Chalmers University of Technology

Shell Eco-Marathon 2007

Drive-Train Design Report

Presented By

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Introduction:

In effort to design a highly efficient chain drive two alternatives were considered:

1. Belt Drive
2. Chain Drive

The belt drive if designed properly could deliver power an overall efficiency up to 98%; this along with the light weight and low moment of inertia made the belt drive an attractive solution to our problem. These advantages are however offset by the fact that we need a large rear sprocket of considerable high weight and inertia similar to the one on last year's car.

The chain drive offers a similarly high efficiency. The chain for our application can be optimized by using the smallest possible pitch and thus minimize the chain weight; the main advantage with using a chain is the ability to manufacture with ease light weight sprockets. This is not only good to reduce the overall weight of the car but also a sprocket with a high moment of inertia will take large amounts of energy to accelerate up to speed. This energy will not be recovered and will be lost in the bearings as we intend to implement a free-wheel in our design.

From the above discussion we decided that the chain drive is a more practical and efficient solution.

Design Considerations

The first design consideration is the fact that in general larger diameter sprockets are more efficient in transmitting power. This has been discussed in many papers and the idea is that for the same gear ratio a pair of smaller sprockets will produce higher internal forces in the chain, this in turn greatly increases friction losses in the chain that more than offset the losses which will be incurred due to the increased weight of larger diameter sprockets especially that we intend to manufacture the sprockets from light weight aluminum.

From this it was decided to make the diameter of the rear driven sprocket as large as possible close to the rim dimension of 16 inches. This will also allow us to achieve the largest possible diameter of the driver sprocket for any gear ratio that we set later.

Another thing to consider is the variation of the speed of the chain as it engages and disengages with the sprocket. This is due to the variation of the lever arm's length from the time the chain bush impacts the sprocket tooth till it gets seated on the sprocket. This speed variation increases with decreased number of teeth and causes increased chain noise and wear.

The percent speed variation is given by the following formula:

$$\frac{\Delta v}{V} = \frac{m_{\max} - v_{\min}}{V} = \frac{\Pi}{N}\left[\frac{1}{\sin(180/N)} - \frac{1}{\tan(180/N)}\right]$$

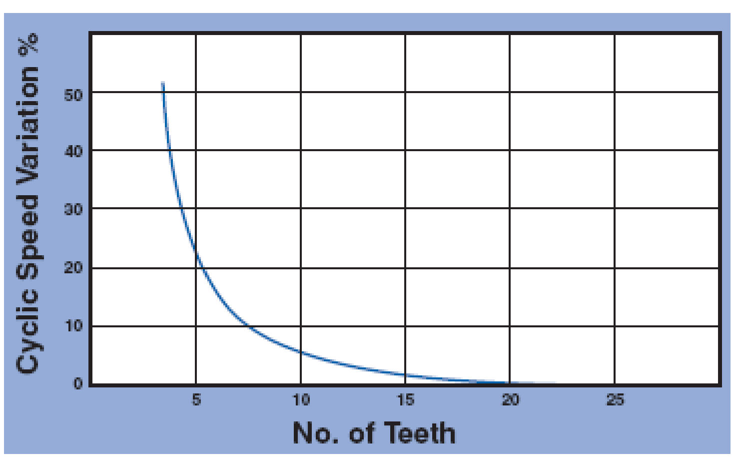

Figure 1

We see from the graph above that with 21 teeth the driver sprocket we will have a negligible speed variation of approximately 1% and thus this is the best condition. But because of the low torque of our engine we will end up with a small driver sprocket for a high gear ratio and thus we will possibly have to use 17 teeth on the driver as an acceptable compromise.

Another design consideration is the chain pitch. One advantage of having a small pitch chain is reducing the angle of articulation. The angle of articulation is the angle that the chain makes when it is tangent to the pitch circle; it is the angle through which the chain rotates to mesh with the sprocket. A larger angle of articulation will produce more wear and as a result increased elongation of the chain and thus it is advantageous to have a larger number of teeth for a given sprocket size to reduce this angle.

In accordance with the above it was decided to find the smallest possible chain. The engine only delivers only a peak 1.25 N.m of torque and as seen from chain catalogues even the smallest pitch chains can withstand more than a 1000N of force; this means that the strength of the chain is not going to be an issue here. The smallest standard chain we found was a ¼ inch 6.3 mm pitch chain which is what we went for.

