Design of a test bench for measuring oil churning losses

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Traineeship report

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Abstract

At the École Catholique d'Arts et Métiers (ECAM) in Lyon they are interested in oil churning losses. This research is preformed together with PSA (Peugeot and Citroën) and TOTAL FINA ELF. Oil churning losses occur in gearboxes, especially where gears rotate in oil. Hereto a test bench is available in the laboratory of ECAM. However the existing test bench has little resemblance with a real gearbox, the distance between the gears and the walls is to big. This report is about the design of a new test bench, especially about the design of a new oil housing and the implementation of a rotary torque sensor to measure the oil churning losses more accurate.

Contents

Introduction

In the laboratory of the École Catholique d'Arts et Métiers (ECAM) in Lyon, there is a test bench to measure oil churning losses. Churning losses are caused by gear striking, pumping or otherwise moving lubricant around in a gearbox. ECAM, PSA (Peugeot and Citroën) and TOTAL FINA ELF are interested in these oil churning losses.

Because there are a lot of disadvantages which occur at the current test bench, ECAM wants to realize a new test bench. The disadvantages which occur now are, the principal of measuring the oil churning losses is delicate, the sealing is ineffective, changing the gears takes a lot of time and it has a long start-up time. The main goal of this test bench is to simulate the behavior of oil in a gearbox and the influence of different oils. But the resemblances with a gearbox are small.

The goal of this traineeship is to design a test bench which can measure oil churning losses and also is able to investigate the influence of different distances between the gears and its surrounding walls. It is also desired to measure the oil churning losses more accurate, this will be realized by using other measuring techniques.

First the principal of the test bench and the disadvantages of the existing test bench will be discussed in chapter 1. In chapter 2 the concept for the new design and the different parts will be further investigated. At the end the conclusions and some recommendations for future work are given.

Chapter 1

Test bench for measuring oil churning losses

To measure oil churning losses a test bench is available in the laboratory of ECAM. First the principal of oil churning losses will be discussed, after this the test bench is investigated. Also a short explanation will be given about how the measurements are done. Though there are some disadvantages with this test bench, these will be inventoried.

1.1 Oil churning losses

In gearboxes heat is generated, caused by load and no-load dependent power losses. Load dependent losses consist of tooth friction and bearing losses. No-load dependent losses consist of oil churning, windage and shaft sealing losses. Oil churning losses occur in gearboxes and other parts of vehicle power trains where gears rotate and move in oil. For PSA and TOTAL FINA it is interesting to know what the influence is of different geometries and oils in gearboxes and how this influences the oil churning losses. Especially the influence of the distance between the walls and the gears.
From previous studies there are established two formulas for oil churning losses, one for low speeds and one for high speeds. For connecting the two formulas, a Reynolds number is formulated. With the test bench these formulas will be verified and where necessary adapted. For the new test bench, the goal is to verify whether these formulas are still applicable when the gears are surrounded with walls, like in real gearboxes. After research at ECAM [3], the drag torque acting on a gear can be estimated with formula 1.1.

$$C_{churn} = \frac{1}{2} \cdot \rho \cdot \left(\frac{\pi \cdot N}{30}\right)^2 \cdot S_m \cdot R_p^3 \cdot C_m \tag{1.1}$$

For calculating the power losses concerning oil churning, formula 1.2 holds.

$$P_{churn} = C_{churn} \cdot \Omega \tag{1.2}$$

The two regimes considered are, for low speeds from 1000 to 3000 rpm and for high speeds from 5000 to 7000 rpm. As result for the low speed regime there is a formula for the dimensionless drag torque coefficient given in formula 1.3, [2].

$$C_m = \left(\frac{h}{R_p}\right)^{0,45} \cdot \left(\frac{V_0}{D_p}\right)^{0,1} \cdot Fr^{-0,6} \cdot Re^{-0,21} \tag{1.3}$$

For the high speed regime formula 1.4 holds for the dimensionless drag torque coefficient, [1].

$$C_m = 3,4 \cdot \left(\frac{h}{R_p}\right)^{0,1} \cdot \left(\frac{V_0}{D_p^3}\right)^{-0,35} \cdot Fr^{-0,88} \cdot \left(\frac{b}{D_p}\right)^{0,85} \tag{1.4}$$

The switching point for formula 1.3 and formula 1.4 is defined with formula 1.5.

$$Re^x = \frac{v_t \cdot b}{\nu} \tag{1.5}$$

The value for which this Reynolds number will switch from the low speed formula to the high speed formula is 6000, [1]. In this report these formulas will not be further investigated. However, they can be used to investigate the differences between the old and the new test bench.

1.2 Existing test bench

The test bench which is available in the laboratory of ECAM can be seen in figure 1.1. In this test bench one or two gears can be rotated in the oil housing. The two gears are mounted on axles, the primary axle is driven by a DC-motor and the secondary axle is driven through the gears. The connection between primary axle and motor is realized with two pulleys and a belt. To measure the churning losses, two measurements are done one with oil in the oil housing and one without oil. First with oil in the housing, the temperature of the oil raises while the gears turn, because the gears turn with a constant speed the temperature will reach a constant value. When the temperature of the oil is constant, the power delivered by the motor is measured. Then the oil housing is emptied and the power delivered by the engine is measured again. Because the gears are not removed, windage losses occur. At high speeds these windage losses have to be taken into account, at low speeds these are negligible. These are calculated with formula 1.6.

$$P_{wind} = a \cdot N^b + c \tag{1.6}$$

With a, b and c depending on the geometry of the gear. For the resulting churning losses, formula 1.7 holds.

$$P_{churn} = P_{motor,\ oil} - (P_{motor,\ no\ oil} - P_{wind}) \tag{1.7}$$

Basically this is the method which is used to calculate the oil churning losses.

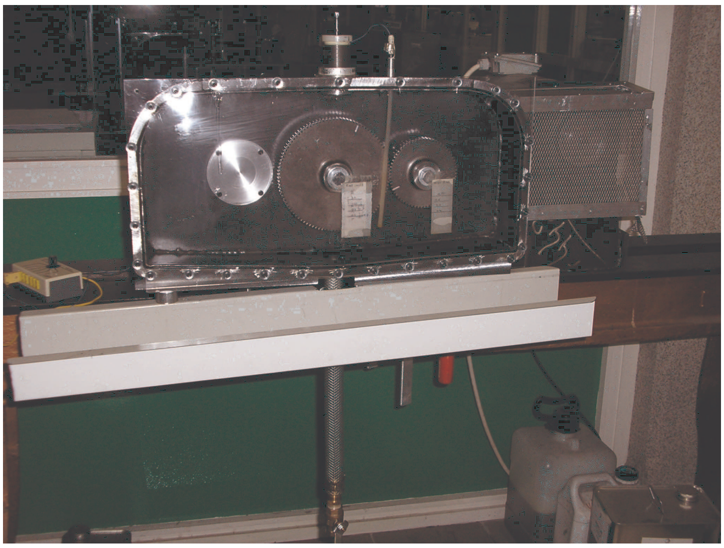


Figure 1.1: Photo of the existing test bench

1.3 Disadvantages

As already mentioned, there are some disadvantages which occur at the test bench. Now an electric power loss is measured and with this, the mechanical power loss is calculated. When calculating the power losses, two big values are subtracted from each other which can give a big deviation. Also the window of the housing does not close very well, this is fixed to the housing with a lot of bolts. Under operating conditions the temperature of the oil can get quite high($\pm\ 80\ ^oC$). Due to these high temperatures the window deforms easy because it is made of thin plexiglass ($\pm\ 5\ mm$) and the oil leaks between the window and the housing. Another problem is changing the gears, first the window has to be removed. To remove the window, all the bolts have to be unscrewed, this takes a lot of time. The goal now is to make it simpler and faster to open the window, and with better sealing. Also the measurements take long because first the oil has to be heated to a constant temperature and when the temperature does not change for more then $0,1\ ^oC$ in 10 minutes, it is possible to make a measurement. It is hard to solve this problem, because the oil has to have time to heat. But it should be considered, so the dimensions must stay small. Besides keeping the dimensions small, a heating should be considered to heat the oil and the housing. With this the problem arises the end temperature is not known, however with the measurements from the old test bench, a good estimate of the temperature can be made.

Chapter 2

New test bench design

For the new test bench, the disadvantages of the existing test bench should be considered. First the specifications for the new test bench will be inventoried, with this the design of the new test bench can be made.

2.1 Specifications

From the disadvantages in chapter 1.3 a list of specifications for the new test bench can be made. In table 2.1 these can be seen. To rotate the gears and to control the velocity of the gears, a frequency converter is available. To reach the specified velocity, the right ratio for the transmission has to be calculated, this will be done in the chapter 2.2.2. To measure the power losses due to oil churning, there is chosen to measure the torque through the primary axle with a rotary torque sensor. This to measure as close as possible to the gears, however this creates certain boundaries. The torque sensor is very sensitive, hereto a torque limiter is added to make sure the torque sensor is not overloaded.

