The Ventilation of Streamlined Human Powered Vehicles

Ben Wichers Schreur*

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Abstract

Streamlined Human Powered Vehicles are often poorly ventilated. This is most obvious from the heated faces of riders and the steamed-up windows of the vehicles during long duration races and record events. Overheating of the rider affects his performance and may ultimately force him to give up, or face a serious health risk. These consequences are avoidable as a well-designed ventilation system can provide adequate cooling without requiring much power.

The necessity of cooling and ventilation

The human engine operates within a narrow temperature range. An increase of the body temperature of only a few degrees leads to a strong decrease of performance. A too-strong increase destabilizes the body's temperature control and causes a life-threatening situation. In an investigation of professional cyclists, who put out an average of a half horsepower (373 Watt) during a one-hour race, it was found that they were only able to sustain that level of output for 5 to 15 minutes on a bicycle ergometer [Abbott and Wilson, 1995]. A world hour-record in a state-of-the-art streamlined HPV requires similar power levels. Obviously adequate cooling is then required.

The heat production Q of an athlete in a streamlined HPV is a function of P, the power delivered, and the thermal efficiency f

$$Q = \frac{1-f}{f} \cdot P$$

The thermal efficiency of humans is less than 25%. An effective power output of 373 Watt therefore results in the production of at least 1119 Watt of excess heat, that will have to be removed by cooling. The mechanisms for cooling are convection (air flowing past the rider) and evapotranspiration (sweating.) This leads to an increase of tempature and humidity inside the fairing.

In equilibrium the heat produced is equal to the heat removed by ventilation

$$Q = m \cdot (c_p \Delta T + L \Delta q)$$

In this equation m is the massflow, $c_p \Delta T$ de heat uptake of an isobarically heated mass-unit of air and $L \Delta q$ de heat uptake due to evaporation. The massflow is equal to the product of the air density ρ and the volumeflow S

$$m = \rho \cdot S$$

The density of air is determined by the gas law

$$\rho = \frac{p}{RT}$$

for a given atmospheric pressure p and temperature T. The gasconstant R of a humid air mixture can be calculated from the specific humidity q en the gasconstants R_{dry} for dry air and R_{wet} for water vapour

$$R = (1-q) \cdot R_{dry} + q \cdot R_{wet}$$

The specific humidity can be related to the more commonly used relative humidity rh

$$q = \varepsilon \cdot p \cdot rh \cdot e_s(T)$$

Values for the saturation vapour pressure e_s as function of the temperature are available in tabular form from thermodynamic handbooks, e.g. [Tysma et al., 1984], $\varepsilon = 0.622$ is a constant.

A combination of the preceding equations leads to an expression for the airflow required for cooling

$$S = \frac{\frac{1-f}{f} \cdot P}{\rho \cdot [c_p \cdot (T_i - T_o) + L \cdot \varepsilon \cdot p \cdot (rh_i \cdot e_s(T_i) - rh_o \cdot e_s(T_o))]}$$

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The index i indicates de conditions within the fairing, the index o the environmental conditions. Values for the constants in these equations are $c_p = 1004\ J/kgK$, $L = 2.42\ MJ/kg$, $R_{dry} = 287\ J/kgK$ and $R_{wet} = 461\ J/kgK$. The effect of various conditions can be studied easily by programming the equation for the cooling airflow in a spreadsheet.[1] As an example the airflow for typical conditions $T_o = 15\ °C$, $T_i = 25\ °C$, $rh_o = 60\%$, $rh_i = 80\%$, $p = 101325\ Pa$, $P = 373\ W$ and $f = 0.25$ is calculated to be $S = 28\ liter/s$.

The humid warm air in the fairing will have a tendency to condense when it comes into contact with the window having approximately the outside temperature. Under the conditions from the preceding airflow calculation the relative humidity at the window is found to be

$$rh_{window} = rh_i \cdot \frac{e_s(T_i)}{e_s(T_o)} = 149\%$$

Such an oversaturation cannot be sustained. Condensing water vapour will fog up the window unless it is directly ventilated with cold unsaturated environmental air.

