

# Mechanical problems of faired tricycles: investigation of features using smartphones (Low consumption electrical vehicles Eco marathon)

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**Abstract:** - The choices involved in designing a road-ready streamlined electric recumbent bicycle are numerous (tyres, suspension, centre of gravity, brakes, lighting, etc.). The fairing offers protection from poor weather conditions and improves the aerodynamics but increases the mass by 20 kg. This weight of the fuselage and an average speed of 50 km/h requires motorization. The vehicle's range is defined by the energy required by a given motor as a function of road grades and average speed. Vehicle performance profiling requires instrumentation and recordings that can be achieved with a simple smartphone (lux meter, deceleration brakes, acceleration suspension, GPS and power, acoustic noise). The mechanical and electrical characterization is presented in this article with the aim of identifying the limits and possibilities of the vehicle. This article demonstrates that the realization of a low-energy vehicle is an excellent teaching tool that can be reintroduced into everyday life on any vehicle.

**Key-Words:** - Eco Marathon Challenge, motor power, project-based teaching, e-bike, e-velomobile, cycle brakes, cycle tyres, suspension, cycle lights, smartphone.

## 1. Introduction

Low energy consumption electric vehicles are becoming active participants in our daily travel. These types of electric vehicles have emerged from the Eco-marathon challenges and meet the demand of minimizing CO<sub>2</sub> emissions as well as future energy transition [2,3,7,9]. velotaxis, cargo bicycles and fully faired tricycles are becoming commonplace, with masses of 50 to 150kg for average speeds of 45 km/h.

Some cycles have a range of 250 km, for average speeds of 45 km/h and a top speed of 80 km/h on the flat road. As can be seen in the following figure, the electrical engineering department of Aisne IUT has made numerous prototypes with low electrical power consumption.

The educational benefit of the classic e-bike for students is the ability to ride the prototype and thus effectively understand the physical-mechanical interactivity.

At high speed, the effects of different settings and of technological choices are clearly felt. Consequently, so-called "extreme" high-performance electric cycles (fitted with a 3000W motor and a 21 elements Lipofer, 70V, 20A.h, 1440W.h [1, 2]) are excellent multi-technology educational projects for students. The full electrical charge takes one hour with a 20A charge from a standard socket.



fig 1: Different prototypes of electrical velomobile  
( 50kg to 85kg)

On these vehicles, many questions arise:

- **What is the impact of the mass and shapes adopted on consumption?**
- **How should the masses be distributed as a function of ergonomics, braking, traction or propulsion, etc.?**
- **What force and braking techniques are required, based on mass, velocity, deceleration desired?**
- **What choice of tyres should be made given their grip and the deceleration required?**
- **How are the lighting and signalling devices to be chosen?**
- **How powerful should the motor be, and what energy capacity should the battery have?**
- **What is the availability of different technologies (battery, instrumentation, motor,...)?**

**- Is pedalling thus non-negligeable supply of energy ?**

A safe, reliable, high-performance vehicle will be achieved through a large number of choices and compromises.

Given the online information sources freely and openly available, these choices ought to be easy. However, manufacturers and distributors no longer provide technical data on their products.

This is due to the comparisons that can be rapidly carried out by the competition, but also because of the lack of scientific background of the equipment buyer.

The technical and marketing arguments are sometimes simplified to the extreme where they become more marketing than technical.

Manufacturers certify their products as meeting the standards but differentiate them with a row of stars, while in fact the technology often exceeds the values required by the standards.

The prototypes have on-board instrumentation with data recording, which allows for dynamic measurements (speed, power, deceleration, energy ...). Accordingly, the vehicle becomes a true testbed.

We will now show how to make a low-energy vehicle and answer some of those questions.

**2. Resistive Power of a velomobile**

The consumption of a vehicle is proportional to its mass [1]. Modeling the resistive power of a vehicle can be given by the following equation (1):

$$P_{resistive}(W) = k_{Aero} \cdot S^3 + (Cr + Slope) \cdot M \cdot g \cdot S(km/h) / 3.6$$

With g, the gravitational constant, M the mass of the vehicle, Cr resistive coefficient of the tyre.

E-bike type	$k_{aero}$ W/(km/h) <sup>3</sup>	$k_{rol}$ (W/km/h) = MgCr3.6
Classical e-bike 30kg	0,0065	5.5
Trike 40kg	0,005	6
Bentriker 40kg	0,003 à 0.004	6
velomobile 50kg	0,001 à 0.002	6.5
e-car 1500 kg	0,015	79

Table 1 : coefficient for a vehicle with a person of 80kg with a Cr=0.005, slope=0

From these previous factors, the resistive power is shown as a function of speed. We can see that the aerodynamic coefficient only starts to be dominant from 30 km/h, prior to that the rolling resistance is decisive.

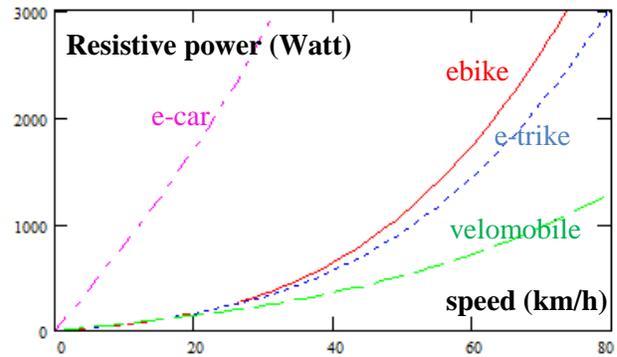


fig 2: Resistive power vs speed on a flat road (without pedaling) of different cycles

Tricycles called “velomobiles”, even with a fairing of 20 kg and 10 kg of “machinery”, demonstrate a good trade off to achieve a vehicle with low power consumption.

The aerodynamic coefficients and rolling resistance can be determined with a GPS smartphone using an application that saves CSV data files (comma-separated values). This data is then processed by a spreadsheet.

One can observe in the following figure speed, power and energy as a function of distance through an Excel treatment of the data file recorded by the smartphone app "my tracks".

**3. Fully faired trike: velomobile**

To reach high speeds safely with low consumption, road holding and reduction of the drag coefficient can be achieved with a fairing.

The fairing is generally made of composite materials (glass fibres, carbon fibres or kevlar). Where the fairing carries the wheels, it must be strengthened in places to support the mass of the driver and the forces generated by pedalling.

A person must be able to enter and exit without assistance through a removable cover.

The velomobile needs to be proof against the weather and tyre spray thanks to a set of seals and gutters. Polycarbonate (Lexan) is often used for windshields. This material is cheap, has a degree of flexibility and is reasonably scratch-resistant.