Rear Sprocket Design

Since we don't yet have an engine map and thus we don't know how high we are going to rev the engine, the gear ratio still unknown. Thus the rear sprocket was designed to have the largest possible diameter so we will have flexibility to choose the ratio we want with a corresponding driver sprocket.

Keeping in mind the possibility for the need of replacement in the case of failure, the sprocket was designed in three parts and thus can be taken apart and replaced without the need to remove the rear wheel. This way we will be sure that we will not have to re-align the rear wheel which is not an easy job.

Below is the picture of the part that makes up ½ of the sprocket. The two ends are of conjugate geometry so the complete sprocket is obtained by assembling the part shown in the picture below twice.

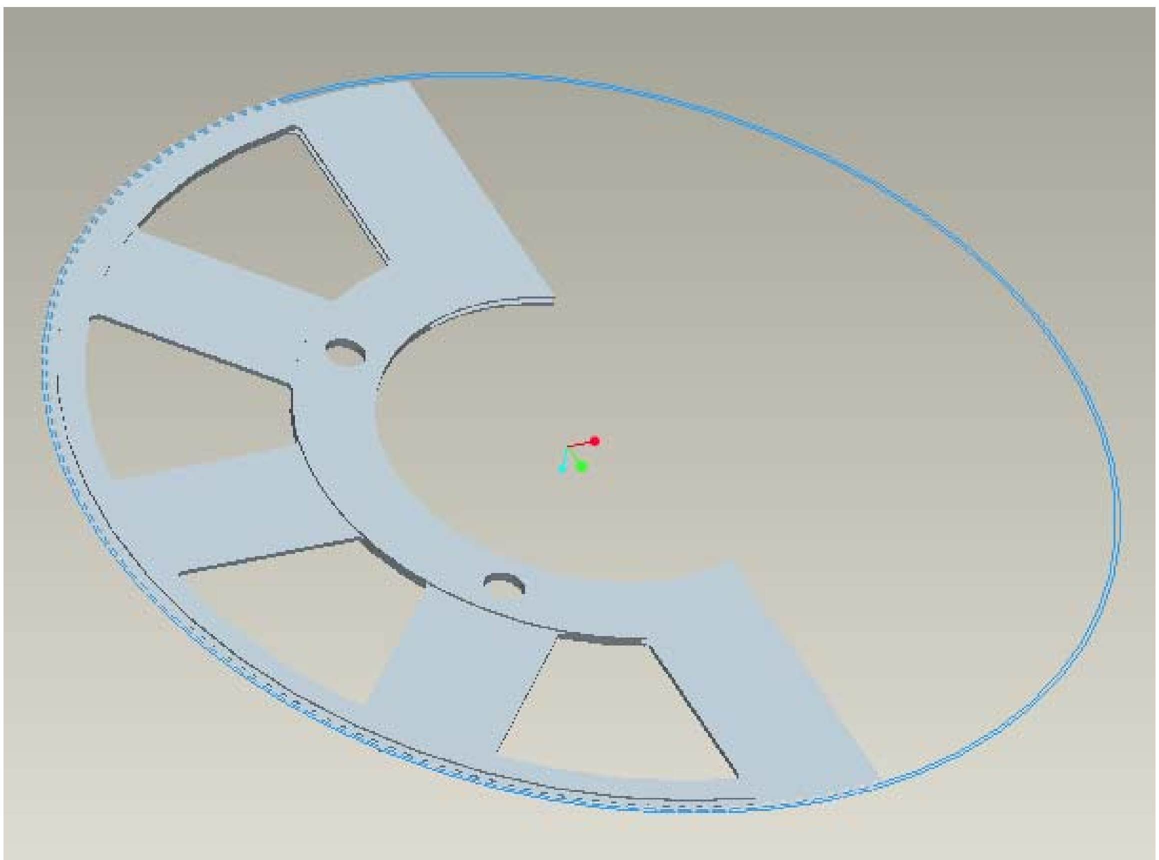

Figure 2

The detail of the geometry that allows the sprocket to be taken apart in two parts is shown in the following two following pictures. The two parts will be screwed in multiple places to achieve a rigid structure given the fact that the sprocket is of a large diameter. The pitch diameter is equal to approximately 39cm (i.e. close to the diameter of the rim 40.64cm). The necessity of such a large diameter will be shown from calculations later in the report.