The liquid which is removed by ventilation is produced by the athlete. In the given example the rider loses 1.15 liter of fluid per hour. A grown-up human can tolerate a fluid loss of about 1.5 liters. Larger losses will quickly lead to a loss in performance. During longer duration events it is therefore important that the athlete compensates fluid losses by continuous drinking. [Abbott and Wilson, 1995] discusses the performance of the human engine in more detail.

The power required for ventilation

The ventilation of a streamlined HPV requires power which under IHPVA rules has to be supplied by the rider. The drag on the HPV resulting from the internal ventilation flow and the accompanying losses can be calculated from the momentum balance, under the assumption that the ventilation flow exits the vehicle at the environmental pressure

$$D = m \cdot (V - V_{ex})$$

where D is the drag, m the massflow, V the velocity of the vehicle and V_{ex} the exit velocity of the ventilation flow. The power that is required to overcome this drag is

$$D \cdot V = m \cdot V^2 \cdot (1 - \frac{V_{ex}}{V})$$

>From this equation it follows that the amount of power required for ventilation can be minimized by restricting the airflow to the minimum required by cooling and by making the exit velocity as large as possible. The latter requires that internal flow losses are limited. This equation also indicates that the maximum power lost to ventilation, under the conditions used in the previous calculations and at a vehicle velocity of $80\ km/h$ is $D \cdot V = m \cdot V^2 = 14\ W$. The power loss due to ventilation thus is limited even in the worst case. It should be remarked that here only internal flow losses are taken into account. In the last section of this article considerations for the design of the ventilation system are given that ensure that also the losses due to the disturbance of the external flow are kept low.

The internal flow losses are characterized by the loss in total pressure H, the sum of the static and dynamic pressures

$$H = p + \frac{1}{2}\rho V^2$$

By good design the ventilation air can be made to exit at the environmental pressure. In that case it follows directly that the pressure loss due to the internal flow is

$$\Delta H = \frac{\rho}{2} \cdot (V^2 - V_{ex}^2)$$

Unless a human powered fan is used the pressure loss will always be positive. The equation for the pressure loss therefore implies that the exit velocity of the ventilation air will always be smaller than the vehicle velocity.

The internal losses are predominantly dictated by the flow velocity within the fairing. It is beneficial to keep this velocity as low as possible. The cross-sectional area of current record-breaking HPVs is approximately $A = 0.3\ m^2$. A large part of the cross-section is taken up by the rider. If it is assumed that only 10% ($0.03\ m^2$) of the cross-section is available for throughflow then the required airflow for cooling, as calculated above, can still be achieved with a small average flow velocity of $1\ m/s$. Therefore it may be assumed that, if the intake air mixes well, the internal flow velocity is practically zero. Application of Bernoulli's law results for that case in [Drela, 1994]

$$V^2 = V_{in}^2 + V_{ex}^2$$

[1] The spreadsheet used by the author for this article, including the saturation vapour pressure table, is available from www.hupi.org/HPeJ

where V_{in} is the intake velocity and as before V the vehicle velocity and V_{ex} the exit velocity. The flow through the vehicle equates to

$$S = V_{in} \cdot A_{in} = V_{ex} \cdot A_{ex}$$

Combining these two equations yields the ratio of the exit velocity to the vehicle velocity

$$\frac{V_{ex}}{V} = \sqrt{\frac{1}{1 + (\frac{A_{ex}}{A_{in}})^2}}$$

and substitution of this ration in the previous result for the power loss yields

$$D \cdot V = m \cdot V^2 \cdot [1 - \sqrt{\frac{1}{1 + (\frac{A_{ex}}{A_{in}})^2}}]$$