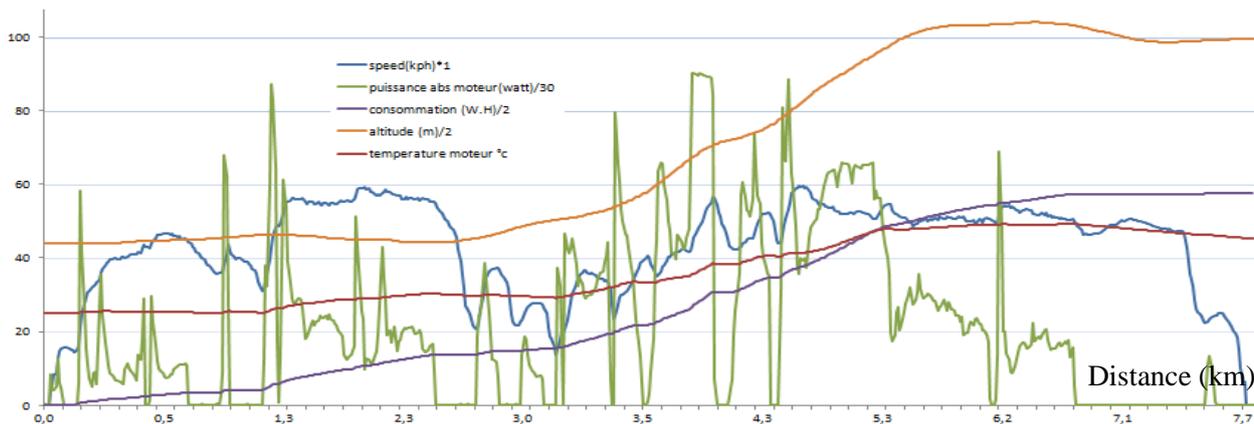


fig 3: the smartphone app "my tracks" Recordings of speed, power, energy, altitude, engine temperature versus the distance

Concerning the aerodynamics, firstly the penetration surface should be minimal to reduce the most significant factor  $C_y$  (front surface). Second, to reduce the  $C_x$  (drag coefficient), the fairing of a velomobile should be carved in the shape called "water drop" and salient forms avoided to the maximum. Thus the rear portion must be "pinched" to minimize turbulence and the front face takes the form of a "lobe". The use of a single rear wheel provides a thinned tail and also avoids the use of a differential. Logically it will be this rear wheel that will be motorized. The choice of two front steering wheels on a tricycle ensures stability at rest, but does not allow for counteracting the centrifugal force when cornering, as one can on 2 wheels. Fully faired two-wheel recumbent cycles have a clear advantage in that they minimize the surface penetration into the air.

On the other hand, at rest they must be equipped with a device to keep them upright. Many 2 wheel faired prototypes have been made, but to date are rarely used. Indeed, their handling can be dangerous because of the turbulence caused by lateral wind gusts, or when overtaking or meeting large vehicles. For the trike, where the center of gravity is concerned, efforts should be made to place it as low as possible in order to minimize penetration into the air and to allow higher speeds when cornering due to the risk otherwise of overturning.

The choice of the wheelbase (130 cm), the track width (70 cm to 80 cm) and the height (less than 100 cm) is subject to the ergonomics of pedalling and the need to clear obstacles such as pavement kerbs, access cycle paths and negotiate gaps (e.g. doorways).

Determining the best position of the masses within the vehicle can be done algebraically or with software such as "Solidworks".

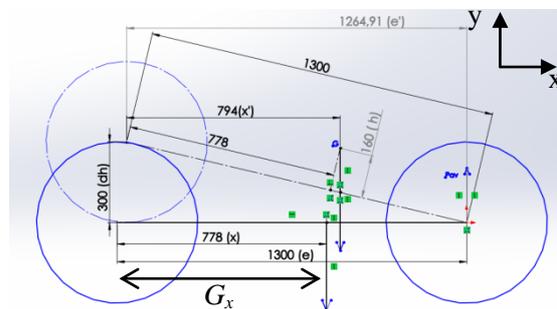


fig 4: Study of trike center of gravity from the masses added using Solidworks.

The maximum speed based on the radius  $R$  of the curve and the width of the vehicle track geometry (TG) corresponds to the following equation theoretical (2):

$$S_{maxi}(km/h) = 3.6 \sqrt{\frac{R}{mass_{total}} \cdot \frac{TG \cdot mass_{front} \cdot 9.8}{2 \cdot height_{gravity\ center}}}$$

The vehicle speed when cornering will be all the greater the wider the track, the lower the centre of gravity, and the nearer the centre of gravity is to the front wheels.

Example: Take a cycle with a mass of 137 kg (person+vehicle+equipment), 20-inch wheels in front, a 26 inch rear wheel and the center of gravity at a height of 412 mm. The weight on each front wheel is 402 N and 566 N on the rear. It follows that the maximum speed for a turn radius of 20 m will be about 45 km/h.

The adoption of a pendulum geometry for the steering would allow for an increase in this speed, though the mechanisms used would greatly complicate the build.

During emergency braking, there is a transfer of weight onto the front of the vehicle generating a danger of tilting over the front axle. The weight on

the two front wheels due to braking deceleration is determined by the following equation theoretical:

$$\text{weight}_{\text{front } y} = \left(-\frac{d\text{Speed}}{dt} \cdot G_y + g \cdot G_x\right) \cdot \frac{\text{Mass}}{\text{distance}_{\text{centre}}} \quad (3)$$

Where  $G_x$  is the gap between the rear wheel and the centre of gravity and  $G_y$  the height of the vehicle's centre of gravity relative to the ground.

The coordinates of the centre of gravity in this case will be  $G_x$  778 mm and  $G_y$  412 mm.

The weight on the rear wheel corresponds to the following equation:

$$\text{weight}_{\text{rear } y} = M \cdot g - \text{weight}_{\text{front } y} \quad (4)$$

Example: with braking causing a deceleration of  $-10 \text{ m/s}^2$ , the weight of the front and rear wheels where they meet the ground is given by (5):

$$\text{weight}_{\text{front } y} = (10 \cdot 412 + 9.8 \cdot 778) \cdot \frac{137}{1300} = 434N + 804N = 1238N$$

$$\text{weight}_{\text{rear } y} = M \cdot g - \text{weight}_{\text{front } y} = 137 \cdot 9.8 - 1238 = 104N$$

During heavy braking, the rear wheel holding weight will therefore decrease. The location of the centre of gravity will therefore be an important point to consider before constructing a prototype.

#### 4. Characteristics of cycle brakes

The fairing and the electric motor resulting in a significant overload, we are entitled to ask whether the traditional bicycle disc brakes can still be used. Moreover, what is the difference in braking for 160 mm and 203 mm disk brakes?

According to the guideline standard DIN 79100, on the basis of 100 kg, the braking values are 340 N at the front, and 220 N at the rear (i.e. a front / rear braking distribution of 60%/40% for a cycle).

In addition, the legislation calls for the brake to be able to dissipate a power of 1050 W at 12.5 km/h during 15 minutes (thus equivalent to a braking force of 300 N).

Note that when the brake pads reach maximum temperature (fading), the friction coefficient (0.5 in general) is divided by 2, thereby reducing the braking force by 50%.