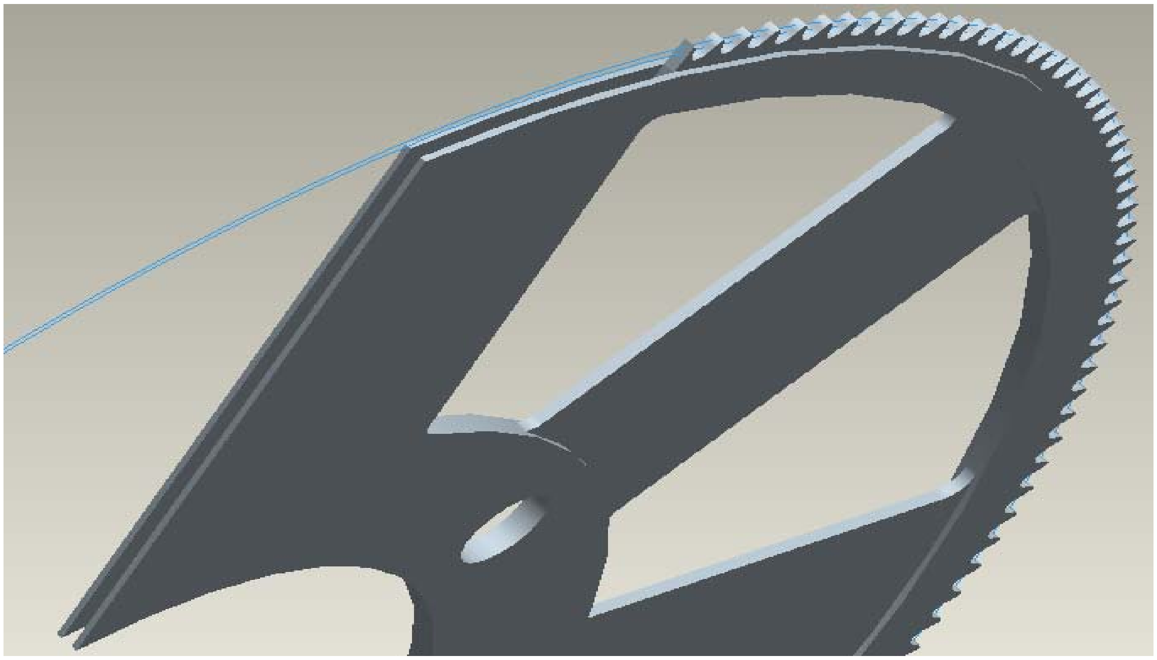

Figure 3

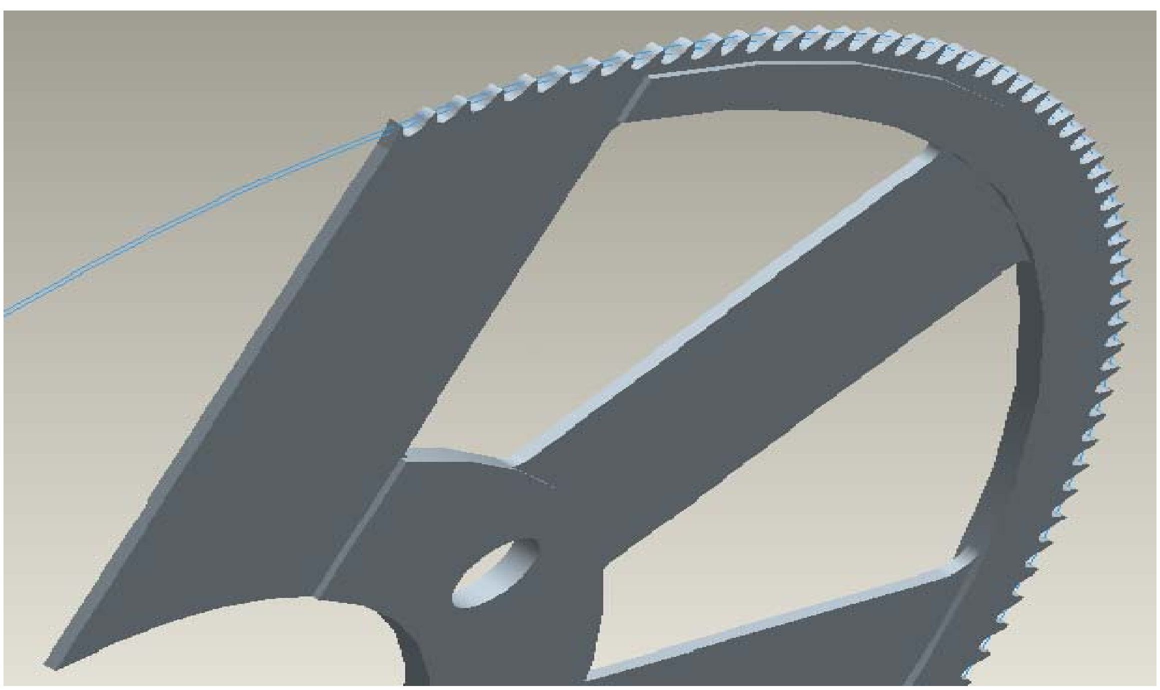

Figure 4

The part shown below fixes the two halves of the sprocket to the free-wheel. The dimensions of the free-wheel are still unknown, so the geometry of the middle of this part will be modified later when we have the free-wheel.

