A study to oil churning losses in a gearbox

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1 Introduction

The last few years ECAM collaborated with PSA (Peugeot and Citroën) and TOTAL FINA ELF on a subject called oil churning losses. Churning losses are caused by gear striking, pumping or otherwise moving the lubricant around in the gearbox.

From previous studies there are established two formulae. These two formulae are established with regard two regimes, namely for low and high speed of the gear. In this report the second formula for high speed will be improved and where necessary it will be adapted to have a good resemblance with the measurement.

To find a connection between the two speeds and so between two flow regimes, there has been formulated a Reynolds number that ascertains which formula will be used. At the end of the report some arbitrary geometries are chosen. For these geometries the influence of the speed and viscosity of oil will be determined.

2 Power losses in a gearbox

The heat generated in a gearbox depends of two sources: load and no- load dependent power losses. The load dependent losses consist of tooth friction and bearing losses. The no- load dependent losses consist of internal bearing viscous friction losses, oil churning, windage losses of gears operating in an oil- air mist and shaft seals losses. These losses can be controlled, for the most part, by careful design and construction. People already derived formulae for most of these losses, but there is only little knowledge about the oil churning effect. In this report a previous formula for oil churning losses at high speeds will be adapted and after there will be found a connection between the low speed and high speed formula for oil churning losses.

2.1 The test bench

To do the necessary measurement there is a test bench available. On this test bench people have made a lot off measurement before. The test bench used for these measurement treated in this report will be shown in figure 2.1.

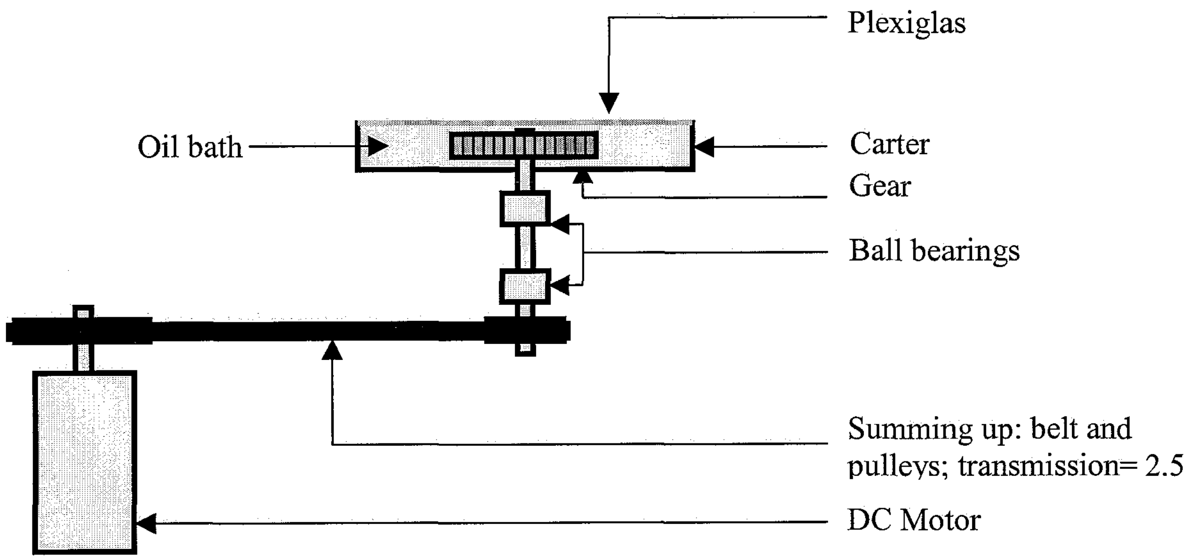

Figure 2.1: picture of the test bench used for measuring oil-churning losses

The gear is driven by a shaft, which will be supported by two ball bearings. The shaft will be speeded up by a belt transmission of 2.5. A 1.5 kW DC motor with a maximum speed of 3000 rpm drives the primary axis. So, the maximum speed of the gear will be 2.5 x 3000 = 7500 rpm. By varying the presented voltage of the inducer of the DC motor, the speed of the gear can be changed. The inductor will always have the same voltage and current. The DC motor changes its speed with 0.06 V/tr/min. For a desired rotational speed you can calculate the voltage to be applied with:

$$V_{DC-motor} = \frac{N \cdot 0.06}{transmission} \qquad \text{(Eq. 2.1)}$$

where N the rotational speed [rpm].