Brake pads are cooled principally by the motion of the disc and the movement of air proportional to the vehicle speed.

The discs are available in conventional stainless steel or "steel-aluminum sandwich" with diameters of 160, 180, or 203 mm. There are floating versions, and they may also be drilled for weight reduction and increased ventilation.

There are also various brake pad technologies supporting different temperature levels and with

shorter or longer lives: The materials used are of type:

- "organic" (250°C, life time =x), 8€
- "semi metallic" (500°C=> life time =2x), 11€
- "ceramic" (sintered)=>650°C=> life time 3x), 18€

Sintering is a method of producing pads from powders through the application of heat and pressure.

At a speed of 20km/h, the thermal resistance  $R_{\text{TH}}$  of a 203 mm brake is 0.1°C/W, while for a 160 mm disk it is 0.15°C/W. The thermal time constant of the two discs is about 40 s.

Therefore, the temperature reaches a steady state in about 2 minutes. The thermal capacity  $C_{\text{TH}}$  of a disc of 160 mm is 400J/°C, and 600J/°C for a disc of 203 mm.

Therefore, at a speed of 20km/h, the pads reach 250°C requiring a dissipation of power of 1666 W (300N) for a disc of 160 mm and 2500 W (450N) for its counterpart of 203 mm.

Thermal resistance decreases proportionally as a function of the speed. So at a speed of 40 km/h, the thermal resistance of a 203mm brake is 0.05°C/W while it is 0.075°C/W for a 160 mm brake. At 40 km/h, the braking power may double to reach a temperature of 250 °C.

The difference in braking torque between a disk of outer radius 160 mm one of and 203 mm with the same caliper corresponds to the following equation theoretical (6):

$$\text{torque}_{\text{braque}} = N \cdot \mu \cdot \text{force}_{\text{pads}} \cdot \frac{2 \cdot (\text{Re}^3 - \text{Ri}^3)}{3 \cdot (\text{Re}^2 - \text{Ri}^2)} \approx N \cdot \mu \cdot \text{force}_{\text{pads}} \cdot \left(\frac{\text{Re} + \text{Ri}}{2}\right)$$

Where  $\mu$  is the coefficient of friction of the pads,  $N$  the number of pads and  $R$  radius of the disc external and internal.

This equation does not take into account any faults in contact pressure between pad and disk.

Example: for a disc of 160 mm,  $\text{Re}$  (=80mm) corresponds to the outer radius of the disk,  $\text{Ri}$  (=65mm) is equal to  $\text{Re}$  less the width of the pad (here 15 mm).

Given a thrust force (exerted by the caliper brake) on the pads of 1000 N, with a coefficient of friction of 0.5, the torque for a disc of 160 mm is 228Nm against 295 Nm for a disc 203 mm.

The maximum braking force tested for a single BB7 caliper is 800N with a disk of 160 mm against 1100 N for the 203 mm. This corresponds to the ratios of the previous equation.

There are caliper brakes with 1, 2 or 4 mechanical or hydraulic pistons. When a temperature of 300°C

is reached in the hydraulic circuit, the contents reach boiling point and then become compressible. In this case, braking is degraded significantly. In addition, this type of incident involves the replacement of the liquid and the seals in the circuit.

The average braking power at a given average speed also corresponds to equation 1.

From equation 1, we can observe in the following figure that the average power which is either recoverable, or requires dissipation by the brakes, is linear in relation to the slope.

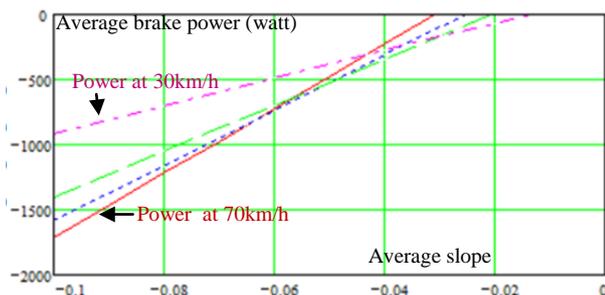


fig 5: Average braking power as a function of the road slope for a velomobile of 137kg with average limited speeds of 30, 50, 60 and 70 km/h.

We can observe that the higher the speed, the greater the braking power and the higher the temperature.

The temperature reached by a 160 mm brake will thus correspond to the following equation (7):

$$\text{Temperature}_{\text{brake}} = \frac{R_{\text{TH}} \cdot \text{Speed}_{\text{RTH}}}{\text{Speed limit}_{\text{average}}} \cdot \text{Power}_{\text{average}} + \text{Tamb}$$

Example: At 30 km/h with a slope of -10% at a temperature of 25°C, the braking power will be -915W. The brake temperature reaches after a few minutes:

$$\text{Temperature}_{\text{brake}} = \frac{0.15^{\circ}\text{C} \cdot 20\text{km/h}}{30\text{km/h}} \cdot 915\text{W} + 25^{\circ}\text{C} = 120^{\circ}\text{C}$$

At a speed of 70km/h, braking power reaches -1715 W, but the decrease in thermal resistance results in a temperature reduction:

$$\text{Temperature}_{\text{brake}} = \frac{0.15^{\circ}\text{C} \cdot 20\text{km/h}}{70\text{km/h}} \cdot 1715\text{W} + 25^{\circ}\text{C} = 98^{\circ}\text{C}$$

In our case, Avid BB7 mechanical calipers were chosen for their reliability and ease of adjustment. In addition, the braking progress relative to the lever with this type of material is exemplary (study comparing the force on the lever with the force on the caliper piston).

During an emergency brake, the mechanical stopping distance is proportional to the mass. It corresponds to the following equation (8) :

$$\text{Distance}_{\text{brake}}(\text{m}) = \frac{1}{2 \cdot \text{deceleration}} \cdot \frac{\text{Speed}^2}{3.6^2} = \frac{\text{Masse}}{2 \cdot \text{Force}} \cdot \frac{\text{Speed}^2}{3.6^2}$$

The emergency braking force with a 160 mm disc is 800N (stopping distance: 14m at 50 km/h, 9m at 40km/h and 5m at 30km/h) against 1100 N with the 203 mm (stopping distance: 10.5 m at 50 km/h, 7 m at 40 km/h and 3 at to 30 km/h).

Braking anticipation and maintaining a safe distance between 2 vehicles are essential.

A rear brake on a trike can be dangerous because it may cause nose-to-tails if not handled carefully (loss of grip of the rear tyre because of weight transfer, known as drift). There is thus no use for a rear brake on the velomobile and none is fitted.

On a tricycle equipped with a single control for the 2 front brakes, it is difficult to correctly adjust the braking share between right and left. Frequent and numerous mechanical adjustments are needed to maintain a balance between the two forces.

Therefore, it is preferable to have separate controls for the two front brakes. In a straight line, this split command allows for equal right and left braking by different pressures on the levers: the steering is done instinctively. In addition, it is possible to slow down slightly on one wheel during turning. For example, if we want to turn right, the right brake alone can be applied gently.