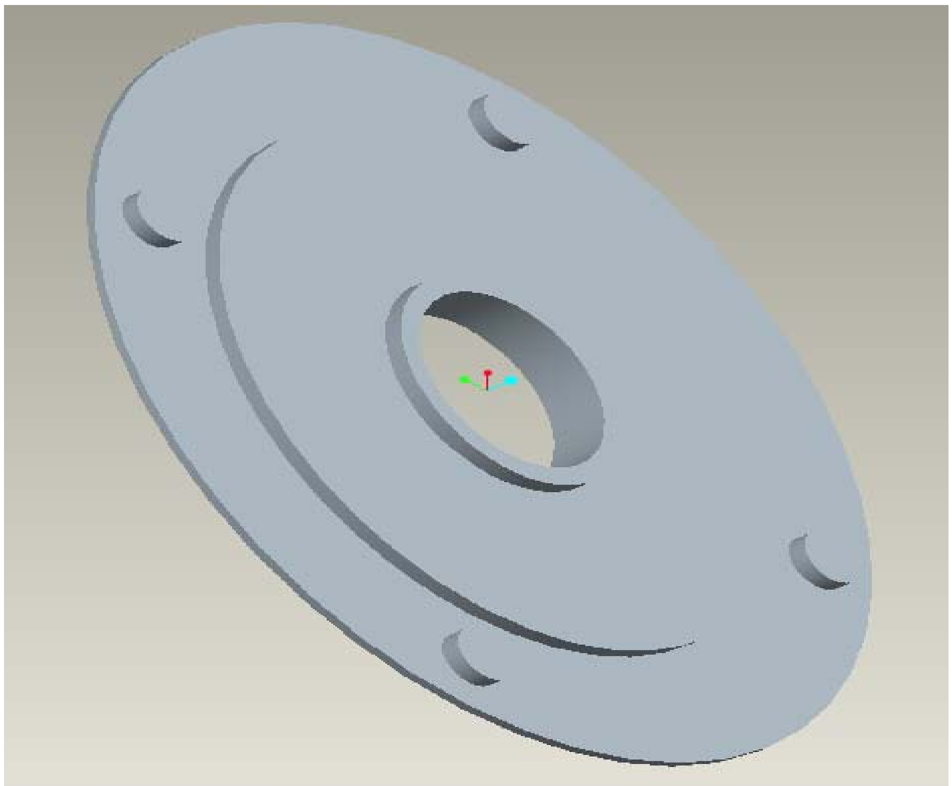

Figure 5

The entire assembly of the sprocket is shown below. The edges of the inside extrusions were rounded to reduce the stress concentrations.

Also one important thing to mention is that a standard tooth profile was created by the use of multiple curves joined together to ensure good operation.

