $\Delta\rho_{\text{fuel},2}/\rho_{\text{fuel},2}$ vs. θ_1 and λ for $p_2 = 1$ bar, pure octane (C_8H_{18}), $\theta_0 = 20^\circ\text{C}$ and $\phi_0 = 0.5$.

Figure 9. This result will be of a great interest in future applications of such process, because optimizing the latter does not require to know the actual value of λ .

4.2. Compression ignited engine fed by pure cetane

According to the previous definitions, for a compression ignited engine fed by pure *cetane* ($\text{C}_{16}\text{H}_{34}$), we have $\varphi = 0$ and $\text{AFR}_{\text{st}} = 14.86$. Considering the same ranges of pressure and temperature as for the previous application, we obtain for example the values of WFR_{eq} and $|\theta_{\text{wb}} - \theta_1|$ vs. θ_1 and p_2 , presented in Figure 10. The comparison with the same results for octane, presented in Figure 6, shows no significant difference, emphasizing once again on the little importance of the fuel presence in the fresh mixture. Incidentally, it allows us to confirm the use of hypothesis H4 of page 3, dealing with the water injection occurring before or after the fuel one. Finally, the fresh mixture density increase vs. θ_1 and p_2 is presented in Figure 11, and leads to the same conclusion about the insignificant presence of fuel within the fresh mixture.

5. Conclusions

The actual effect, and so the practical interest of water injection (WI) process for engines performance improvement, has been so far assessed in an experimental manner only. However interesting these results are, they are not relevant to anticipate the increase of performance we can expect from the setup of such process on an existing engine. Furthermore, the same results do not say anything about the specific consequences of WI on the intake stroke, on the compression stroke, or on the combustion process.

The results presented in this paper are a first try to split up the effect of WI on the different parameters of a given engine. They can also be used in a very practical manner, thanks to the python software specifically created for so, whose use is presented in §A, and freely available at github.com/avaudrey/Py-ICE-WaterInjection, in order to forecast the consequence of the setup of a WI system on an existing engine, and more specifically on the fresh mixture density increase hence produced. In the future, we can hope that the framework here presented shall be used for example for a “real-time” control of the amount of water injected in engines.

Figure 10: Equilibrium Water-Fuel Ratio WFR_{eq} (dashed line) and resulting temperature drop $|\theta_{wb} - \theta_1|$ (solid line) vs. temperature θ_1 and intake pressure p_2 , for a stoichiometric combustion ($\lambda = 1$) of pure cetane ($C_{16}H_{34}$) with $\theta_0 = 20^\circ C$ and $\phi_0 = 0.5$.

Figure 11: Relative fuel density increase regarding to a “dry” intake $\Delta\rho_{fuel,2}/\rho_{fuel,2}$ vs. θ_1 and p_2 , for a $\lambda = 1$ and pure cetane ($C_{16}H_{34}$) with $\theta_0 = 20^\circ C$ and $\phi_0 = 0.5$.

Such an approach deserves now to be extended on other parameters affecting the performance of internal combustion engines, i.e. firstly the mechanical work consumed by the compression stroke, and then the rest of the engine thermodynamic cycle.

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Nomenclature

Notations		Subscripts	
c_p	Specific heat at cst pressure, [J/(kg · K)]	0	Surroundings
c_v	Specific heat at cst volume, [J/(kg · K)]	air	Dry air
h	Specific enthalpy, [J/kg]	d	Engine displacement or swept volume
ℓ	Specific enthalpy of vaporisation, [J/kg]	eq	Equilibrium vapour pressure of water
m	Mass, [kg]	dfm	Dry fresh mixture
$\dot{m}$	Mass flow rate, [kg/s]	fuel	Fuel
M	Molar mass, [kg/mol]	i	Intake point
n	Amount of matter, [mol]	liq	Liquid water
p	Pressure, [bar]	st	Stoichiometric proportions
r	Specific constant of the gas, [J/(kg · K)]	vap	Water vapour
T	Temperature, [K]	w	Water
V	Volume, [m ³]	wb	Wet-bulb temperature
W	Mechanical work, [J]	Acronyms	
x	Mass fraction	AFR	Air-Fuel Ratio, see (2)
Greek symbols		CI	Compression Ignited (Engines)
γ	Adiabatic index	HVAC	Heat, Ventilation and Air Conditioning
λ	Equivalent Air-Fuel Ratio	ICE	Internal Combustion Engines
ϕ	Relative humidity	SI	Spark-Ignition (Engines)
φ	Fuel presence indicator function	UHC	Unburnt HydroCarbons
ω	Specific water content, see (5)	WI	Water Injection
θ	Relative temperature, in [°C]	WFR	Water-Fuel Ratio, see (6)

A. The water injection python package

A.1. Introduction

As previously explained, the results presented in this paper have been obtained thanks to a specifically created software, coded in `python`, and freely available — under the licence GNU GENERAL PUBLIC LICENSE⁴ Version 3 — at github.com/avaudrey/Py-ICE-WaterInjection. This package uses two `python` scientific libraries called `numpy` (numpy.org) and `scipy` (scipy.org). Any help, advice or contribution is obviously welcome in order to improve or extend the performance of this code.