The temperature increase on the brake discs (during emergency braking) corresponds to the following equation (9):

$$\text{Temperature}_{\text{Brake}} = \frac{\text{mass}_{\text{vehic}} \cdot \text{speed}_{\text{max}}^2}{2 \cdot C_{\text{th}}} + \text{Temperature}_{\text{brake average}}$$

The average power to be dissipated for the vehicle to stop is proportional to the mass and depends on the temperature reached by the disk (which may also depend on various previous braking). Indeed, during emergency braking one must take into account the average temperature of braking and not ambient temperature.

Example : to stop a vehicle of 100 kg moving at 40 km/h (kinetic energy of 1.7Wh) with a deceleration of -6m/s<sup>2</sup>, the braking force must be 600 N with a stopping time of 1.85s, average power of 3.3 kW and the temperature of the brake discs will reach 120 °C.

However, to stop a vehicle of 140 kg in the same conditions, the braking force must amount to 840 N, the average power 4.6 kW and the temperature will reach 139°C.

On the following figure, we can observe the temperature of a brake disk after a descent causing a temperature rise of 86°C (The average braking power is 700 W for 30 seconds).

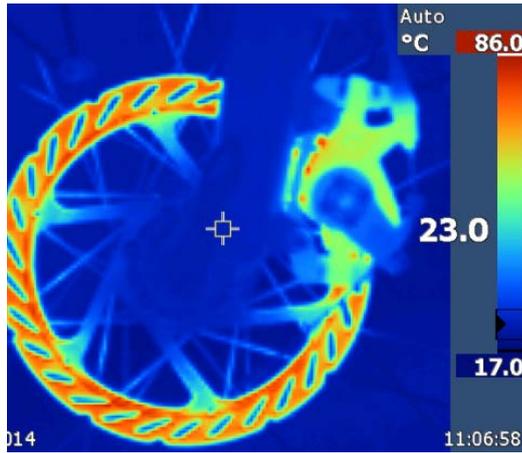


fig 6: Temperature of the brake disc of 160mm and its caliper.

Given the low price and mass differences between the 160 mm and 203 mm discs, the 203mm were chosen.

With the electric motor, it is possible to recover some of the kinetic energy into electrical energy. The regenerative power will be limited by the battery capacity to absorb this electric charge without deterioration. The electric brake will have the advantage of limiting the use of the mechanical brake.

The BB7 bicycle brake on a velomobile can be used in emergency braking and downhill. We now check the grip of the tyres in these conditions.

### 5. Characteristics of tyres

In this field, it is important to reflect on the size and the rolling resistance of tyres, and any loss of grip during braking or when cornering in dry and wet conditions.

Depending on the structure and type of the compound, a tyre can be characterized by its longitudinal friction coefficient  $\mu$ , lateral  $\tau$  and its rolling resistance coefficient  $C_r$ . This information is not generally provided by the manufacturers.

These three coefficients are dependent on the compound-ground pairing and the surface condition at the instant considered (temperature, cleanliness, presence of water, pressure, etc.). For a given pairing,  $\mu$  varies slightly with the mass of the vehicle, but greatly with the slip ratio described by this equation theoretical :

$$\text{slip} = (\omega(\text{rad.s}) \cdot R(\text{m}) - V_{\text{vehicle}}(\text{m/s})) / V_{\text{vehicle}} \quad (10)$$

R: radius of the wheel,  $\omega$ : angular velocity of the wheel in rad/ s and V: vehicle speed in m/s.

On the front wheels, a 100% loss of grip, or slip (locked brakes) leads to a complete steering-loss of the vehicle. If the locking-up occurs on the rear wheel, there is a risk of drift.

Heavy braking (Figure 7) has two phases:

- rise to a maximum deceleration, which corresponds to a rise to a maximum coefficient of friction  $\mu_{max}$  which is reached with a slip rate approaching 0.1 (10%).
- then a decrease in the friction coefficient towards a value  $\mu_{min}$ .

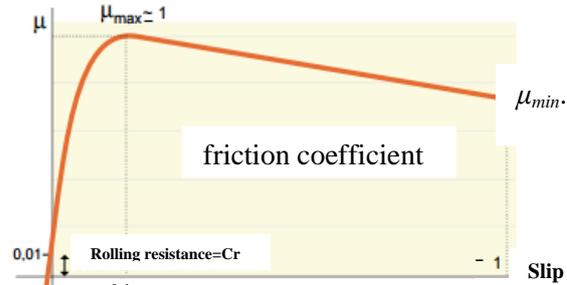


fig 7: Friction coefficient (grip) vs slip of the tyre and the  $C_r$

The rolling resistance of a tyre,  $C_r$ , is on average 0.01 but may reach 0.003 in the best case [6].

The resistive force and resistive power are determined by the following equations theoretical:

$$F_R (\text{N}) = M \cdot g \cdot C_r \quad (11)$$

$$\text{Resistive Power (W)} = F_R \cdot \text{Speed (km/h)} / 3.6 \quad (12)$$

Example : For a vehicle with a total weight of 100 kg and a  $C_r$  of 0.005, the rolling resistance force  $F_R$  will be 5N at 45km/h and the power lost will be 62W.

The  $C_r$  coefficient can be determined by a natural deceleration for a speed below 15 km/h with the following equation theoretical:

$$C_r = \frac{1}{2 \cdot \text{distance}_{\text{stop}}(\text{m}) \cdot g} \cdot \frac{\text{speed}_{\text{initial}}^2(\text{km/h})}{3.6^2} \quad (13).$$

The resistive power also depends on the roughness of the road surface, the inflation and width of the tyre but especially on the area in contact with the road and the tyre's loss of shape. On the following figure can be observed the power as a function of the tyre width (60mm and 37mm) and its pressure. The widest tyre has a lower resistance than the narrow 37mm tyre.

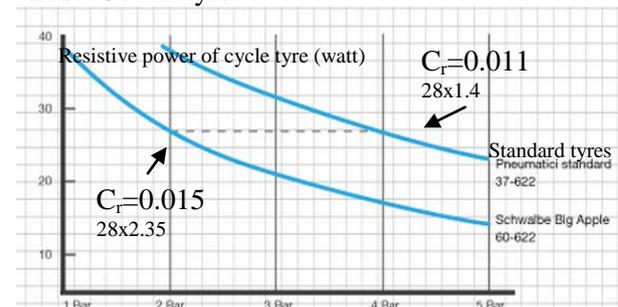


fig 8: Resistive power for 2 tyres of different widths at different pressures [8] M=40kg, Speed 15km/h.

The tread pattern is of no importance on the road. Indeed, a slick tyre adheres better than a treaded tyre as the contact area is larger whether on a dry or wet road.

There are straight-walled tyres with strengthened sides designed for tricycles. These products allow a lower consumption of energy than round-walled tyres which tend to crush under load and when cornering. (Recall that a tricycle does not tilt into corners as does a two-wheeled cycle). In general, the hardness of a tyre reduces rolling resistance but will penalize turning and braking performance.