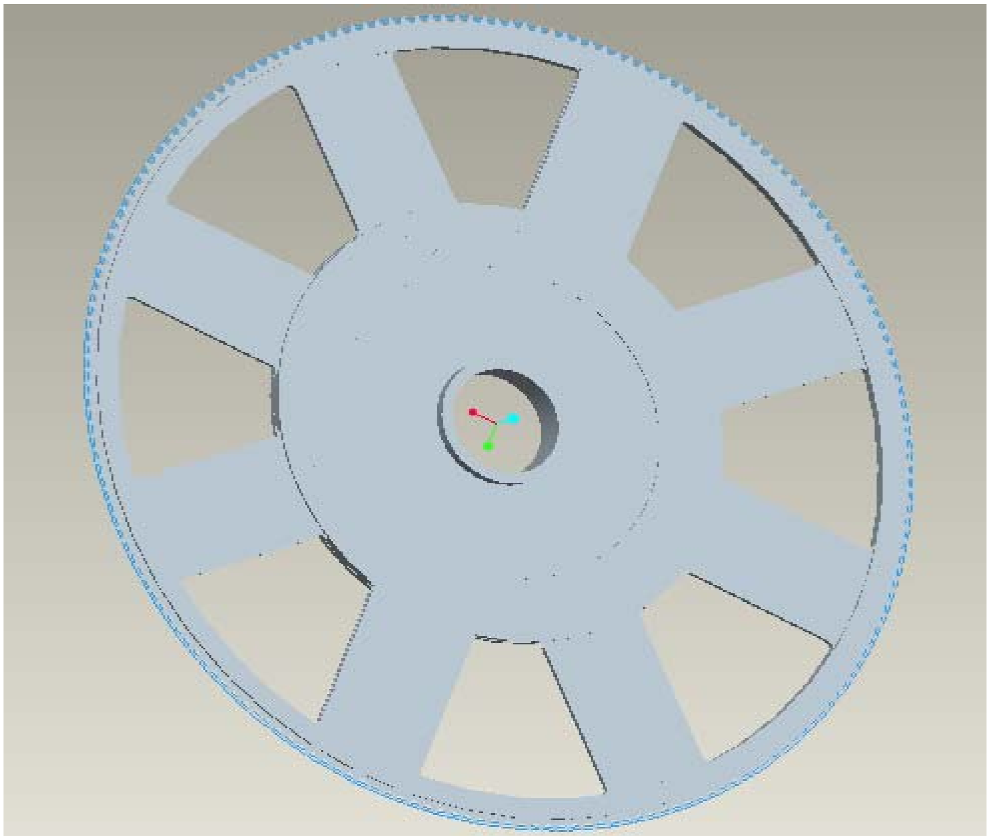

Figure 6

Calculations

The length of the chain can be calculated from the equation:

$$L = \frac{Z_1 + Z_2}{2} + \frac{2C}{P} + \frac{P\left(\frac{Z_1 + Z_2}{2\Pi}\right)^2}{C}$$

Where

C is the center to center distance of a two sprocket chain drive (set at 35mm)

Z_1 is the number of teeth of the driver sprocket (17)

Z_2 is the number of teeth of the driven sprocket (192)

P is the chain pitch in mm (6.3mm)

Substituting into the equation we get:

L=236 pitches

Now we recalculate the exact center distance from the equation:

$$C = \frac{P}{8}\left[\left(2L - Z_1 - Z_2\right) + \sqrt{\left(2L - Z_1 - Z_2\right)^2 - \left(\frac{\Pi}{3.88}\left(Z_2 - Z_1\right)^2\right)}\right]$$

Where L was calculated previously.

This equation takes into account the constraint that there should be a minimum of 6 teeth of engagement or 120 degrees as shown below.

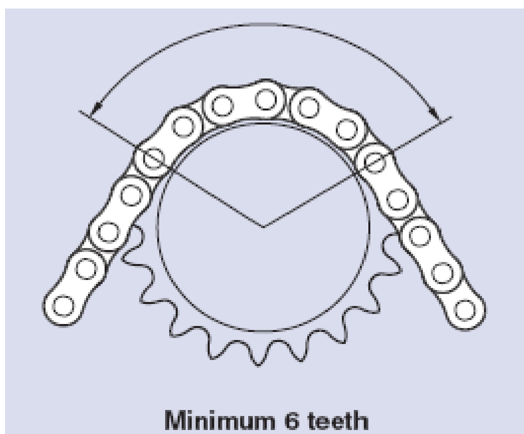

Figure 7

Substituting all the variables we get the exact center distance.