In this test bench many parameters can be modified. The following parameters can be changed:

- Gear: it means you can change the pitch diameter, the module and the face width
- Rotational speed of the gear
- The volume and the dynamic height of the oil bath
- Oil and so the viscosity

The gears used for measurement will have the parameters given in table 2.1:

Table 2.1: parameters of the used gear

Geometry	1	2	3	4	5
Module m [m]	0.0015	0.0015	0.003	0.003	0.005
Number of teeth Z []	64	102	30	53	30
Face width b [m]	0.014	0.014	24	24	24
Tooth depth h_{dent} [m]	0.003375	0.003375	0.00675	0.00675	0.01125
Pressure angle α [°]	20	20	20	20	20
Pitch diameter Dp [m]	0.096	0.153	0.090	0.159	0.150

The carter of the test bench is made of steel. Only the front side of the carter is made of Plexiglas, which makes it possible to see the oil level in the carter and the oil flow around the gear.

Many measurements have been done with oil H6965. But TOTAL supplies ECAM some other oils and so these oils have to be tested too. After some new tests with these oils the theoretical formula to quantify oil churning losses, which was derived before by Macian-Voutay **[1]**, should be improved or adapted. Namely, this formula is only tested for oil H6965.

2.2 Principle of measuring

For driving the primary axis, a DC motor will be used. In figure 2.2 the electric circuit of the motor is shown.

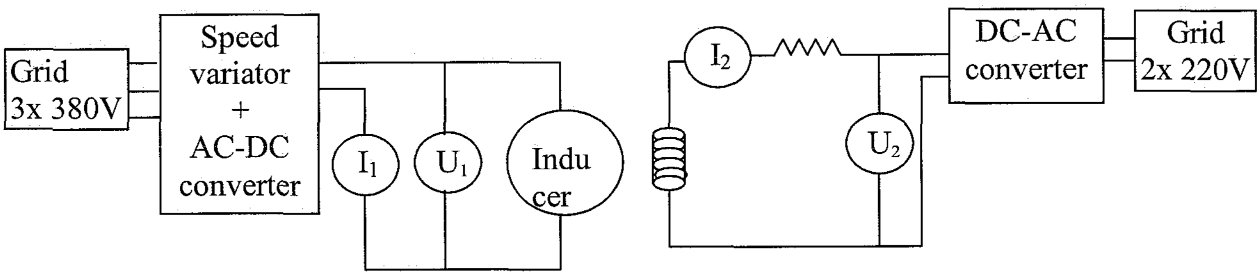

Figure 2.2: electric circuit of the DC motor

For measuring the oil churning losses equation 2.2 will be used:

$$P_{motor} = U_1 \cdot I_1 - U_{10} \cdot I_{10} - R_1 \cdot \left(I_1^{\,2} - I_{10}^{\,2}\right) - 2 \cdot U_{bal} \cdot \left(I_1 - I_{10}\right) \qquad \text{(Eq. 2.2)}$$

Where U_1, I_1: tension and current of the inducer/ rotor with a filled carter
U_{10}, I_{10}: tension and current of the inducer/ rotor with a empty carter
U_2, I_2: tension and current of the inductor/ stator with a filled carter
U_{20}, I_{20}: tension and current of the inductor/ stator with a empty carter
R_1: inducer resistance of the rotor
U_{bal}: drop voltage between cupper and carbon of the brushes

R_1 and U_{bal} will have the following constant values:

$R1=1.3\Omega$ and $Ubal=2V$

Equation 2.3 will be used when the motor runs at high speed (high speed means above 4000 rpm). To remove the oil from the carter the motor should be stopped. This stop should not take to much time because the efficiency of the motor will change with a changing temperature. This makes it impossible to remove the gear during a measurement. When the gear is still in the housing during a measurement without load, it is necesarry to calculate the losses due to the air moving around the gear. These losses are called windage losses. Windage losses will only be taken into account at high speeds, because at low speeds the influence of windage is very small.

$$P_{vent}=a \cdot N^{b}+c \quad \text{(Eq. 2.3)}$$

The windage losses are a function of the module, the face width and the tooth number of the gear. The constants a, b and c will have the value given in table 2.2.