Tyre manufacturers perform a trade-off between the rolling coefficient and the adhesion of their compounds to position their products. The presence of a camber in the front wheels of a trike to improve stability prohibits the use of straight-walled tyres.

The losses from the tyres due to the viscoelastic mechanisms can be approximated by the relationship:

$$P_{lost}(W) \cong Vol \cdot hys \cdot A \cdot Fr \tag{14}$$

With: Vol: Volume of distorted compound, hys: the hysteresis loss of the compound, A: amplitude of deformation and Fr: the stress frequency (Hz).

Each tyre can be modelled as a spring-damper system. As shown in the following figure, there will therefore be a loss of energy as a function of the frequency. The red curve corresponds to a high hysteresis rubber with good traction while the blue curve is a compound with low hysteresis. The green line is a mixture balancing low rolling resistance and good grip. According to Figure 7, the maximum grip coefficient  $\mu$  of a tyre in dry conditions is approximately 1, while in wet it will change to 0.5.

So in theory, a tyre could produce a braking force without loss of grip corresponding to two previous equations :

$$F_{brake} = M \cdot \mu \cdot g = M \cdot \frac{dv}{dt} \tag{15}$$

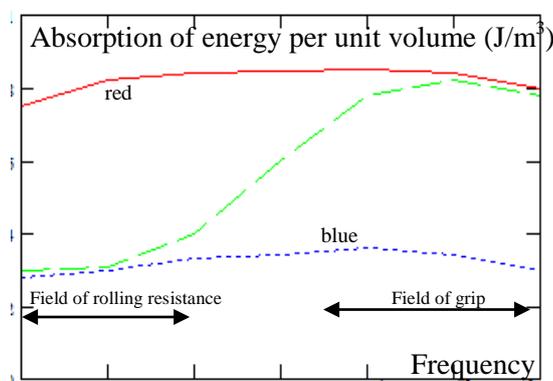


fig 9: Absorption of the energy of a tyre vs frequency.

The deceleration without loss of grip is independent of the mass and cannot reach below  $-10m/s^2$  in the dry or less than  $-5m/s^2$  in the wet.

Therefore, oversized brakes will cause skidding if the manner of their use is not controlled as it is in cars with an ABS (Anti-lock braking system).

Just as with tyres, suspension systems absorb the irregularities in road surfaces and play a crucial role in relation to the behaviour of the vehicle.

## 6. Suspension, steering

The suspension is intended to reduce the oscillations of the vehicle due to road irregularities by energy dissipation. The suspension improves the contact of the wheels on the road, relieves tyres during heavy impacts (pothole, etc.) and improves operator comfort.

The suspension should absorb low frequency undulations of the road (for example a speed bump), but also filter out all the high frequencies generated by the rough road surfaces. This frequency depends on the speed of the vehicle and is expressed by the following equation:

$$f(Hz) = Speed(km/h) / (width(m) \cdot 2 \cdot 3.6) \tag{16}$$

For example, with a bump in the road of width 10 cm at a speed of 36 km/h, the frequency is 50Hz whereas at 72km/h, it will be 100 Hz.

If the width of the obstacle is divided by 10, the frequency will then be multiplied by 10.

In general, the oscillation is not maintained except when doing tests to make measurements of the suspension according to the frequency (Figure 12)

All suspensions use a stiffness k (helical metal or pneumatic spring) and a hydraulic piston for damping.

The spring allows the suspension of the vehicle in its initial position while permitting movement of the wheels relative to this position. Damping attenuates the oscillations caused by deflections of the springs.

On the tricycle, deflections of 5cm at the rear and 2cm at the front were selected in large part due to space constraints.

In our case, the rear suspension is supported by an oscillating fork. On the suspension, 3 adjustments are available: preload, extension and compression [10].

Preload determines the initial height of the loaded vehicle (person in the vehicle). In general, it is at 1/3 of the maximum deflection, set with the collar of the metal spring.

Compression is the absorption coefficient when the wheel rises. This coefficient determines the force

required to compress the shock absorber, and is proportional to the speed of movement. The extension determines the same coefficient as the wheel moves downwards.

Characteristics of the force as a function of the movement of the suspension can be observed in the following figure.

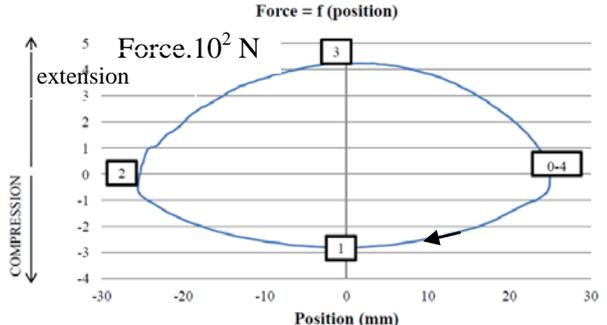


fig 10: Force vs the position of the rear suspension

From figure 10, the spring rate  $k$  may be determined by the following equation as a function of displacement:

$$k = 300 \text{ N} / 0.025 \text{ m} = 12 \text{ kN/m} \Rightarrow 120 \text{ N/cm}$$

On the spring itself, stiffness is usually indicated for a given deflection value (the unit can be N/cm or lbs/inch).

This stiffness ensures that the maximum deflection will not be reached (no bottoming out).

In this case, thanks to the preload ring, for a rear-wheel load of 550 N and a setting at 2/3 of the stroke (5cm), the following force will be required to bottom out:

$$\Delta F = \Delta x \cdot k = (0.05 \text{ m} \cdot 2/3) \cdot 12000 \text{ N/m} = 400 \text{ N} \quad (17)$$

The compression and extension damping is given by the following curve:

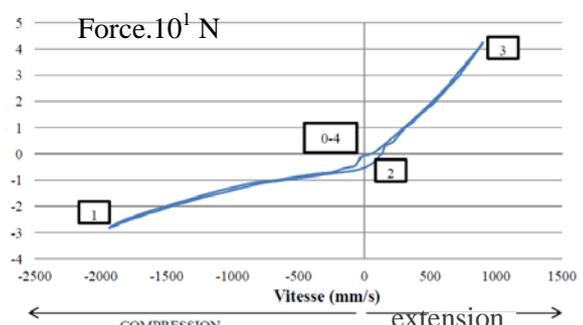


fig 11: Damping force vs the speed of the main piston.

From the previous curve, for compression and extension, the ratio of the speed depending on the force corresponds to the following values:

$$\begin{aligned} c_{\text{compression}} &= F / V = 30 / 0.002 = 15 \text{ KN/m/s} \\ c_{\text{extension}} &= F / V = 40 / 0.001 = 40 \text{ KN/m/s} \end{aligned} \quad (18)$$

Usually, to ensure good contact between the wheel and the road, the compression damping coefficient must be lower than that for extension. On mid-range suspensions, compression damping and rebound are adjustable with a ratio of 4 and 1/4. The theoretical study of the differential equation describing a mass-spring-damper system provides the following features with a natural frequency  $\omega_0$  and the quality factor  $Q$ :

$$\omega_0 = \sqrt{k/M} \quad Q = (\sqrt{k \cdot M}) / c \quad (19)$$

For our example, still with a load of 550 N at the rear, stiffness with a coefficient 100 times greater than the weight giving a small displacement and a natural frequency of 1.6Hz offers an interesting case.