C = 373 mm

Diameter of smaller Sprocket:

To calculate the diameter of the smaller sprocket we use the following formulae:

$$\alpha = \frac{360°}{17} = 21.17 \text{ degrees/tooth}$$

$$\sin(\alpha/2) = \frac{P/2}{R} \rightarrow R = \frac{P/2}{\sin(\alpha/2)} = \frac{3.15}{\sin(10.588)} = 17.14mm$$

$$\rightarrow D = 34.3mm = 3.43cm$$

Gradability:

The driving strategy will be to accelerate to 40 km/hr followed by rolling to 20km/hr. The question that we still need to answer is: What will the rpm of the engine be at 40km/hr?

This will depend on the engine map of the engine after all the modification that will determine how high we are going to rev the engine.

Given this, the calculations below are based on having the highest gear ratio possible to be sure that the car will be able to climb the highest possible slope with a two sprocket configuration.

The gear ratio for the case of a 17 tooth sprocket:

$$R = \frac{Z_2}{Z_1} = \frac{192}{17} = 11.294$$

From the graph below we see that at 4000 rpm the GX35 engine delivers approximatly 1.5N.m of torque:

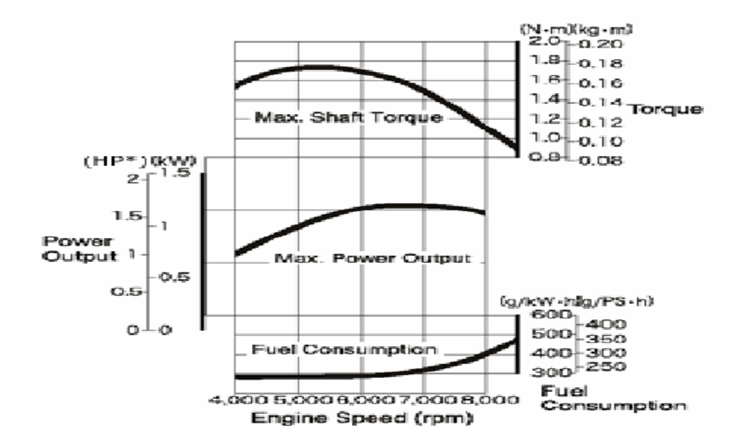

Figure 8

Thus:

$$\tau_{driver} = 1.5N.m \rightarrow \tau_{driven} = 1.5N.m \times 11.294 = 16.941N.m$$

$$\rightarrow F_{wheel} = \frac{\tau_{driven}}{R_{wheel}} = \frac{16.941}{0.2386} = 72.21N$$

Assumin 94% efficiency for the drive:

$$\rightarrow F_{wheel} = 72.21N \times 0.94 = 67.9N$$

From the two figures below we can see that the maximum slope is between points A15 and A16 thus:

$$\frac{\Delta_y}{\Delta_x} = \frac{(97.42 - 95.48)}{(1439.7 - 1387.1)} = 0.03688 = 3.688\%$$