Table 2.2: constant values for a, b and c, which depends on the values of module, face width and tooth number

Module [mm]	1.5	1.5	3	3	5
Face width [mm]	14	14	24	24	24
Tooth number [-]	64	102	30	53	30
a	$2.51.10^{-11}$	$3.52.10^{-11}$	$5.4.10^{-11}$	$3.44.10^{-10}$	$4.22.10^{-10}$
b	2.99	2.99	2.99	2.99	3
c	5.1	14	7.55	-	16.6

These losses by windage have to be taken into account when you analyse the oil churning losses. Equation 2.4 gives the total losses with respect to oil churning:

$$P_{churn}=P_{motor}+P_{vent} \quad \text{(Eq. 2.4)}$$

Two other parameters, which are important for the analysis, are the temperature of the oil sump and the temperature of the surrounding air. These temperatures will be measured with a thermocouple.

For measuring the desired parameters, first the system should be stable or operate at a steady state conditions. The system should be stable because the viscosity of the oil will decrease with a rising oil temperature. And when the viscosity of the oil decreases, also the resistance for turning the gear will decrease. There has been made an agreement to call the system stable when the temperature of the oil fluctuates less than 0.1 °C within 10 minutes.

One of the difficulties for making a good measurement at high speed is to verify the right immersion factor for the gear. This factor is defined as follows: It is the ratio of the submerged depth of a pinion to its pitch radius (see equation 2.5). This will be difficult because of the waving behavior of the oil.

$$immersionfactor = \frac{h}{Rp} = I \qquad \text{(Eq. 2.5)}$$

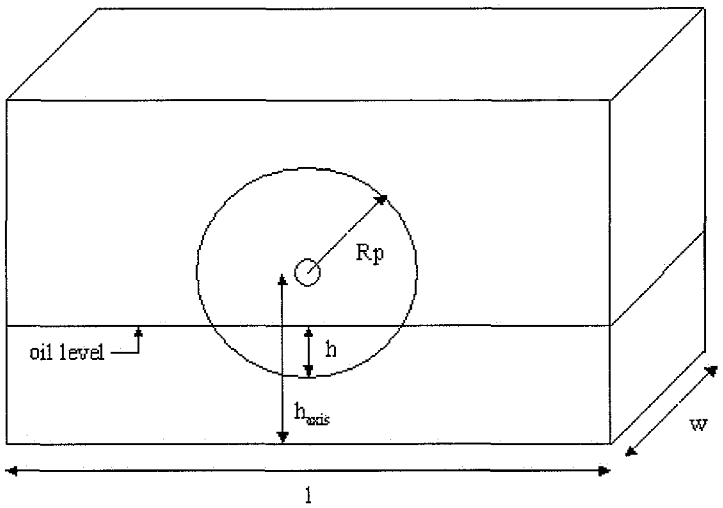

Figure 2.3: view of carter and definition of immersion factor

Another difficulty because of the waving behaviour of the oil is to verify the current on the ampere meter. Because of the waves this current fluctuates and so there should be found an average value for the current.

2.3 Coherence of the measurement

To simulate the thermal behaviour of mechanical power transmissions, ECAM has designed a computational program **[2]**. With this program, which is called RVI, it is possible to compare the measured heat exchange and power losses in a reduction gear with the results given by a numerical model.

For the test bench that is used for the measurement of oil churning a thermal analysis can be executed too. The test bench contains some machine elements that act as heat sources, for example a gear and bearings. For a thermal analysis, it is important to know that these sources are linked up together. So the test bench can be considered as a thermal network.

The creation of a thermal network can be summed up in the three following steps:

- Define and calculate power losses
- Divide the gearbox into elements
- Connect these nodal elements together by thermal resistance which depend on the kind of heat transfer

When you have calculated the power losses, in this case only the oil churning losses, you have to divide the gear unit into elements. Elements of the gearbox are: the oil, bearings, gears, shafts, carter and the air surrounding the gearbox. The final step consists in connecting these elements by thermal resistances, which depend on the kind of heat transfer, namely:

- Conduction
- Free or forced convection
- Radiation

The thermal network for the test bench is given in figure 2.4:

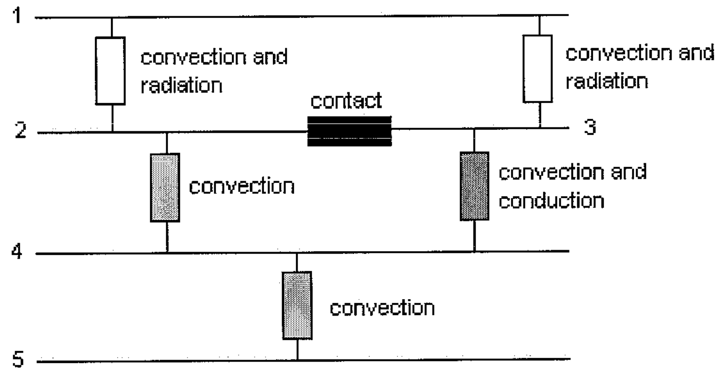

Figure 2.4: thermal network for the test bench

Where: 1. Air
2. Part of carter made of steel
3. Part of carter made of Plexiglas
4. Oil sump
5. Gear wheel

Now, for each element (i) of the network the first principle of thermodynamics can be written (see equation 2.6).

$$Q_i = \sum_j \frac{T_i - T_j}{R_{TH}(i,j)} + m_i \cdot c_i \cdot \frac{dT_i}{dt} \qquad \text{(Eq. 2.6)}$$

Where Q_i: power lost at node i [W]
T_i: temperature at node i (or j) [°C]
$R_{TH}(i,j)$: thermal resistance between the elements i and j [°C/W]
$m_i.c_i$: thermal inertia of element i
t: time [s]

By considering all the elements of the network, an equation in matrix form is obtained (see equation 2.7).

$$[Q] = [S_{TH}][T] + [m \cdot c]\left[\frac{dT}{dt}\right] \qquad \text{(Eq. 2.7)}$$

Where [dT/dt]: vector of temperature time derivative
$[S_{TH}]$: matrix of parameters $S_{TH}(i,j)$

$$\text{if } i = j \quad S_{TH}(i,i) = \sum_k \frac{1}{R_{TH}(i,k)} \qquad \text{(Eq. 2.8)}$$

$$\text{if } i \neq j \quad S_{TH}(i,j) = -\frac{1}{R_{TH}(i,j)} \qquad \text{(Eq. 2.9)}$$

Using a Runge-Kutta's method makes the numerical resolution of this differential equation. The program, with the implemented Runge-Kutta's method, calculates the temperature for the oil and the other elements. The purpose of the method of thermal network is to check if the temperature at the end of the test for the oil calculated with the program will be the same as the real oil temperature measured with the thermocouple. A variation of 10 °C in the temperatures of oil is allowed. If the difference is too large, the measurement is not reliable and has to be done again.

2.4 Theoretical model of ECAM

The Mechanical Engineering Department of ECAM has made a lot of work before on the subject of oil churning losses **[3]**. As a conclusion of these studies, it appears that the drag torque acting on a gear can be estimated as follows:

$$C_{churn} = \frac{1}{2} \cdot \rho \cdot \left(\frac{\pi \cdot N}{30}\right)^2 \cdot Sm \cdot Rp^3 \cdot Cm \qquad \text{(Eq. 2.10)}$$

Where C_{churn}: drag torque [Nm]
ρ: oil density [kg/m^3]
N: rotational speed of gear [tr/min]
Sm: wetted surface area of gear [m^2]
Rp: pitch radius of gear [m]
Cm: dimensionless drag torque coefficient [-]

For calculating the oil churning losses P_{churn} this C_{churn} should be multiplied by the rotational speed Ω:

$$P_{churn} = C_{churn} \cdot \Omega \qquad \text{(Eq. 2.11)}$$

Where P_{churn}: oil churning losses [W]
C_{churn}: drag torque [Nm]
Ω: rotational speed of gear [rad/s]

Because of the expected difference in two flow regimes at low speed (1000 till 3000 rpm) and at high speed (5000 till 7000 rpm) there has been worked in different projects on the oil churning effect. As result for the low speed project **[4]**, there was found a formula for the dimensionless drag torque coefficient given in equation 2.12:

$$C_m = \left(\frac{h}{Rp}\right)^{0.45} \cdot \left(\frac{V_0}{Dp}\right)^{0.1} \cdot Fr^{-0.6} \cdot \mathrm{Re}^{-0.21} \qquad \text{(Eq. 2.12)}$$

Where h: submerged depth of the gear [m]
V_0: oil volume in the carter [m^3]
Dp: pitch diameter [m]
Fr: Froude number [-]
Re: Reynolds number [-]

The oil volume V_0 within the carter can be calculated with the formula given in equation 2.13.

$$V_0 = w \cdot l \cdot \left(h_{axis} - Rp \cdot \left(1 - \frac{h}{Rp} \right) \right) \qquad \text{(Eq. 2.13)}$$

Where w: width of carter
l: length of carter
h_{axis}: height of axis till bottom carter

For definition of w, l and h_{axis} see figure 2.3.