To guard against oscillation, i.e. to obtain a quality coefficient of less than 0.5, a damper coefficient of 4 times less than the stiffness value should be chosen. The equation and the following figure show the transfer function of the amplitude of movement of the vehicle based on the amplitude of the road surface depending on the frequency:

$$\frac{y_{\text{vehicle}}}{y_{\text{road}}}(i \cdot \omega) = \frac{1 + i \cdot \omega / \omega_0 \cdot Q}{1 + i \cdot \omega / \omega_0 \cdot Q - (\omega / \omega_0)^2} \quad (20)$$

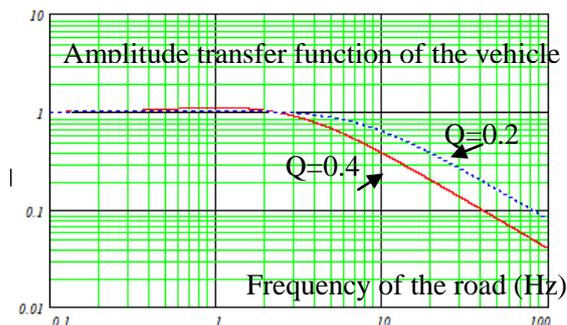


fig 12: Attenuation of movement amplitude through the suspension as a function of the frequency  $f_0=1.6\text{Hz}$  and 2 different quality factors: 0.2 and 0.4.

If the damping is equal to the stiffness divided by 2, the quality factor is 0.2. In this case, the attenuation will be lower as can be seen on the curve above.

Example: at a frequency of 50Hz, the attenuation will be 0.079 with the quality coefficient of 0.4, and 0.152 with a coefficient of quality of 0.2.

When the quality factor is less than 0.5, then the dominant time constant is equal to the following equation:

$$\tau(s) = c/k \quad (21)$$

With  $c=k/4$  then the dominant time constant is equal to 0.25s. Therefore, after dropping down a kerb, the fork returns to its original position (without no

rebound) in a response time of 3 times  $\tau$ , or 0.75s. This time allows for a comfortable suspension.

It may be noted that the spring rate and damping are highly dependent on each other to obtain satisfactory suspension. The difference between the rebound damping and the compression means that the theory can only give an approximation of how to adjust a suspension system.

Unfortunately, manufacturers do not reveal much technical data about their suspension. Therefore, it is necessary to test and identify the values.

The acceleration and movement of the vehicle resulting from the suspension can be measured with the accelerometer of a Smartphone, properly attached to the vehicle.

For example to adjust the suspension extension, a person must climb into the vehicle, pushing down on the rear of the vehicle until the suspension bottoms out, then measure the acceleration of the rebound. To set the resistance, you must drop down from a kerb in the vehicle to measure the acceleration of compression.

The aim of these operations is to return to the initial position relatively quickly with few or no oscillations.

The acceleration measurement is carried out with a sampling period of 10ms with a resolution of 0.1m/s<sup>2</sup>.

Indeed, during the descent from a kerb of a certain height with a damped response of  $Q < 0.5$ , acceleration versus time can be approximated by the following equation theoretical :

$$acce_y(m/s^2) = (height / \tau^2) \cdot e^{-t/\tau} \tag{22}$$

In the case an response under damped where  $Q > 0.5$ , acceleration, speed and position as a function of time can be approximated by the following equations  $\omega_r > 1/\tau_a$ :

$$acce_y(m/s^2) \approx (height \cdot \omega_r^2) \cdot e^{-t/\tau_a} \cdot \cos(\omega_r \cdot t) \tag{23}$$

$$speed_y(m/s) \approx (height \cdot \omega_r) \cdot e^{-t/\tau_a} \cdot \sin(\omega_r \cdot t) \tag{24}$$

$$movement_y(m) \approx height \cdot e^{-t/\tau_a} \cdot |\cos(\omega_r \cdot t)| \tag{25}$$

The damping time constant  $\tau_a$ , the pseudo pulse period, corresponds to the following equations (26):

$$\tau_a(s) = 2 \cdot M / c \quad \omega_r = \omega_0 \cdot \sqrt{1 - (1/2 \cdot Q)^2} \approx \omega_0 = \frac{2 \cdot \pi}{T_0}$$

From the preceding equations, it is possible to identify approximately the characteristics of the suspension. Where the response is not fully damped, stiffness  $k$  and the damping coefficient correspond to the following equations;

$$k(N/m) = (2 \cdot \pi / T_0)^2 \cdot M \tag{27}$$

$$c(N/m/s) = 2 \cdot M / (T_a / 3) \tag{28}$$

On the following figure, the period  $T_0$  and the time  $T_a$  can be observed during the extension of the damper after putting it in compression. In the following case, the suspension setting also changes the stiffness and damping. Nor is it possible to adjust the extension and the compression separately. The app used is "accelerometer Monitor", the speed and displacement are determined by numerical integration.

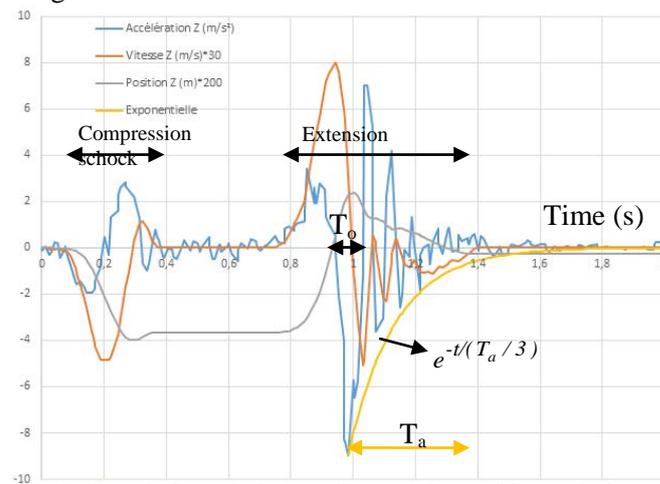


fig 13: Dynamics of " low " suspension adjustment with a preload of 50 kg.  $c=750N/m/s$ ,  $k=410kN/m$

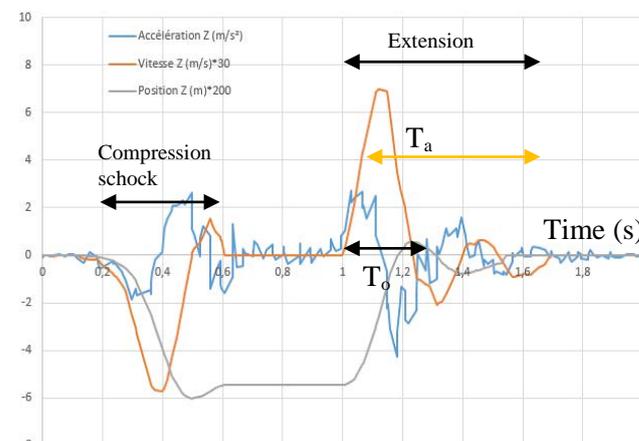


fig 14: Dynamics of the " high " suspension setting with a preload 50kg.  $c=500N/m/s$ ,  $k=40kN/m$

In the previous figures, the speed to return the oscillating fork to its original position is virtually identical with a low or high suspension setting. For the high setting, there is virtually no overshoot of the position because the amplitude of acceleration is strongly damped, while for the low setting suspension, the opposite applies

This is far from the desired values  $c=2750N/m/s$ ,  $k=5500KN/m$  (but for this, we must invest in a new suspension).