Table 2.1: Specifications

	Specification	Characteristics
1	Rotate gear(s)	Both directions
2	Ability to control the velocity of the gear	500 to 7500 rpm
3	Indicate the velocity of the axle	
4	Measure the torque through the primary axle	
5	Reduce the duration of the test	Heat the oil, in around 20 minutes
6	View how the oil flows	Window for visualization
7	Adjust the oil level	Measure the oil level with an integrated ruler
8	Fill the reservoir with oil	Around 3 l., smaller if possible
9	Empty the oil reservoir	Within 10 s.
10	Change the gears	Use the current sets of gears. Easy access (simple opening and closing of the window)
11	Positioning of the inserts	Inserts, removable parts, resemblance gearbox. Easy access (simple opening and closing of the window)
12	Measure the temperature (oil and ambient)	10 to 150 ^{o}C, for the oil
13	Guarantee the seals	
14	Guarantee the safety	
15	Catch the oil in case of leakage	

Secondly, flexible couplers are introduced to bear the possible misalignment with the torque sensor. And at last there is also an axle with the same dimensions as the torque sensor, so in the beginning the torque sensor can be replaced with this axle to protect it from breaking. With this the couple limiter can also be tested. Another possibility for this axle is to begin the test, when the oil is at a constant temperature the axle can be replaced with the torque sensor. This to regard the lifetime of the torque sensor. The torque sensor can also be used to measure the velocity of the primary axle. The speed of the gears is chosen in between 500 to 7500 rpm, because this is a reasonable speed to measure the oil churning losses. When the speed is higher there are other techniques for making sure there is oil in between the gears. To reduce the duration of the test a possibility is to heat the oil or the housing with the oil. The rest of the specifications will be discussed when the concerning parts are investigated.

2.2 Realization of the new test bench

Before the beginning of this traineeship, a first design of the primary axle was made, with the torque sensor and its necessary parts. Because of this design, several parts were already bought which can be seen in table 2.2. In this design there were also made some choices with respect to other parts, like

Table 2.2: Already bought parts

Name	Brand	Part number
Couple limiter	R+W	SK1 /2/F/10H7/2/0.5-2.2Nm
Flexible couplers	HUCO	460.26.2228 & 460.26.2828
Frequency converter	Danfoss	VLT 2815
Motor	BBC Compax	M EUC 90 S2, n^o 1999945
Torque sensor	FGP instrumentation	CD1140-1A

bearings and snap rings, this can be seen in figure 2.1. This design is made in Solidworks©, based on the design of Mr. Ricol which was made on paper. This design will be explained later on. First there will be given some more information about the global layout.

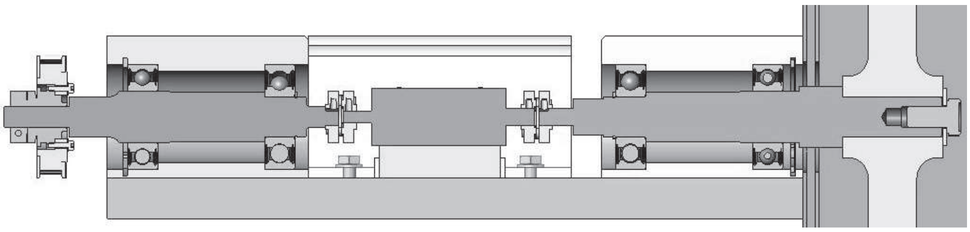
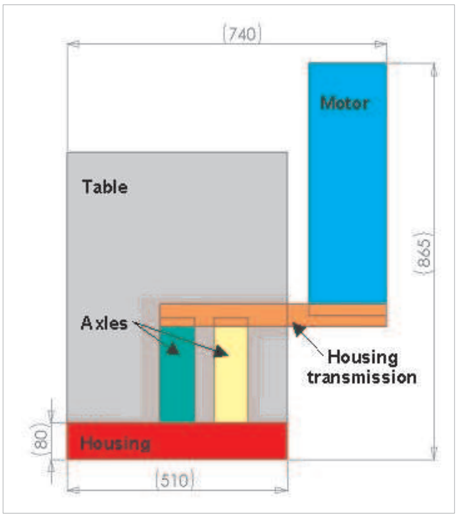


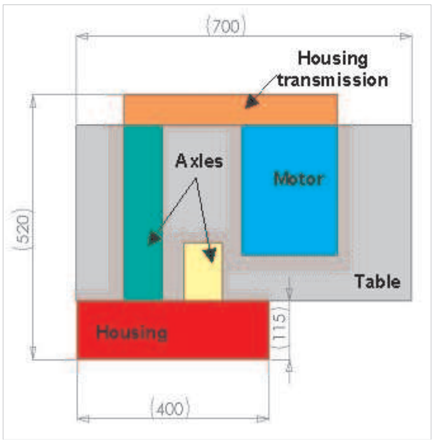
Figure 2.1: Design of primary axle

2.2.1 Layout

The layout of the new test bench should be small, this to keep the dimensions within proportion. A schematic top view of the old test bench can be seen in figure 2.2(a). As we can see, the motor is long and makes the total size of the design large. Also the major part of the table is not used. For the new design there is chosen to place the motor on the same side as the axles. This is possible because the new motor is much shorter. Although this would make the design much shorter, the primary axle in the new design is much longer. This because a torque sensor and flexible couplers are integrated. The secondary axle however, is shorter as the axles from the old test bench. The housing differs also qua dimension, but this is not the biggest difference. The housing of the old test bench rests on the table while the housing of the new test bench is connected to the side of the table. The housing is also



(a) Top view of the existing test bench



(b) Top view of the new test bench

Figure 2.2: Top view of the different layout

fixated to the bearing housings of both the axles. A schematic top view of the new design can be seen in figure 2.2(b).

2.2.2 Transmission

The transmission is important to determine the minimum and maximum velocity for the test bench. Here the calculations are given to determine the dimensions of the transmission. For the velocity of the gears, the following region is specified. The lowest velocity should be 500 rpm and the maximum speed should be around 7500 rpm. The motor is able to cover a region from 0 to 2865 rpm, to reach 7500 rpm the following ratio is needed:

$$r = \frac{7500}{2865} = 2,618 \tag{2.1}$$

To reach this ratio there are various standard sprocket combinations available. First the selection procedure of Gates® will be followed, [4].

Step 1: Determine design horsepower

The design horsepower is determined by multiplying the rated horsepower (usually the nameplate rating) by the service factor. The service factor of the drive train depends on average hours per day of service and the relative severity of the drive train. In this case a service factor of 1, 6 is determined. To calculate the rated horsepower formula 2.2 is used

$$RatedHP = \frac{P}{736} = \frac{1500}{736} = 2,04\ HP \tag{2.2}$$

To calculate the design horsepower formula 2.3 is used.

$$DesignHP = RatedHP \cdot Servicefactor = 2,04 \cdot 1,6 = 3,26\ HP \tag{2.3}$$

Step 2: Select belt pitch

With the belt pitch selection guide graphs (page 12 - [4]) and with the $DesignHP$ and the velocity of the faster shaft, the belt pitch can be chosen. With the $DesignHP$ being 3, 26 HP and the velocity of the faster shaft at 7500 rpm, 5 mm satisfies for the PowerGrip® GT®2 belt pitch.

Step 3: Select sprockets and belt length

a Determine speed ratio:

$$r = \frac{\Omega_{fast}}{\Omega_{slow}} = \frac{7500}{2865} = 2,618 \tag{2.4}$$

b Select sprocket combination and belt length:

For the sprocket at the motor side a large sprocket is desired, however not to big because this will result in high inertia forces. And for the sprocket at the side of the primary axle, it should be relatively big because the connection with the torque limiter should be made here. In table 2.3 an overview of the possible combinations can be seen. There is chosen for the combination 90/36, this in consultation with the commercial agent of Gates®. This combination results in a ratio of $2,5$, which is not the desired ratio. However a reasonable speed can be reached, to know $\omega = 2865 \cdot 2,5 = 7162,5\ rpm$. This maximum speed is sufficient to research the influence of different distances between the gears and the walls. For the belt length there has to be taken into account that the motor should not touch the secondary axle. Hereto the center distance should be $300\ mm$ minimally, this result in a belt length of 950 teeth, which has a center distance of $314,56\ mm$.