It thus follows that the ratio of the size of the exit opening to the size of the intake should be as small as possible and because the size of the exit opening is fixed by the airflow required, it follows that the intake should be as large as other design considerations allow. In practice there are two important limitations to the size of the intake: the allowable disturbance of the external flow (see below) and the minimum airflow velocity around the body of the rider that is required to achieve efficient evapotranspiration and cooling. According to [Abbott and Wilson, 1995] this minimum velocity is about $3\ m/s$, based on a cooling surface of $1.8\ m^2$, an environment temperature of $15\ °C$ and a relative humidity of 80%. Using this velocity as the intake velocity in the previous calculations yields a power loss due to ventilation $D \cdot V = 0.15\ W$. This power loss is really next to nothing and certainly much smaller than the losses that result from the flow disturbances due to the decals that are stuck to the outside of HPVs to glorify the sponsor's name. There is then no reason to deny the rider adequate cooling and ventilation, provided that care is taken in the design of the ventilation system, in particular to avoid unwanted disturbances of the external flow.

Design considerations

In the previous sections it has been demonstrated that the power required to provide adequate cooling is in all cases small, even if a disadvantageous ratio of the intake and exhaust sizes is chosen. There are however certain aspects of the design of the ventilation system that have to be taken into account to realize an efficient cooling with minimum power loss in practice.

Placement of intake and exhaust

In the calculations of the losses due to ventilation only the internal flow losses are accounted for. To really achieve low drag care has to be taken that the intake and exhaust flows disturb the external flow as little as possible. If for instance the ventilation air is injected into the outer flow perpendicular to the wall it will (at least) displace the outer flow and lead to an increase in the form drag [Rogallo, 1940].

The best choice for the placement of the exhaust is at the back of the vehicle, where the ventilation air is blown out backwards directly into the vehicle wake. If the flow velocity in the wake is noticeably lower than the freestream velocity the injection of ventilation air at practically the freestream velocity may even lead to a drag reduction.

As was indicated the intake velocity must be low. Because the external flow near the stagnation point at the front of the fairing already has been retarded, the immediate surrounding of the stagnation point is the obvious place for the intake. There is then no need to slow down and thereby disturb the flow at some other point on the fairing.

As an alternative to a stagnation point intake sometimes a so-called 'NACA submerged duct' is used.

Figure 1: This magnificent tricycle of Jürg Birkenstock has a so-called 'NACA submerged duct' for ventilation. On the photograph the duct is the dark triangle in front of the window. This placement of the intake ensures that the ventilation air is aimed directly at the rider. Not visible on the photograph are two additional air intakes at the underside of the front of the vehicle (photograph courtesy of Robert Biesemans, 2001.)

Figure 1 gives an example of a NACA duct on an HPV. Such an intake sucks in the 'dirty' boundary layer air. The boundary layer on a record breaking HPV with extended laminar flow will be thin and a submerged duct may then not provide enough cooling air without disturbing the outer flow.

Apart from the intake and exhaust the fairing has to be airtight to avoid unintended outflows and disturbances. As an example extra openings for the ventilation of the window should be avoided. It is better to tap the windscreen ventilation air from the stagnation point intake.

Size of the intake and exhaust

Ideally the sizes of the intake and the exhaust are adjustable to allow the airflow to be varied according to need. To allow optimization of the external flow it seems prudent however to opt for a fixed intake at the nose of the vehicle. This fixed intake could be combined with a variable internal diffusor to control the flow.

The minimum size of the intake can be calculated from the required volume and speed range of the airflow. The minimum speed of the airflow was given above to be $3\ m/s$. The maximum speed will be determined by what the rider considers to be comfortable. This will be in the order of $10\ m/s$. Using values from the previous examples the minimum cross-sectional area of the intake is $28\ cm^2$, corresponding to a diameter of $6\ cm$. Using an internal diffusor the intake velocity can be higher and therefore the diameter of the intake smaller, but probably not much considering the fact that the flow velocity near the stagnation point is already reduced.

The optimal size of the variable exhaust can be calculated from the fixed size of the intake using the equations given above and is found to be of the same order as the intake. In the example given the size of the exhaust is $14\ cm^2$.