The Froude and Reynolds numbers are given by:

$$Fr = \frac{\Omega^2 \cdot Rp}{g} \text{ and } Re = \frac{\Omega \cdot Rp^2}{\nu} \qquad \text{(Eq. 2.14, Eq. 2.15)}$$

Where υ: kinematic viscosity of oil [m^2/s]
g: gravitation acceleration [m/s^2]

For high speed there is also found a formula for the dimensionless drag torque coefficient **[1]**. This formula for high speed is given in equation 2.16:

$$C_m = 25.1 \cdot \left(\frac{h}{Rp} \right)^{1.39} \cdot \left(\frac{V_0}{Dp^3} \right)^{-0.827} \cdot Fr^{-0.6} \cdot Re^{-0.21} \cdot \left(\frac{b}{Dp} \right)^{1.23} \qquad \text{(Eq. 2.16)}$$

Where b: face width of gear

Now all parameters to calculate the theoretical oil churning losses are known or can be measured. We only have to express a formula for the wetted surface area of the gear. This wetted surface area exists of the surface of teeth and the surface of the flanks. So the total wetted surface area is given by:

$$Sm = Smd + Smf \qquad \text{(Eq. 2.17)}$$

with respectively the surface area for teeth and flanks given by:

$$Smd = Rp \left(2.b.Arcos(1 - \frac{h}{Rp}) \right) + \frac{4.Z.h_{dent}.b.Arcos(1 - \frac{h}{Rp})}{2.\pi.\cos\alpha} \qquad \text{(Eq. 2.18)}$$

$$Smf = Rp^2 \left(2.Arcos(1 - \frac{h}{Rp}) - \sin(2.Arcos(1 - \frac{h}{Rp})) \right) \qquad \text{(Eq. 2.19)}$$

Where Z: number of teeth [-]
h_{dent}: tooth depth [m]
α: pressure angle of tooth [°]

3 Measurement at high speed

To improve the formula derived by Macian- Voutay first there has to be made new test with other oils supplied by TOTAL.

3.1 Results from the measurement at high speed

The first step in this project is to make some measurement with two different kinds of oil at high speed (5000, 6000 and 7000 rpm). In table 3.1 four different kinds of oil are given with their properties. The third oil in this table has already been tested for some geometry of gears and the results are known.

Table 3.1: different kinds of oil with their properties

	Viscosity at 100°C (cst)	Viscosity at 40°C (cst)	Density of oil (kg/m^3)
H50298	6.16	31.6	874
H50410	6.56	28.4	845
H6965	8.3	48	873
MG 632	320	24	897.8

For calculating the viscosity at a certain oil temperature, measured with the thermocouple, the following equation will be used:

$$\log(\log(\nu + 0.6)) = A.\log T + B \qquad \text{(Eq. 3.1)}$$

Where ν: oil viscosity (cst)
T: oil temperature (°K)
A, B: constants

Now, when the measurement for the first two oils from table 3.1 have been done there is a difference between the measured power losses for oil H6965 and oils H50298/H50410.

In table 3.2 the results of the measurement for the three oils are compared with each other.

Table 3.2: power losses for different speeds with their viscosity at oil temperature

Measurement		H6965	H50298	H50410
Geometry 2 – N5000	Power losses [W]	183	162	165
	Viscosity (cst.)	20.93	21.89	23.23
	End temperature (°C)	63.3	50.2	46.2
	Temperature with RVI (°C)	63.5	58.5	60.3
Geometry 2 – N6000	Power losses [W]	232	194	192
	Viscosity (cst.)	15.92	17.68	19.56
	End temperature (°C)	73	56.8	51.9
	Temperature with RVI (°C)	74	65.9	64.6
Geometry 2 – N7000	Power losses [W]	271	247	247
	Viscosity (cst.)	13.42	15.30	14.97
	End temperature (°C)	79	61.6	61.6
	Temperature with RVI (°C)	82.2	77.4	77.5

From this table it can be seen that oils H50298 and H50410 have almost the same amount of power loss, but there is a difference in power loss with oil H6965. Looking at the viscosities for the different oils you see there is only a little difference in their value.