Table 2.3: Possible sprocket combinations

Teeth S*	Diam. S [inch]	Teeth L**	Diam, L [inch]	Ratio	Diam. S [mm]	Diam. L [mm]
18	1,128	45	2,820	2,500	28,651	71,628
20	1,253	50	3,133	2,500	31,826	79,578
24	1,504	60	3,760	2,500	38,202	95,504
32	2,005	80	5,013	2,500	50,927	127,330
36	2,256	90	5,639	2,500	57,302	143,231
19	1,191	48	3,008	2,526	30,251	76,403
22	1,379	56	3,509	2,545	35,027	89,129
44	2,757	112	7,018	2,545	70,028	178,257
25	1,566	64	4,010	2,560	39,776	101,854
28	1,754	72	4,511	2,571	44,552	114,579
20	1,253	52	3,258	2,600	31,826	82,753
23	1,441	60	3,760	2,609	36,601	95,504
26	1,629	68	4,261	2,615	41,377	108,229
19	1,191	50	3,133	2,632	30,251	79,578
34	2,130	90	5,639	2,647	54,102	143,231
18	1,128	48	3,008	2,667	28,651	76,403
21	1,316	56	3,509	2,667	33,426	89,129
24	1,504	64	4,010	2,667	38,202	101,854
30	1,880	80	5,013	2,667	47,752	127,330

* : Small sprocket
**: Large sprocket

c Check belt speed:

Do not exceed 6500 fpm with stock sprockets.

$$v = \frac{PD \cdot N}{3,82} = \frac{2,256 \cdot 7162,5}{3,82} = 4230\ fpm \tag{2.5}$$

This value is within the limit so the combination of sprockets and belt-length is possible.

Step 4: Select belt width
The possible belt-widths start at $9\ mm$, this is sufficient for this power train, because the base rated horse power is between $5,71$ and $6,88\ HP$. This value should be multiplied with the length correction factor, i.e. here $1,07$, still this value is much higher then the required $3,26\ HP$. Gates® also recommends to observe the following rules:

1. Larger sprockets mean less belt width.

2. Larger sprockets yield extra long service life.

3. Avoid drives where the belt width (i.e. $9\ mm$) exceeds sprocket diameter (i.e. $57,302\ mm$).

4. Avoid drives where center distance (i.e. $314,6\ mm$) is greater than eight times the diameter of the smaller sprocket (i.e. $8 \cdot 57,302 = 458,416$).

All these rules are respected in the design. The final specifications of the power train are now:

- Small sprocket: 36 teeth, diameter of $57,302\ mm$

- Larger sprocket: 90 teeth, diameter of $143,231\ mm$

- Belt: the width is $9\ mm$, and it has a length of 950 teeth which results with the above sprockets in a center distance of $314,6\ mm$

2.2.3 Intermediate part

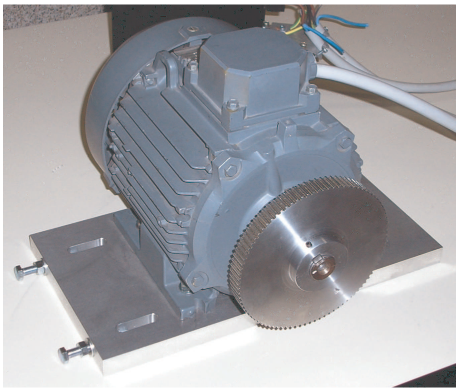


Figure 2.3: Motor on intermediate part

With the power train defined above, the belt has to be tensioned. There is chosen to realize this with a intermediate part, this is placed between the table and motor. The motor is fixated to this intermediate part and the intermediate part is fixated to the table. In figure 2.3 the motor on the intermediate part can be seen. The two bolts on the left side are to regulate the tension of the belt. This part is made of aluminium and is symmetric, so if there is any wear the part can be rotated. This makes it possible to create new tapped holes for the bolts which are present for regulating the tension in the belt. If the part has to be replaced, this is not complicated because of its simple design and easy to make. The only withdrawal is the placement of the holes, this is delicate because if they are not good aligned the belt will be misaligned. Which will cause noise and belt wear, which is not desired. The exact measurements can be seen in appendix C.1.

2.2.4 Transmission housing

Now the dimensions for the transmission are known, the housing can be designed. The transmission housing is to protect the people working on the test bench. It also protects against belt pieces flying around should the belt break. To check whether the belt runs smoothly there is chosen to make one side of the housing transparent, this is realized with plexiglass. Another point is, there has to be easy access to the torque limiter. Because when the torque through the limiter is to high, it releases and no torque is transmitted. To engage the torque limiter, a ring has to be pushed which is a part of the limiter. This is done by making the housing out of two parts, one piece to protect the major part of the belt and the pulley at the side of the motor. The other part is smaller and only covers the pulley at the side of the primary axle. This part can be removed by loosening two bolts, after it is removed the torque limiter can be re-engaged. This can be seen in figure 2.4. The dimensions of the housing are not very strict, however some considerations should be taken into account.

- The distance between the motor and the primary axle can be varied to tension the belt, so the distance between housing and the pulley at the motor side should at least be $30\ mm$
- Possibly the belt vibrates at high speeds, so there should be enough distance between housing and belt

The exact measurements can be seen in appendix C.2.

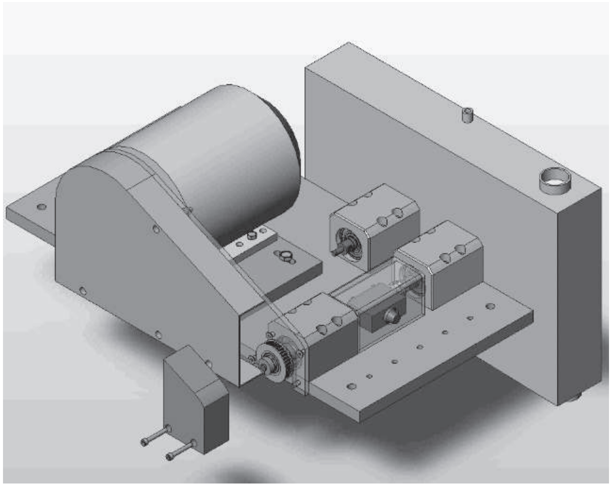


Figure 2.4: Disassembly of the housing to re-engage the torque limiter

2.2.5 Primary axle

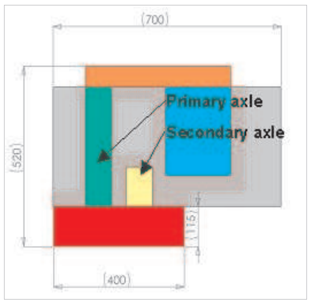


Figure 2.5: Axle definition

For the axles the design of Mr. Ricol is used, although there are made some minor adjustments. In figure 2.5 the definition for primary and secondary axle can be found. First the primary axle will be further discussed.
The primary axle is the axle which is driven by the motor. This axle consists of three parts; the axle at the pulley side, the torque sensor and the axle at the housing side. This can also be seen in figure 2.6. When beginning at the left first the axle at the pulley side, this consists of, the pulley, the torque limiter, a retaining ring, a bearing, the shaft, the bearing housing, a bearing and a retaining ring. Secondly the part with the torque sensor, which consists of, a flexible coupler, the torque sensor and an other flexible coupler. And at the right, the axle at the housing side, which consists of, a retaining ring, a bearing, the bearing housing, the shaft, a bearing, a retaining ring and the gear. The shafts are positioned in bearing housings, because there is chosen to base the two axles on the same design, the bearing housings are also based on the same design. The biggest differences between the shaft at the pulley side and the housing side are the connections with the pulley/torque limiter and the gear. The dimensions of these connections differ a lot but the rest of the shafts are the same. The exact measurements can be seen in appendix C.3.

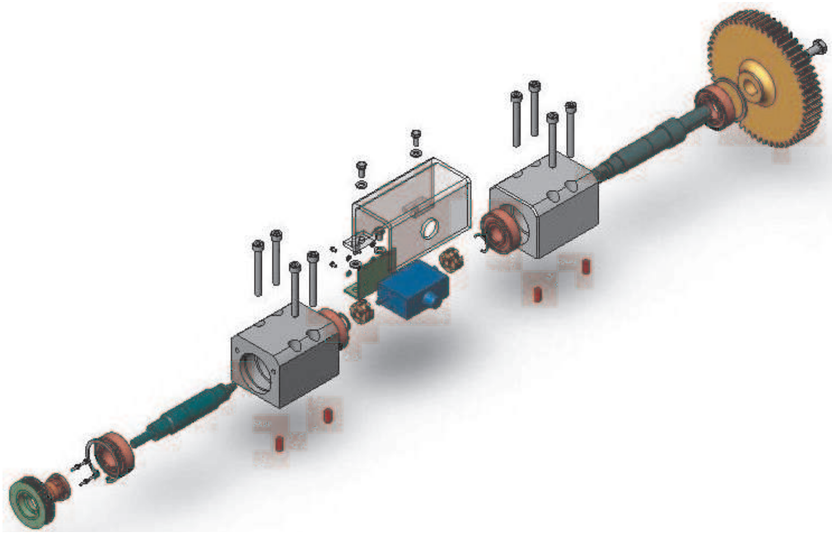


Figure 2.6: Exploded view of the primary axle

2.2.6 Secondary axle

The secondary axle is based on the design of the primary axle. The part of the primary axle at the side of the oil housing is completely copied. However this is not necessary, to have a connection for the torque sensor on the axle, this part is copied to make it possible to exchange the axles with each other in case of wear. In figure 2.7 an exploded view of the secondary axle can be seen. Here we can see from left to right, the gear, a retaining ring, a bearing, the axle, the bearing housing, the other bearing and the other retaining ring. The exact measurements can be seen in appendix C.4.