The tyre also has a damping role. In fact, the damping of the tyre can be increased by using "suspension" tyres such as the "Big Apple" from Schwalbe which has a special side wall and must be inflated to a low 2 to 3 bar.

One can observe the following bar chart but how is the gain in comfort measured by the manufacturer?

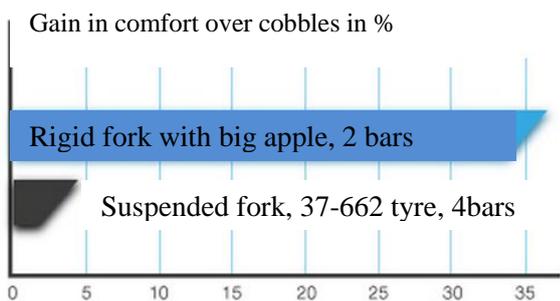


fig 15: Gain in comfort with Big Apple tyre [8].

The kerb-drop test, which is not presented here, may be used to determine the role of this tyre in affecting the pressure in the suspension.

Since 2013, "fat bikes" use 26x4.00 (102-559) tyres inflated to between 0.5 bar to 1 bar that replace relatively expensive and heavy suspension systems found in traditional mountain bikes.

The frequency spectrum of the accelerometer can also make use of the acceleration data. The spectrum is used to display the amplitude of the oscillation for each frequency while moving.

With the Discrete-time Fourier transform (DTFT) using a sampling frequency  $f_e$  of 100Hz, the maximum frequency of the spectrum will be  $f_e/2$ . With 256 samples over a time of 2.56s, the accuracy will 0.39Hz ( $f_e/256sample$ ). The precision depends on the number of samples (example 100 samples of 10ms in 1s, the accuracy will 1Hz).

In the following figure, one can observe the spectrum of the acceleration during 2.56s with 256 samples. The effective value of the amplitude of the acceleration is the sum of the square root of the amplitude for each frequency.

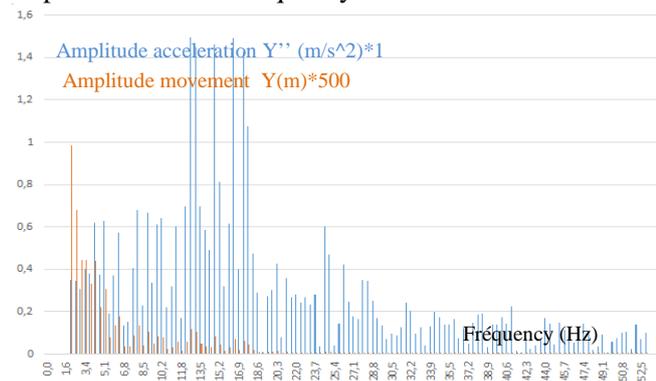


fig 16: Spectrum of the acceleration signal in height Y, of the body of a Velomobile at 20km/h on tarmac/asphalt.

The amplitude of the displacement  $y$  of movement with respect to the amplitude of the acceleration corresponds to the following equation :

$$movement_{y(m)} \approx \left( \sum_{n=1}^{128} \left( \frac{acc_{y_n}}{(2 \cdot \pi \cdot f_e \cdot n / 256)^2} \right)^2 \right)^{1/2} \quad (29)$$

This equation corresponds to the double integration of a sinusoidal signal. This movement is attenuated hyperbolically against the frequency dependent on the amplitude of the acceleration, and thus is important at low frequencies.

Example : from the previous figure, taking only 2 significant frequencies 2Hz and 12,6Hz the following equation permits calculation of the displacement:

$$movement_{y(m)} \approx \frac{1}{2 \cdot \pi} \left( \frac{0.34^2}{(2Hz)^4} + \dots + \frac{1.44^2}{(12.6Hz)^4} + \dots \right)^{1/2}$$

From the data recorded by the smartphone application, measurements are taken and processed by a specialized spreadsheet or software. Thus from the previous figure, the RMS value (Root Mean Square) of the acceleration is  $3.36m/s^2$ , that of movement is 2.2mm.

But to properly study the suspension of a vehicle, it is best to use a forced oscillation as in roadworthiness tests for cars.



fig 17: Roadworthiness testing of brakes and shock absorbers

By a system of plates oscillating at variable frequency, the system compares the dynamic and static weight (percentage ratio). In other words the capacity of the suspension to maintain the wheel in contact with the ground.

A value of 0% corresponds to a wheel which at any given moment may lift off the ground (not particularly helpful in terms of roadholding). At 100%, the wheel exerts a force on the plate, i.e. will never lose touch with the ground.

The study of shock absorption covering both front and rear wheels together is also very important. Indeed, when the rear wheel reaches the same obstacle, it will generate the same cycle as the front wheel. The interference between the front/rear and right/left oscillations cause pendular movements (pitch, roll or galloping) which are relatively easily modelled but will not be presented in this article.

The front suspension calls for a wishbone with ball joint which holds the spindle of the wheel hub. This triangle wishbone can be achieved very easily in fibre.

The profile of the fairing does not allow a very large steering angle of the front wheels. But negotiating corners is also an essential element in daily commuting, dictating the choice of non-enclosed wheels, which allow for a 3m turning circle at the expense of the aerodynamics..

To simplify the steering linkages, some have fitted a steerable rear wheel. However, this solution introduces a slight delay in response.

While the steering of a bicycle is relatively easy to build, that of a tricycle is much more restrictive because of the linkages involved and the need for precise adjustment of the wheel alignment.

The following figure shows a U-shaped handlebar system allowing for one-handed steering. This system unfortunately requires greater width between the sides of a central passenger compartment compared to a tiller system, but offers greater stability above 50km/h.



fig 18: Handlebar (inside the electric Velomobile Leiba XStream) [5]

Cycle lighting has always been problematic, the question being: see or to be seen ?

## 7. Lighting and signaling

Cycle lighting is a crucial issue in terms of safety. LED lighting requires advanced technical

knowledge if only for an understanding the manufacturers' datasheets.

The components require selection based on characteristics such as lux, lumens, the candela, the angle of emission, power consumption, different lenses, etc.