Shape of the intake and exhaust

In the previous calculations all dimensions of the airchannels are their effective dimensions. Poor design of the intake or exhaust can make their effective size considerably smaller than their geometrical size. Sharp edges on the inflow side of the airchannels will cause flow separation and choke the airchannel. The inflow edges of the intake and exhaust must therefore be sufficiently rounded. Another cause of choking the airchannels is the separation of the flow over valves or tubes, for instance those that are mounted in the airchannel to extract ventilation air for a window. It is better to separate off the window ventilation by splitting up the intake airchannel. A third reason for a reduction of the effective cross-section is the presence of sharp bends in the channel and these should therefore be avoided. Lastly the airchannels should be hydraulically smooth to avoid the build-up of a thick boundary layer.

A sketch of an intake conforming to these principles is given in figure 2.

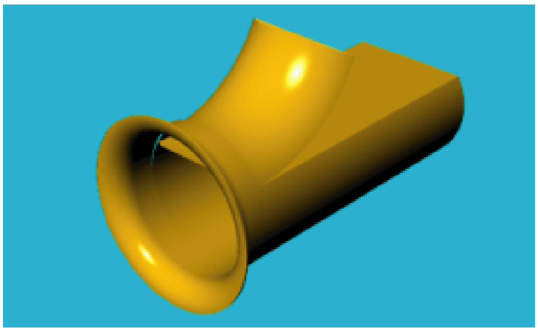

Figure 2: Principal design of a stagnation point intake. Properties are a well-rounded inflow edge, a smooth finish of the inner surfaces and a split of the intake in a small window ventilation channel and a larger cooling air channel.

Figure 3 suggests how a smooth constriction in the tail of the fairing can be used to accelerate the internal ventilation flow to almost the vehicle velocity before it is blown straight backwards into the vehicle wake. In this sketch the exhaust is placed at the top of the tail, because this shortens the route from the head and shoulders, where the ventilation is aimed at, to the exhaust, thus limiting the internal flow losses. A different positioning of the exhaust may be considered when it is known, from windtunnel tests or otherwise, that the external flow at that position is disturbed and can be improved by blowing out ventilation air (see also [Rogallo, 1940].)

Directed cooling and ventilation

The need for ventilation can be limited by using it as effectively as possible. The airflow therefore should be directed at the head and shoulders of the rider, as those are the prime areas of the body for heat exchange. This can be achieved by connecting the intake with a tube that ends near the rider's face. Finally, as has been shown before, part of the ventilation flow will have to be directed at the window to avoid condensation.

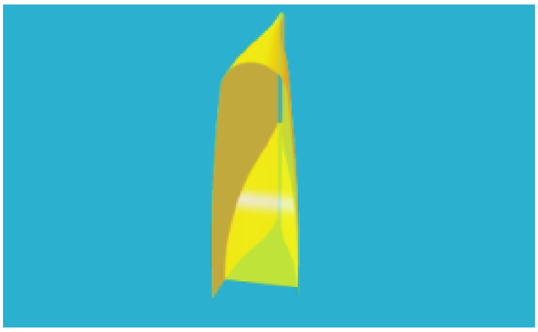

Figure 3: Principal design of an exhaust. The ventilation flow is accelerated through a smooth constriction in the tail of the fairing and blown out with the highest possible velocity into the wake.

References

[Abbott and Wilson, 1995] Abbott, A.V. and D.G. Wilson ed., 1995: Human-powered vehicles, Human Kinetics, Champaign IL, USA.

[Rogallo, 1940] Rogallo, F.M., 1940: Internal-flow systems for aircraft, NACA Report No. 713, National Advisory Committee for Aeronautics.

[Becker and Baals, 1940] Becker, J.V. and D.D. Baals, 1940: Wind tunnel tests of a submerged-engine fuselage design, NACA Advanced Confidential Report Oct. 1940, National Advisory Committee for Aeronautics.

[Drela, 1994] Drela, M., 1994: All-sealed fairings: ventilation gives very low drag, Human Power, vol. 11, no. 3, International Human Powered Vehicle Association.

[Tysma et al., 1984] Tysma, Sj. et al., ed., 1984: Polytechnisch Zakboekje, Koninklijke PBNA, Arnhem, Netherlands.