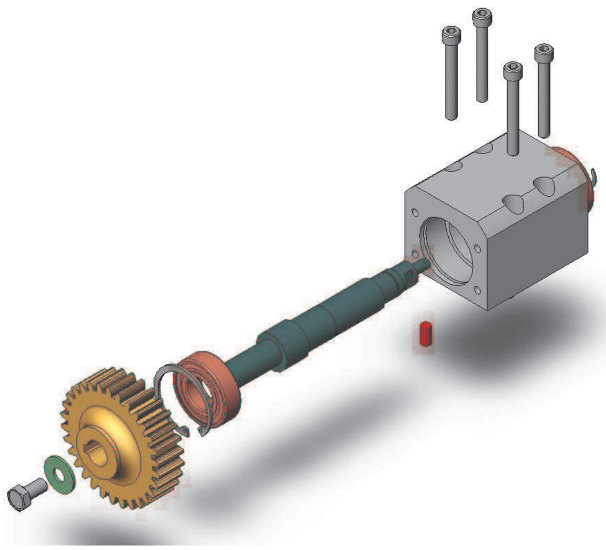


Figure 2.7: Exploded view of the secondary axle

2.2.7 Table

Now the major parts for the test bench are designed the layout for the table can be made. The table is to position the bearing housings, the intermediate part with the motor, the transmission housing and the oil housing. The table is supported by four legs, these are present in the room where the test bench is going to be realized in the laboratory of ECAM. The positions of these legs are flexible, within certain boundaries. However the tolerances for the connections with the legs are low, other tolerances are high. To know, the positions of the holes for connecting the bearing housings are delicate, when they are not accurate the alignment can not be guaranteed. At first the bearing housings were designed to be connected to the table with only four bolts, but with this there rests an uncertainty because of the play between bolts and holes. Here for there is chosen to make small cylinders ($\varnothing$ $8\ mm$) with high tolerance, which can be placed in holes in the table which also have high tolerance. And the bearing housings have the same high tolerance holes, so when all is assembled the bearing housings are well positioned with respect to the each other. For future use there are designed some extra holes, this to make it possible to add features like microphones or other measuring devices. These extra holes are positioned with the same accuracy as the positioning for the bearing housings. With this the table has the possibility to obtain extra functions, because it is difficult to make holes in the table when everything is assembled. In figure 2.8 the table can be seen with all the pieces which are connected to the table, except for the legs. The exact measurements can be seen in appendix C.5.

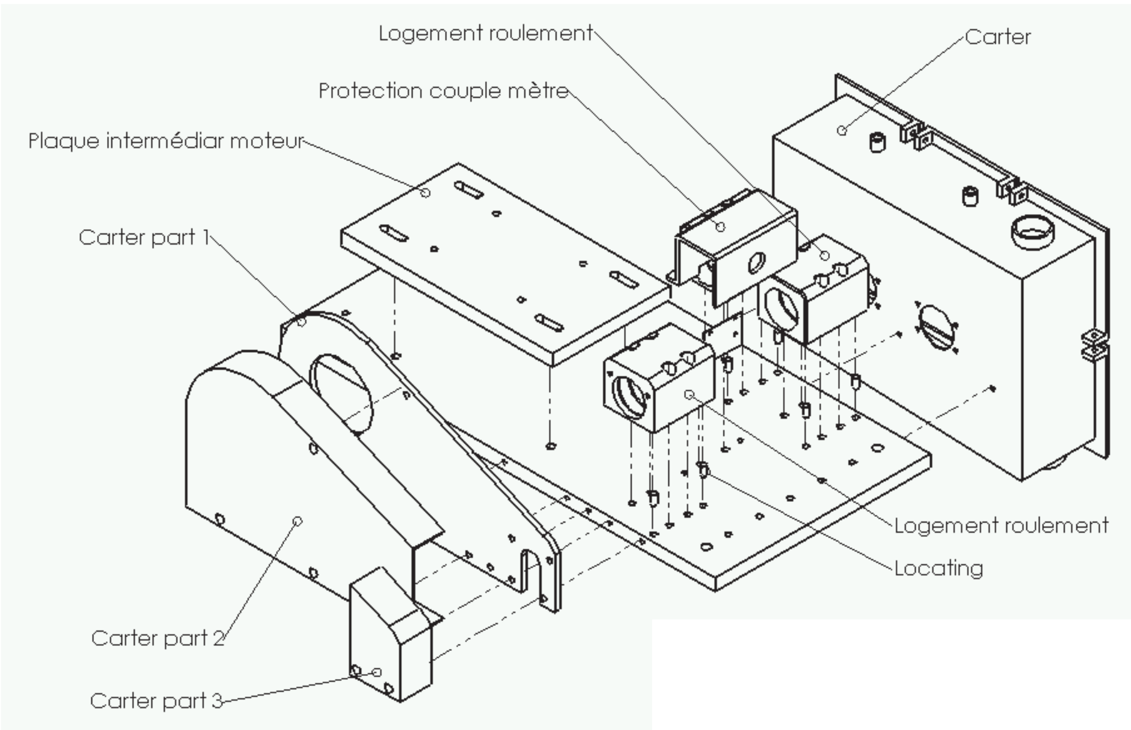


Figure 2.8: Table with all the parts connected to it

2.2.8 Oil housing

The part where this project all started about is the oil housing. Because the existing bench has a lot of disadvantages and there are new points of view with respect to the tests. The disadvantages of the existing test bench considering the oil housing are:

Table 2.4: Specifications of the gears

Geometry	1	2	3	4	5	6	7	8
Module $[mm]$	$1,5$	$1,5$	$3,0$	$3,0$	$5,0$	$5,0$	$5,0$	$5,0$
Outer diameter $[mm]$	96	153	95	164	110	159	135	135

- The gasket between the window and the housing is not sufficient because there is a lot of oil leakage
- Changing the gears takes a lot of time because first the front window has to be removed which is mounted with many bolts
- Doing a measurement takes a lot of time because the measurement can only be done when the temperature in the housing is constant, now this takes more then 4 hours
- The temperature is only measured at one point, it is interesting to measure at several points to investigate the heat-distribution
- There is little resemblance with a real gearbox because the distances between wall and gear are large

For the new oil housing the goal is to resolve these disadvantages. Because the existing housing is to small to insert parts to resemble a gearbox there is decided to design a new housing. To adjust these inserts it is important to have easy access into the housing, so it should be possible to open the housing by loosening a couple of bolts. Now there are so many bolts to guarantee the sealing, but this is a result of the thin plexiglass. For the new window it is necessary to have at least thicker plexiglass