Three almost constant values for the viscosity at different speeds:

- At 5000 rpm: viscosity ~ 22 cst.
- At 6000 rpm: viscosity ~ 18 cst.
- At 7000 rpm: viscosity ~ 15 cst.

So, from this data we can say that the differince in power loss cannot be explained by viscosity.

The difference in temperature that is measured with the thermocouple and calculated with the computational program RVI for the two new oils are not so small as expected. A possible explanation for this big difference is the composition of these oils to a chemical point of view.

3.2 Ventilation

Because the influence of viscosity at high speed in not clear, the influence of ventilation is examined. During a test an amount of air will interfere into the oil. To find out what the influence of this air will be, the amount of air is measured for different conditions. For these test a volume of one litre is taken after the test. After, the volume will decrease because the air escapes. To express the amount of air into the oil α will be defined in equation 3.2.

$$\alpha = \frac{V_2}{V_1} \qquad \text{(Eq. 3.2)}$$

Where V_1: the volume of oil with air [m^3] = 1 litre
V_2: the volume of oil without air [m^3]

The values for α which are found for several tests are given in table 3.3.

Table 3.3: different values of α for different conditions

Oil/ speed (rpm)	H6965 m=1.5 D=153	H50298 m=1.5 D=159	H50298 m=3 D=159	H50410 m=1.5 D=153	H50410 m=3 D=159	MG632 m=1.5 D=159
5000	0.95	0.96	0.95	0.98	0.97	0.95
6000	0.94	0.95	0.95	0.96	0.96	0.93
7000	0.94	0.94	0.94	0.96	0.96	0.93

With this factor α, the new values for the oil viscosity and density can be calculated. These values are given by equation 3.3 and 3.4.

$$v_{oilbath} = \alpha . v_{oil} + (1-\alpha) v_{air} \qquad \text{(Eq. 3.3)}$$

$$\rho_{oilbath} = \alpha . \rho_{oil} + (1-\alpha) \rho_{air} \qquad \text{(Eq. 3.4)}$$

Now, because there is a little difference between the viscosity of the oil (about 20 cst) and the viscosity of air (about 16 cst) the influence of equation 3.3 is small and so ventilation has no influence on the viscosity of the oil sump.

However, the air volume in the oil has more influence on the density of the oil sump. Because the density of the air (about 1.2 kg/m^3) is smaller than the density of oil (about 874 kg/m^3 for some particular oil), the present air decreases the total density. Running at high speeds, the density will decrease from:

- 873 kg/m^3 to about 820 kg/m^3 for H6965
- 874 kg/m^3 to about 820 kg/m^3 for H50298
- 845 kg/m^3 to about 810 kg/m^3 for H50410
- 898 kg/m^3 to about 835 kg/m^3 for MG632

The most important conclusions from these results are:
there is no difference between ventilation for different oils and the total density is only decreasing for a few percents. The amount of decreasing of the density could taken as a constant value in the formula.

Looking at the influence of air to the results of oil sump viscosity and density it can be concluded the influence of ventilation is very small. Because of the little influence of ventilation it is not possible to explain the difference in power losses by ventilation. This is the reason why ventilation is not taken into account for calculating the power losses.

To be sure there is no influence of the viscosity on the power loss another test has been done. Now the fourth oil with a very high viscosity has been chosen (320 cst. at 100 °C). The results are given in table 3.4.

Table 3.4: power losses for two different oils at three different speeds

Geometry		H6965	MG632
Geometry 2 – N5000	Power losses [W]	183	190
	Viscosity (cst.)	20.93	78.51
	End temperature (°C)	63.3	67.0
	Temperature with RVI (°C)	63.5	64.8
Geometry 2 – N6000	Power losses [W]	232	233
	Viscosity (cst.)	15.92	50.58
	End temperature (°C)	73	77.8
	Temperature with RVI (°C)	74	74.7
Geometry 2 – N7000	Power losses [W]	271	280
	Viscosity (cst.)	13.42	46.05
	End temperature (°C)	79	80.3
	Temperature with RVI (°C)	82.2	84.0

From this table its clear there is only a small difference in the value for the power losses, whereas there is a big difference between the two viscosities for the two different oils. When we look at the meaning of Reynold number

$$Re=\frac{inertial\_forces}{viscous\_forces},$$

and we know for high speed the inertial forces are much more bigger as the viscous forces, than the viscosity has no more effect for high speed.