At our chosen speed of 54 km/h (15m/s), it is clearly important to be able to illuminate the roadway and the edge to a minimum distance of 30m (2s to spot obstacles). Furthermore, French law requires the front and rear lights of a vehicle to be visible at 150m.

LEDs use approximately 10 times less power than halogen lamps.

In order not to adversely affect the aerodynamics, the light must be integrated into the fairing.

Using the light meter of a smartphone, one can map the actual lighting values against distance and direction.

In our case, front lights with 15W LEDs (€50) with a 9° lens and positioned 75cm from the ground will light an ellipse at a distance of 10 to 30m.

The ground lighting is 300 lux at 10m, at 17m 100lux, at 80Lux 24m, 27m 63 lux.

The illumination is theoretically proportional to the square of the distance can be determined by the following equation (30) but it depends on the optics also:

$$E_{\text{clairment}}(\text{lux}) = E_{\text{clairment}}(1^{\text{er}} \text{ distance}) \cdot \frac{(1^{\text{er}} \text{ distance})^2}{(2^{\text{eme}} \text{ distance})^2}$$

The illuminance can be determined by the following equation with  $\theta$  the half angle of the optical:

$$E_{\text{clairment}}(\text{lux}) = \frac{\phi(\text{lumen})}{\pi \cdot (\tan \theta \cdot \text{distance})^2} \quad (31)$$

With 1500 lumen, the illumination at 10 m should be 770 lux, but there is a part of the light flux that is greater than 9°.

Brightness will be 75lux at 10m, 35lux at 20m and 18 lux at 35m.

For the rear lights, a power of 3W with an angle of 8° is ample for daylight visibility at 150m ( 62lux at 1.7m, 11lux at 3.5m). For rear signalling at night, the choice of lens is not very important and the wider required angle of 120° was used with 10 LEDs.

Turn signal indicators will require at least 6 LEDs with a 40° angle to be visible by day.

There are commercially available strips of adhesive LEDs for the stop and turn signal lights (€10 per meter). They are easy to install and highly weather-resistant IP65.

The technical information available concerning lenses and tape LEDs lacks sufficient detail. Equipment will need testing to validate its choice. Acceptable lighting including signalling for a velomobile thus requires a consumption estimated at around 24 W.

Lighting and signalling equipment being available in 12V, a DC/DC converter is required to adjust the battery voltage (72V or 48V to 12V).

### 8. Acoustic noise of a velomobile

The movement of a vehicle causes vibrations inside and outside thus creates a noise pollution.

From 35 km/h, with visor open, there is an air displacement in the ears that causes significant and annoying noise. Using a motorcycle helmet or ear plugs will reduce this noise.

Sonometers applications such as (digi Sound meter and analysis) were used. Smartphone must be checked and calibrated to the microphone, its amplification chain and its bandwidth. A windscreen must be used with an external microphone that will reject wind noise.

Measurements were carried out with the sound level meter to 1 meter outside the vehicle on the sidewalk to know the inconvenience to pedestrians and internal vehicle.

It may be noted that noise is attenuated inside the car in comparison to noise outside. While it is the opposite for the Velomobile.

	20 km/h	30 km/h	40 km/h	50 km/h	60 km/h
velomobile smooth road	47 dBA	54	58	60	62
velomobile grainy road	67 dBA	74	77	79	83
Car standard smooth road	51 dBA	60	64	66	69
Car standard smooth road	73dbA	74	75	77	79dBA
	60dbA		62		65dBA

Table 2 : Decibel, exterior and interior vehicle vs speed

Note: 3 dB corresponds to a doubling of increase of sound pressure:

$$10\log 10^6 \text{ more } 10\log 10^6 = 60\text{dB more } 60\text{dB} = 10\log(2 \cdot 10^6) = 63\text{dB}$$

In addition, if the distance doubles then the surface of the noise quadruple. Therefore, the acoustic intensity is divided by four corresponding to the square of the distance.

So Doubling the distance in free field is removed 6 dB. Ear plugs can reduce noise by 21 dB to 37 dB SNR (signal to noise ratio).

On the following figure, a microphone was placed at 30cm from the electric motor to measure the noise of the wheel. A 75km/h, the velomobile wheel motor runs at 600 rev/min and produces a noise of 84 dB. 9s and 20s, the motor is coasting and the

noise decreases logarithmically. A few revolutions per minute, the noise is only 60 dB (corresponding to a sound of a normal conversation at 1 m).



fig 19: Noise wheel electric motor to 30 cm

To give an idea, the sound power is 40dB for a public libraries and 84dB for a busy street.

Again a another smartphone application that allows to understand the noise pollution by the students and to have notions about acoustic noise.

### 9. Application to education

Many simple experiments can be used to test the theoretical aspects, after which the student has the opportunity to carry out real-world tests and adjustments. With the aid of a smartphone he/she can determine the performance of a vehicle, the losses caused by the tyres, aerodynamics (GPS, accelerometer), the performance of the lighting (light meter) and measure the noise generated by movement (decibel meter).

Not all apps are of equal quality and do not replace the right measurement tool. Consequently, the app itself should first be tested, often leading to a reworking of the data based on the sensors used (standardisation, calibration, linearization, filtering, data processing ...).

There is very little information about these apps, their accuracy, their measurement methods ... however, it is highly educational to test an application, or to build one's own.

The construction of a vehicle destined for the race-track will be completely different from one designed for use on the road.

In a high performance velomobile (track-ready), all the vehicle's faults will become evident and require optimization and advanced adjustments.

A compromise must constantly be made between performance, comfort, consumption and costs.

In creating these prototypes all the strategies familiar to the field of engineering can be found: the choice of materials, sizing, energy saving, eco-design, weight gain, safety, legislation, ergonomics. However, students need a sound basis in mathematics and science to process the recorded data and interpret their measurements to produce a synthesis of the vehicle's behaviour.

Man is a sensor, but with very fuzzy accuracy and a subjectivity that precludes a scientific approach to making improvements.

Students also use videos with the overlay data from the instrumentation still with using a simple smartphone.

Students do also videos with the data from the instrumentation still with using a smartphone [10].

## 10. Conclusion

The article presented some of the problems involved with a faired tricycle to demonstrate that it can be a complete multi-technological system.

These issues are case studies in mechanical and electrical engineering. This article has described the main features of the velomobile to give an idea of the current state of the art.

In addition, the velomobile is a study medium for addressing all issues relating to CO2 emissions in the transport sector. For example, greater range can be achieved by the addition of solar panels [7, 9].

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fig 20: Eco-marathon prototype

[https://www.youtube.com/watch?v=Hn\\_kl141dxA&index=45&list=PLfZunVn\\_gcq7EOurXuWU2sRFmh6CbiLiL](https://www.youtube.com/watch?v=Hn_kl141dxA&index=45&list=PLfZunVn_gcq7EOurXuWU2sRFmh6CbiLiL)



fig 21: The vehicles of yesterday and today



fig 22: the velomobiles does not fear the rain