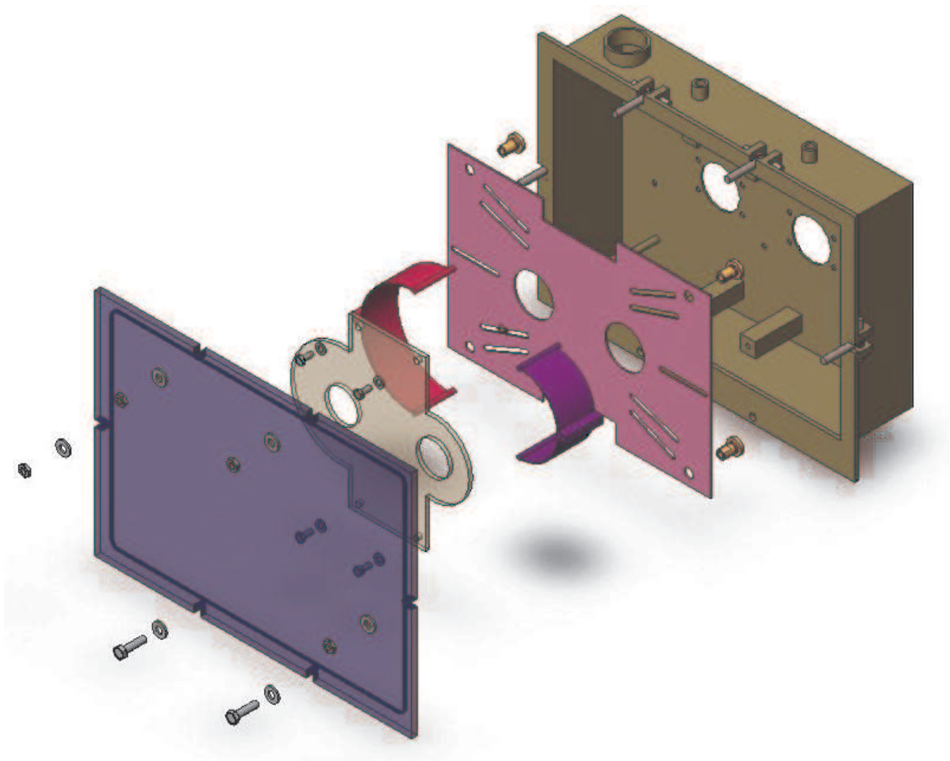


Figure 2.9: Exploded view of the oil housing

and better gaskets which can withstand the large temperature changes.
For the removable inserts it is necessary they are easy to adjust to investigate the influence of different distances between gears and walls. There are several distances which should be able to vary, to know the walls around the gears but also the distance of the front and back with respect to the gears. The specifications of the gears can be found in table 2.4. Geometry 1 and 2 are one pair, these can be run together because they have the same module. This holds also for geometry 3 and 4, geometry 5 and 6, and for geometry 7 and 8. The biggest differences in diameter between each pair of gears are around $40\ mm$ for the small gears and $29\ mm$ for the big gears. The inserts should have a minimum distance of $5\ mm$ to the gears and a maximum of $30\ mm$, because when this gets bigger it is assumed this will not influence the test results. In figure 2.9 the design of the oil housing can be seen. The part which closes the front of the gears is made of plexiglass this to maintain the view on the oil flow. With the plexiglass window it is also possible to make pictures or video recordings. The parts around the gears are made of thin steel, so they are flexible, this way they can be adjusted for each pair of gears without the need to replace these parts. They are mounted on the part which can regulate the distance of the gears with respect to the back. There is chosen to fixate the window into a steel frame, this so the plexiglass rests in the gaskets when the window is opened. The gasket between frame and housing is less sensitive for opening and closing because the gasket lies in between two steel frames. Also the plexiglass can be fixated with more bolts into the frame and the frame with less bolts to the housing, so there is easy access into the housing. The exact measurements can be seen in appendix C.6.

2.2.9 Assembly

Now the designs of all the different parts are realized, a view of the assembly can be made. The bearing housings are designed to make it possible to remove the axles at one side. For the housings at the oil housing this can be done via the oil housing. For the bearing housing at the side of the torque limiter the axle can be removed via this side. The torque sensor can be removed by only loosening the flexible couplers, the sensor can now be replaced by an axle to guard the lifetime of the torque sensor.

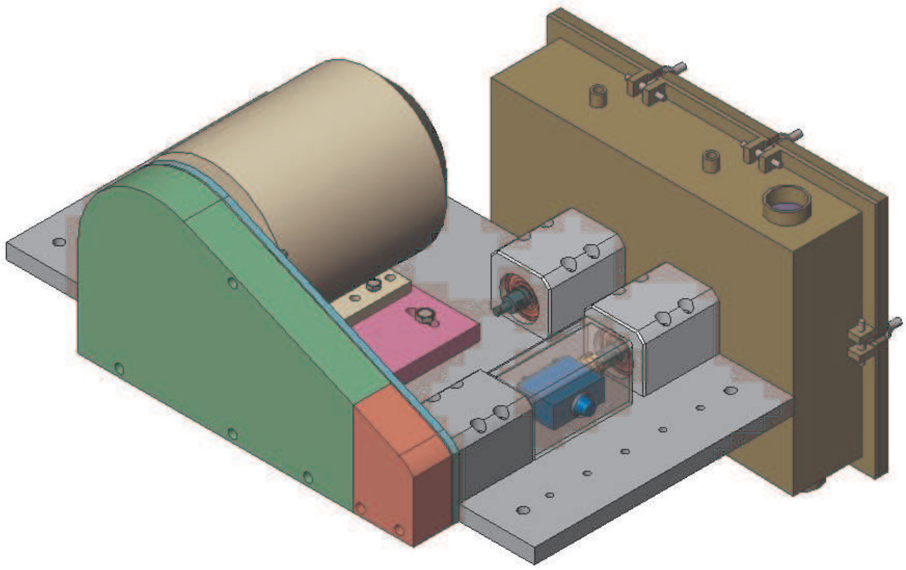


Figure 2.10: Assembly of the test bench for measuring oil churning losses

Chapter 3

Conclusions & Future work

The goal of this traineeship is to realize a test bench for measuring oil churning losses. First the old test bench was investigated and the disadvantages were inventoried. For the new test bench a design is made which should be an improvement of the old test bench. The remaining goal for this traineeship is to realize the test bench. The parts which can be bought, are bought and available in the laboratory of ECAM for assembly. The parts which are made by ECAM are in progress. The table, intermediate part, axles, bearing housings, axle to replace the torque sensor and the connection between torque limiter and pulley are finished. The parts which should be made now are the belt housing and the oil housing. For safety reasons the belt housing should be finished, but this is not necessary for operating the test bench so the first test runs can be made now. To make the first real tests the oil housing should be realized. But this is a complex part and the design should be further investigated, wether it is possible to make and it satisfies the needs. The idea for a heating is proposed but not further investigated with respect to the power and size needed, it is recommended this should be realized to shorten the duration of the tests.

For future work it is recommended to design a control strategy for the frequency converter. This so future users cannot break the test bench, especially the torque sensor. Hereto the acceleration and deceleration are important because of the mass-inertia of the gears, the torque through the torque sensor can get quite high at start-up and shut-down. A smooth acceleration and deceleration profile should be used. Further, a control strategy to automate the measurement. So when the oil temperature reaches a constant value, with certain bias, the measurement starts and is preformed for a certain time. With this, average values can be calculated and this should result in a more accurate results.

Appendix A

Symbols

The symbols used stand for:

Table A.1: Symbols

symbol	description	unit
ρ	oil density	$[kg/m^3]$
ν	viscosity at oil temperature	$[m^2/s]$
Ω	rotational speed	$[rad/s]$
C	drag torque	$[Nm]$
C_m	dimensionless drag torque coefficient	$[-]$
D_p	pitch diameter	$[m]$
$DesignHP$	design horsepower	$[HP]$
Fr	Froude number	$[-]$
N	rotational speed	$[tr/min]$
P	power	$[W]$
PD	pitch diameter	$[inches]$
R_p	pitch radius of gear	$[m]$
$RatedHP$	rated horsepower	$[HP]$
Re	Reynolds number	$[-]$
Re^x	new Reynolds number	$[-]$
S_m	wetted surface area of gear	$[m^2]$
V_0	oil volume in the carter	$[m^3]$
b	face width of gear	$[m]$
h	submerged depth of the gear	$[m]$
r	speed ratio	$[-]$
v	belt speed	$[fpm]$
v_t	rotational speed $(= \omega \cdot R_p)$	$[m/s]$

And the subscripts stand for:

Table A.2: Explanation of subscripts

symbol	description
$churn$	churning losses
$fast$	faster shaft
$slow$	slower shaft
$wind$	windage losses

Appendix B

Costs

Table B.1: Costs

Description	Company	Number	Quantity	Price
Torque sensor	FGP	CD1140-1A	1	€. 3.306,94
Torque limiter	R+W (VDP)	SK1/2/F/10H7/2/0.5-2.2Nm	1	€. 350,39
Flexible connector	HUCO	460.26.2228/460.26.2828	2	€. 72,45
Frequency converter	DANFOSS	VLT 2815	1	€. 383,92
Electric motor	BBC Compax	M EUC 90 S2	1	€. 0,00
Pulley (small)	GATES	5MR 36S	1	€. 13,31
Pulley (large)	GATES	5MR 90	1	€. 138,90
Belt	GATES	950 5MR 09	1	€. 11,75
Transport costs	GATES		1	€. 10,95
Gasket bearing	SKF	LSTO 25x52	3	€. 16,02
Snap ring	SKF	F3-04-20	4	€. 5,64
Snap ring	SKF	F3-04-52	4	€. 29,12
Bearing	SKF	6304-2Z	3	€. 3,47
Bearing	SKF	6205-2Z	3	€. 3,12
Transport costs	SKF		1	€. 9,45
Table		Atelier d'Apprentissage de Gorge de Loup	1	€. 613,00
Total				€. 4.968,43