We will now briefly present how to use such a package, for example for the practical assessment of a possible water injection process on an existing engine. We will assume that the reader knows more or less what a *version control system* is — here we use `git` (git-scm.com) — and is familiar with the basics of *oriented object programming*. No real proficiency in `python` is required to use this package, which can be called from an external `*.py` file or from a command line, in pure `python` or using `ipython` or `jupyter` (ipython.org) [41].

⁴gnu.org/licenses/gpl-3.0.en.html

A.2. Creation of a fresh mixture

Basically, all the calculations can be done using only one `python` object, of the `FreshMixture` class. Any use of such a package must then begin with the creation of a new fresh mixture, as presented in Listing 1.

Listing 1: Instantiation of an object from the `FreshMixture` class, called here `mixture`.

```
1 # We import the water injection package, here called 'wi'
2 import water_injection_package as wi
3 # And from this package, we create a new fresh mixture called for
4 # example 'mixture', by an instantiation of an object of the
5 # 'FreshMixture' class
6 mixture = wi.FreshMixture()
```

Once the `mixture` object has been created, we can now select a specific fuel, here for example neat *ethanol* ($\text{C}_2\text{H}_5\text{OH}$), as presented in line 4 of Listing 2. The only other physical properties required for this fuel is the value of *specific heat at constant pressure* c_p , in $[\text{J}/(\text{kg} \cdot \text{K})]$.

Listing 2: A new fuel can be considered, from its chemical composition of the type $\text{C}_x\text{H}_y\text{O}_z\text{N}_u\text{S}_w$ and from its specific heat at constant pressure value c_p .

```
1 # If we want for example to use ethanol as fuel, we just need its
2 # chemical composition and its specific heat at constant pressure,
3 # aka 'cp'. Such a modification can be done in a very simple way:
4 mixture.fuel_composition = {'C':2, 'H':6, 'O':1, 'N':0, 'S':0}
5 mixture.fuel_specif_heat_at_cste_p = 1415.
6 mixture.fuel_name = 'ethanol'
7 # The combustion is supposed here as stoichiometric
8 mixture.air_fuel_equivalence_ratio = 1.0
9 # Actual Air–Fuel Ratio (AFR)
10 actual_AFR = mixture.air_fuel_ratio()
```

If the `air_fuel_equivalence_ratio`, so far noted λ , is entered, as in line 8 of Listing 2, a method called `air_fuel_ratio()` can be directly called in order to compute the actual AFR value, as presented in line 10. All attributes and methods included in this object are presented in Figure 12.

A.3. Other parameters and first results

The system surroundings and intake parameters can also be chosen using specific attributes, as presented in the same figure and in Listing 3.

Listing 3: The ambient and intake parameters can be entered in the same very simple way, and can lead to calculations of the equilibrium Water-Fuel Ratio (WFR_{eq} of Equation (13)) and of the corresponding wet-bulb temperature.

```
1 # Let us suppose the engine operating in a cold and quite wet
2 # ambience, the pressure being the default one, equal to the
3 # usual atmospheric pressure of 1 bar
4 mixture.ambient_temperature = 0.0
5 mixture.ambient_relative_humidity = 0.8
6 # The intake duct is at a higher temperature but at the same
7 # pressure, as in a 'naturally aspirated engine'.
8 mixture.intake_duct_temperature = 60.
9 # We can now extract the Equilibrium Water–Fuel Ratio value,
10 # corresponding to a maximum temperature drop
11 eqWFR = mixture.equilibrium_water_fuel_ratio()
12 # The minimum intake valve fresh mixture temperature reached
13 # is then the wet–bulb one
14 Twb = mixture.wet_bulb_temperature()
```

The main interesting results of such analysis, as for instance the equilibrium Water-Fuel Ratio or the corresponding wet-bulb temperature, can then be calculated in a very simple manner, as presented at the end of Listing 3. It is also possible to adjust the actual Water-Fuel Ratio value and to study its consequences on different parameters of the engine, as for example the fresh mixture fuel density at the intake valve point, defined by Equation (45), using the `intake_valve_fuel_density()` method.

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