Thus we will asume we have 4% slope

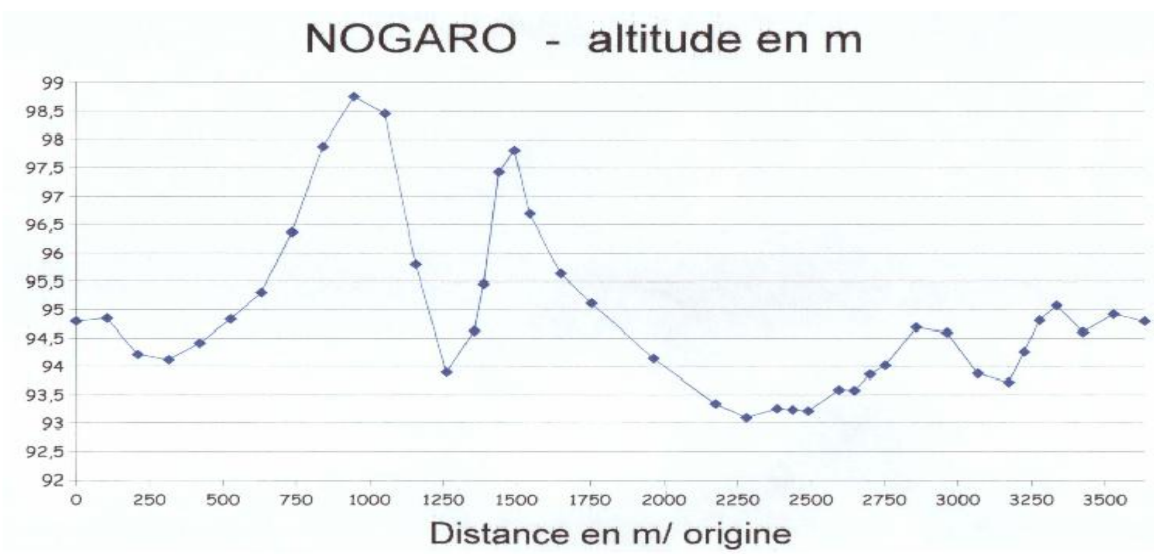

Figure 9

TIM

Profil établi le 6/02/2000 par l'équipe TIM UPS-INSA de TOULOUSE
L'origine est prise à la sortie des stands

Point Ai	Longeur (m)	Distance (m)	Altitude (m)
A1	0,0	0,0	94,80
A2	105,1	105,1	94,85
A3	105,1	210,2	94,21
A4	105,1	315,3	94,11
A5	105,1	420,3	94,41
A6	105,1	525,4	94,84
A7	105,1	630,5	95,31
A8	105,1	735,6	96,37
A9	105,1	840,7	97,88
A10	105,1	945,8	98,76
A11	105,1	1050,9	98,46
A12	105,1	1156,0	95,80
A13	105,1	1261,0	93,90
A14	94,6	1355,6	94,63
A15	31,5	1387,1	95,46
A16	52,5	1439,7	97,42
A17	52,5	1492,2	97,81
A18	52,5	1544,8	96,69
A19	105,1	1649,9	95,64
A20	105,1	1755,0	95,12
A21	210,2	1965,1	94,14
A22	210,2	2175,3	93,34
A23	105,1	2280,4	93,10
A24	105,1	2385,5	93,25
A25	52,5	2438,0	93,23
A26	52,5	2490,6	93,21
A27	105,1	2595,6	93,59
A28	52,5	2648,2	93,58
A29	52,5	2700,7	93,88
A30	52,5	2753,3	94,02
A31	105,1	2858,4	94,60
A32	105,1	2963,5	94,60
A33	105,1	3068,5	93,87
A34	105,1	3173,6	93,72
A35	52,5	3226,2	94,26
A36	52,5	3278,7	94,81
A37	57,8	3336,5	95,08
A38	89,3	3425,8	94,61
A39	105,1	3530,9	94,93
A40	105,1	3636,0	94,80
Total:	3636,0		

Figure 10

The horizental component of the weight represents the force that we will need at the wheel to climb the hill and is equal to:

$$W_x = mg\sin\alpha$$

$$\alpha = \tan^{-1} 0.04 = 2.29\%$$

Assuming that the mas of the car is 90Kg

$$W_x = 90 \times 9.81 \times \sin 2.29° = 35.28N$$

Thus the load repreasents only 52% of the produced force. This is without taking into account all the losses in the bearings. But since we have almost ½ of the power unused i assume it will be enough.

Now at 4000 rpm we have:

$$\omega_{driver} = 4000rpm \times 1\min/60\sec \times 2\Pi rad/round = 418.87 rad/\sec$$
$$\to \omega_{driven} = 418.87/11.294 = 37 rad/\sec$$
$$\to V = \omega_{driven} \times R_{wheel} = 37 \times 0.2386 = 8.85 m/\sec = 32 Km/hr$$

This hardest incline we concidered comes after a downhill as we see from the altitude graph. Thus if the driver starts to accelerate at 32km/hr or more then we will be able to climb the hill.

A similar calculation shows that if the driver lets the car decelerate to 20Km/hr then the force produced at the wheel will ammount to 36N and thus with all the bearing losses the car wouldnt be able to climb the hill. This is due to the fact that the engine delivers very little ammounts of torque at low engine speeds.

This is a worrying consequence but the fact is that these values are for the unmodified engine and modifications that will be performed will improve its torque charachtaristics.

Another possible solution will be to use a two step transmission which is very unfavorable because the losses in the transmission will double, added to that is the significant increase in weight.