Appendix C

Designs

C.1 Design of the intermediate part

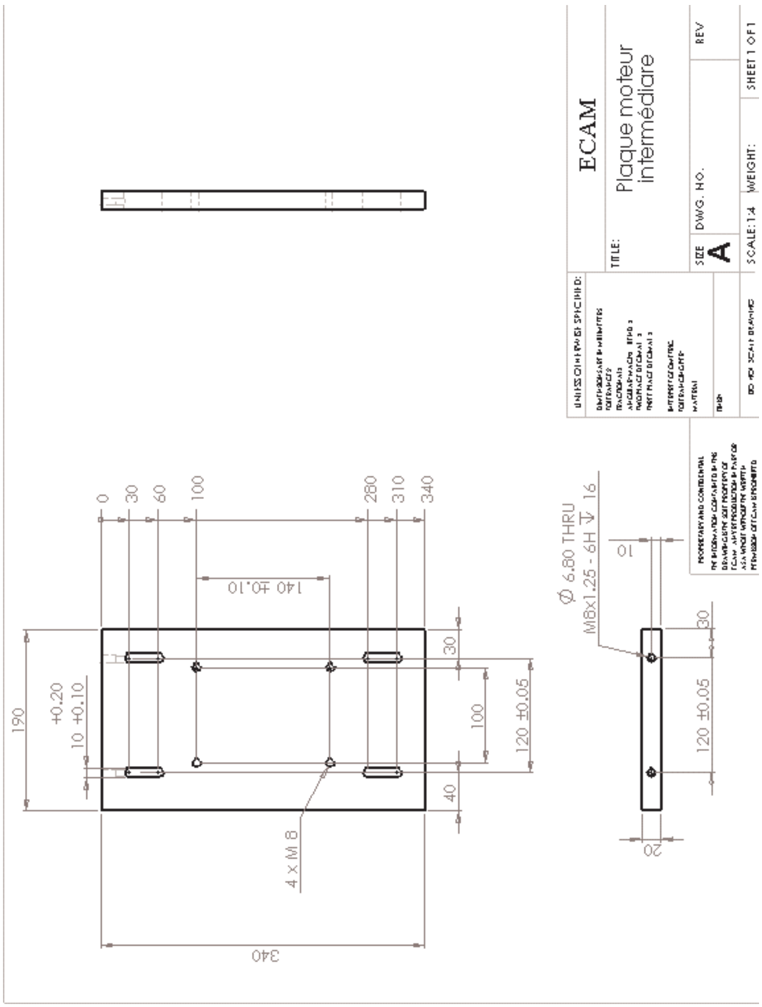


Figure C.1: Design of the intermediate part

C.2 Design of the belt housing

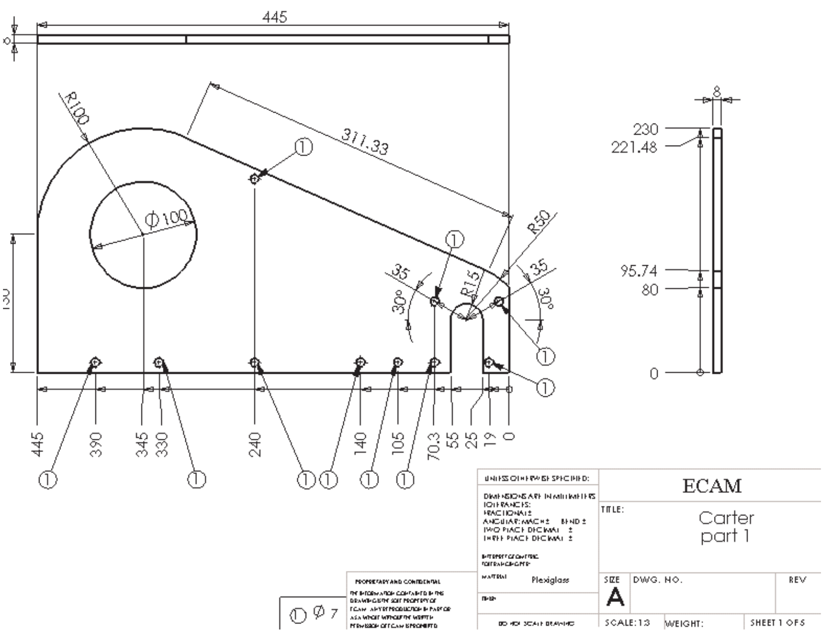


Figure C.2: Design of the belt housing, plexiglass part

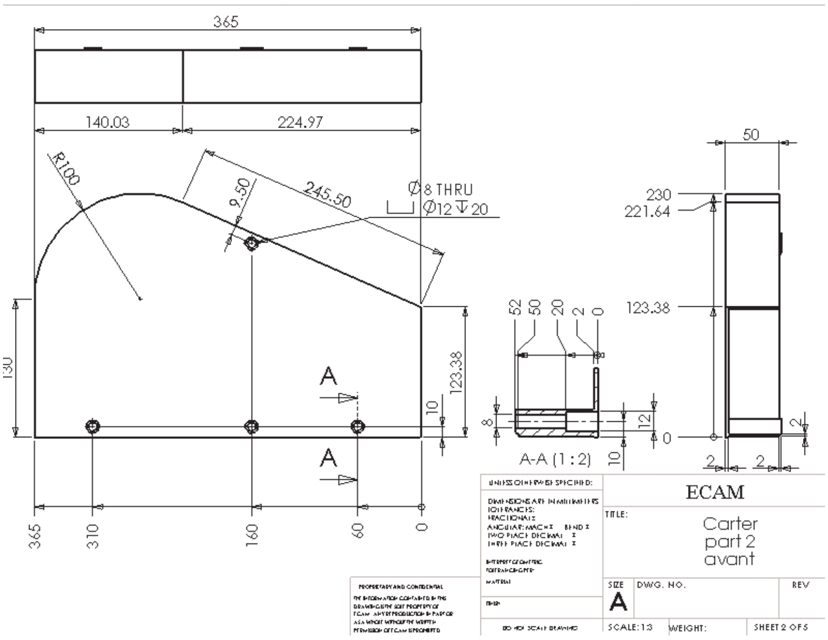


Figure C.3: Design of the belt housing, front of the big cover

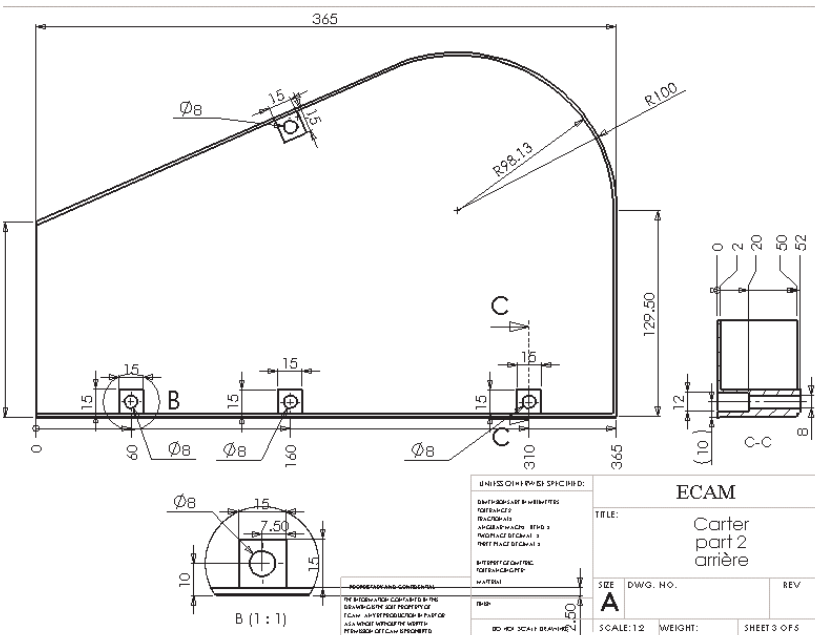


Figure C.4: Design of the belt housing, back of the big cover

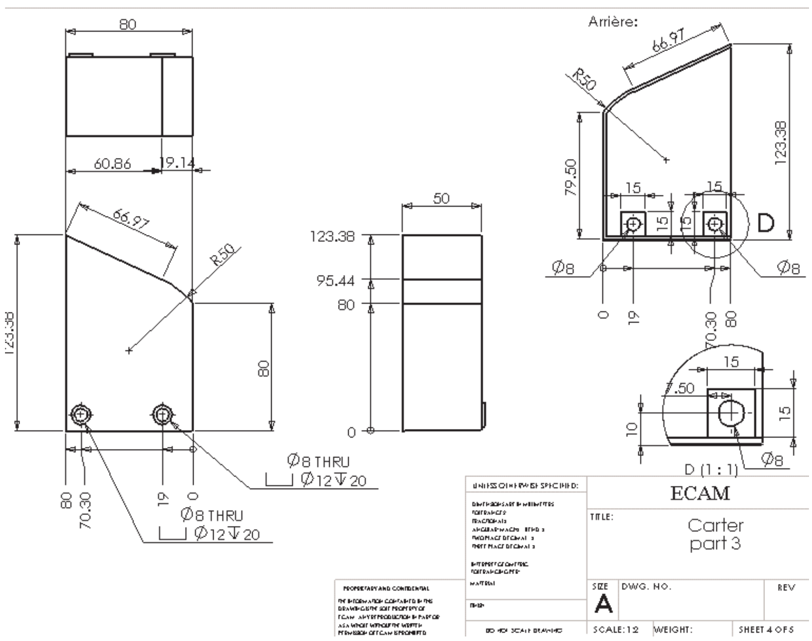


Figure C.5: Design of the belt housing, small cover

C.3 Design of the primary axle

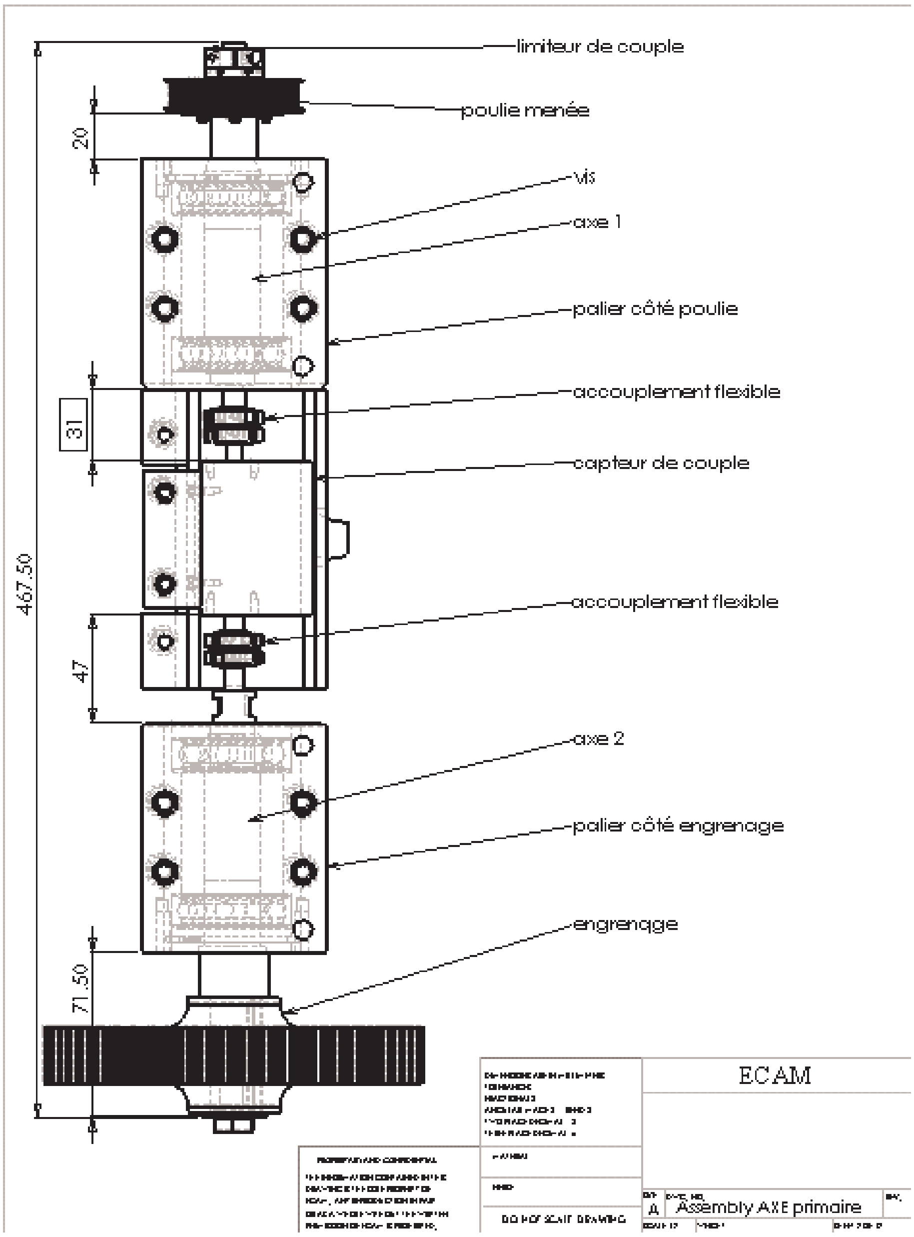


Figure C.6: Design of the primary axle, top view

C.4 Design of the secondary axle

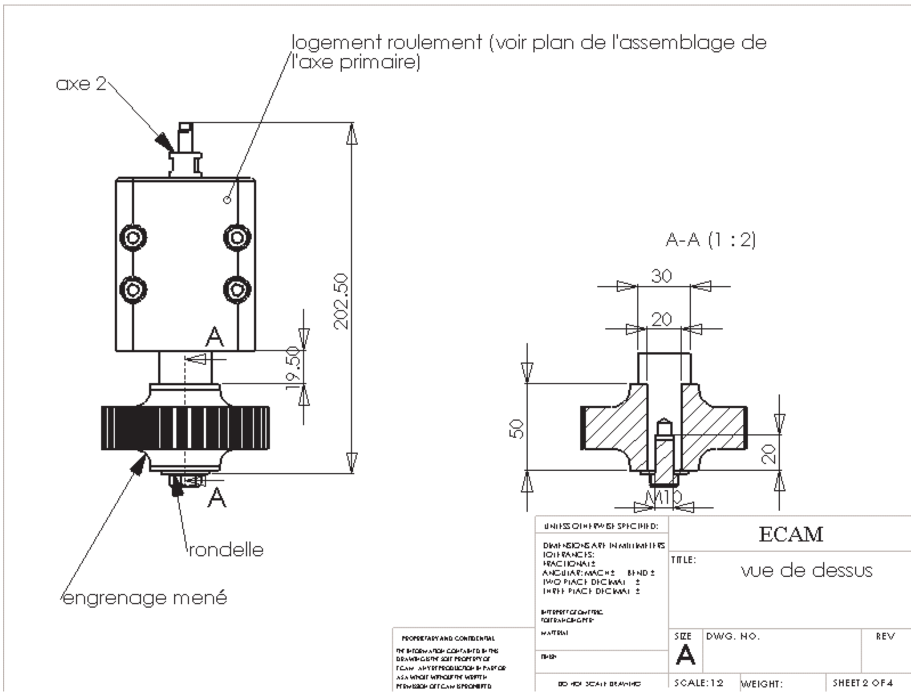


Figure C.7: Design of the secondary axle, top view

C.5 Design of the table

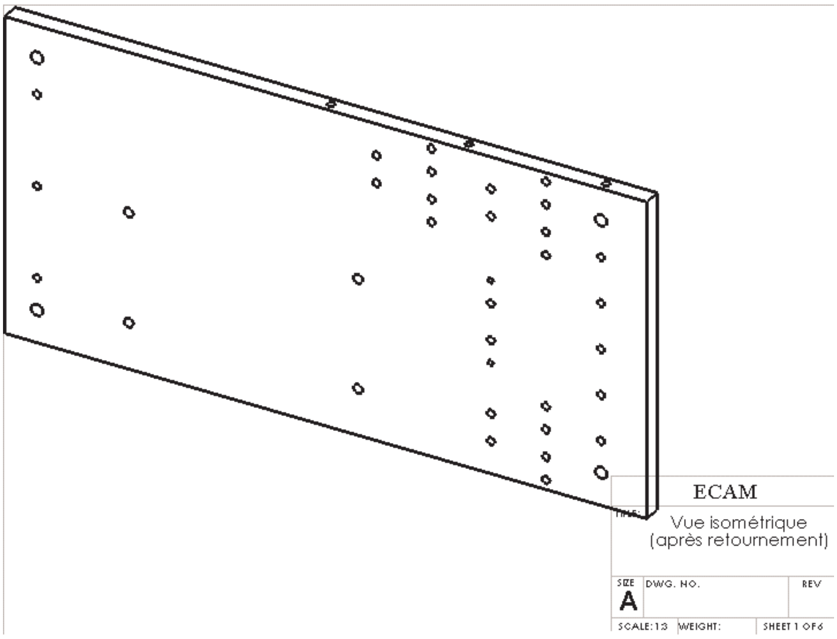


Figure C.8: Isometric view of the table

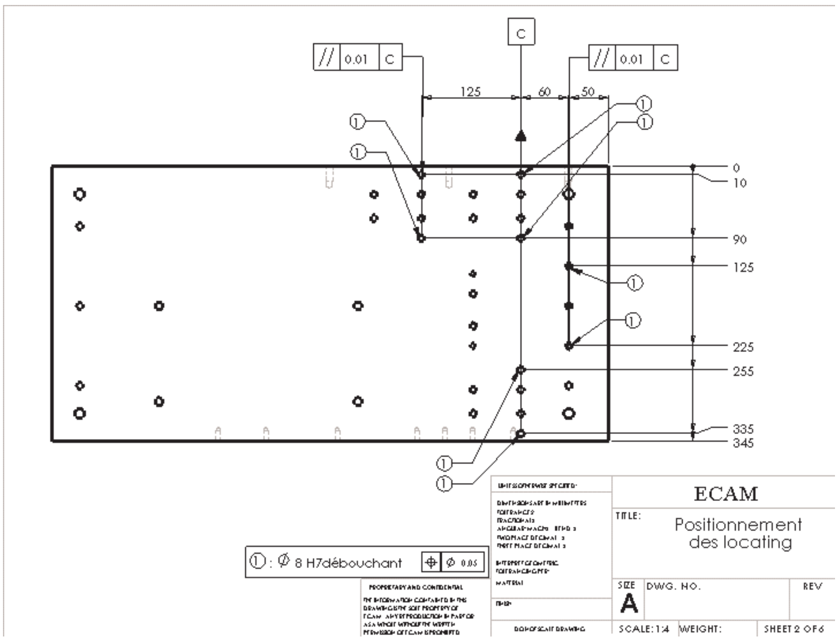


Figure C.9: Positioning of the holes for locating the bearing housings

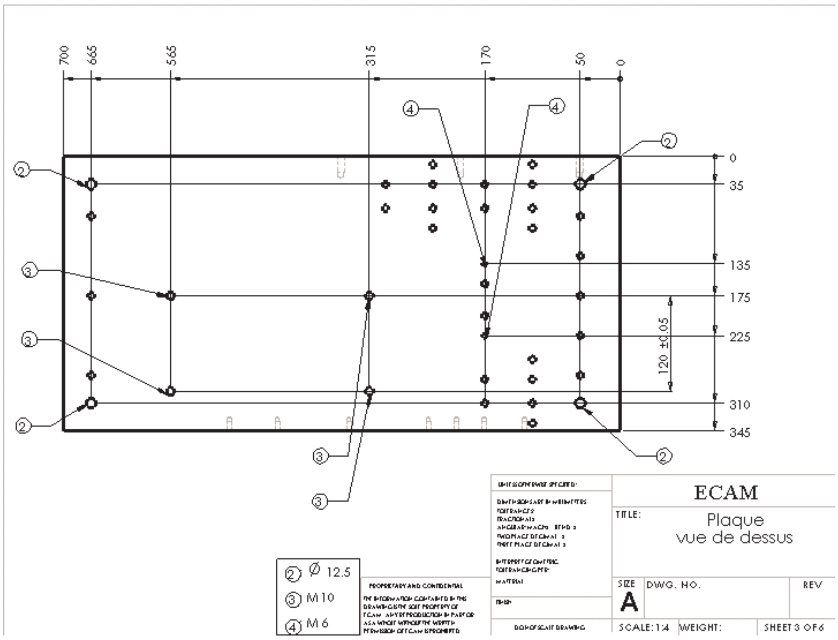


Figure C.10: Positioning of the major part of the holes

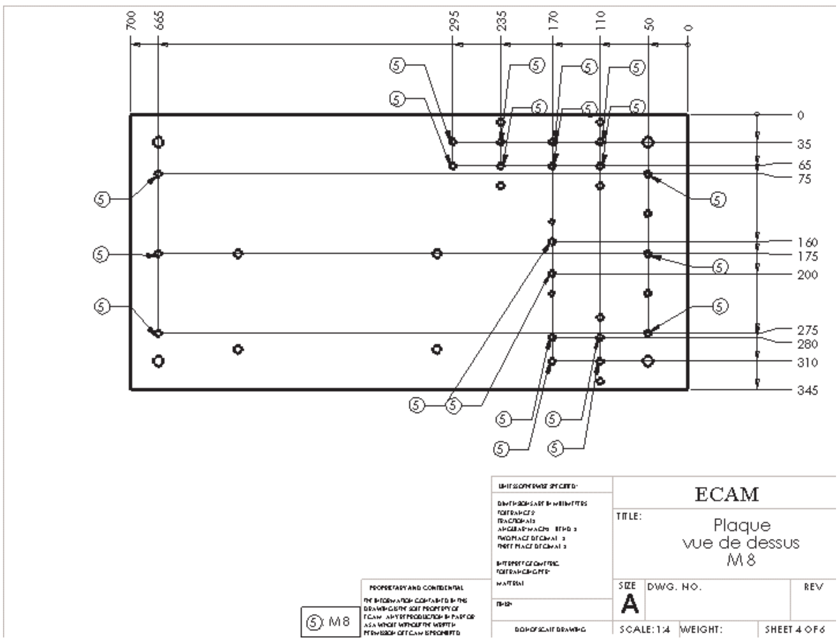


Figure C.11: Positioning of the holes, size M 8

C.6 Design of the oil housing

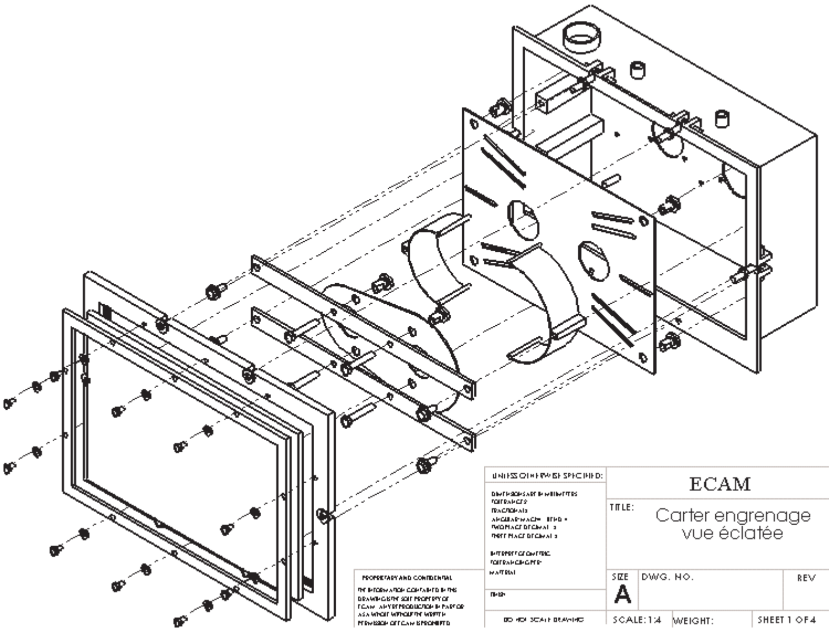


Figure C.12: Exploded view of the oil housing

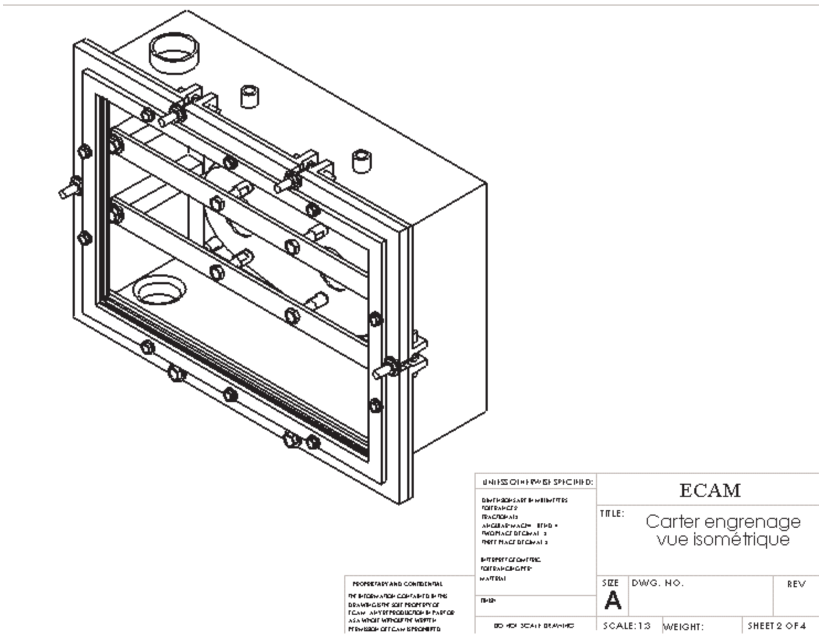


Figure C.13: Oil housing, assembled

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