3.3 Derivation of a new formula

One of the goals of this internship is to find a representative dimensionless drag torque coefficient C_m. So, with this drag torque coefficient the best result will be found between measured power losses and theoretical power losses. Next, the drag torque coefficient given by equation 2.14 is written again, only with unknown coefficients A, α, β, γ, δ and χ.

$$C_m = A \cdot \left(\frac{h}{Rp}\right)^{\beta} \cdot \left(\frac{V_0}{Dp^3}\right)^{-\chi} \cdot Fr^{-\gamma} \cdot \mathrm{Re}^{-\alpha} \cdot \left(\frac{b}{Dp}\right)^{\delta} \qquad \text{(Eq. 3.5)}$$

α: coefficient of Reynolds

From chapter 3.1 it already appeared that viscosity has no influence on the power losses for high speeds. To prove this the next formula will be used:

$$\frac{Cm_a}{Cm_b} = \left(\frac{\mathrm{Re}_a}{\mathrm{Re}_b}\right)^{\alpha} \qquad \text{(Eq. 3.6)}$$

with index a for oil H6965 and index b for oil MG632 (the data of these measurement is given in appendixes B1 and B2). All other variables with respect to geometry and so on will not change when there is chosen a different oil, so they don't effect the value for α.

At least equation 3.6 will result in the following formula for the value of α:

$$\alpha = \frac{\ln\left(\frac{Cm_a}{Cm_b}\right)}{\ln\left(\frac{\mathrm{Re}_a}{\mathrm{Re}_b}\right)} \qquad \text{(Eq. 3.7)}$$

With equation 3.7 the following values for α are found:

Table 3.5: values for coefficient α

Speed (rpm)	α
5000	-8.10^{-3}
6000	19.10^{-3}
7000	$-2.8.10^{-3}$

Finding these values for α it can be concluded the viscosity indeed has a very small influence on the power losses and the number of Reynolds can be removed from the formulation, there the Reynolds number is given by: $\mathrm{Re} = \frac{\Omega \cdot Rp^2}{\nu}$.

γ: coefficient of Froude

To ascertain the value for the coefficient of Froude the next formula will be used:

$$\gamma = \frac{\ln\left(\frac{Cm_a}{Cm_b}\right)}{\ln\left(\frac{Fr_a}{Fr_b}\right)} \quad \text{(Eq. 3.8)}$$

with index a and b the parameters for different speeds. The other parameters in this measurement are constant and will not influence the coefficient of Froude. The calculated values for γ are shown in appendix A.

β: coefficient of immersion

For the influence of the immersion factor formula 3.9 will be used:

$$\beta = \frac{\ln\left(\frac{Cm_a}{Cm_b}\right)}{\ln\left(\frac{I_a}{I_b}\right)} \quad \text{(Eq. 3.9)}$$

with index a and b referring to immersion factors of 25 and 50%. In appendix A several values for β are calculated. The values for β vary between -0.35 and 1.17. The value of 1.17 has been found for geometry 2, which geometry is close to the geometry of a disk. Because for all other geometries the coefficient of immersion is near to 0, we have to be carufull when we use this result. For the best solution of the dimensionless drag torque coefficient every coefficient should be dependent of the geometry of the gear. After trying to find the best solution for β the next formula is found which represent for different geometries the value of β:

$$\beta = \beta_0 - \left(\frac{m}{m_0}\right)^{-0.23} \cdot \ln\left(\frac{b \cdot m}{b_0 \cdot m_0}\right) \quad \text{(Eq. 3.10)}$$

where the parameters with index 0 are referred to geometry 2.

The resulting values for β give a good estimation of the β belonging to a geometry, but this formula is tested on to less geometry to be sure if this formula for β is right. That is the reason there is taken a constant value for β for all different geometries. This value is the mean value of the five different geometries and will be around 0.1.

δ: coefficient of b/Dp

To ascertain the influence of the face width formula 3.11 will be used:

$$\delta = \frac{\ln\left(\frac{Cm_a}{Cm_b}\right)}{\ln\left(\frac{b_a \cdot Dp_b}{b_b \cdot Dp_a}\right)} \quad \text{(Eq. 3.11)}$$

with index a and b for different geometries. For calculating δ geometries 2, 4 and 5 will be used, because they have almost the same pitch radius but not the same face width. Geometry 2 has a face width of 14 mm and geometries 4 and geometry 5 has a face width of 24 mm. This makes it possible to compare the measured result of geometry 2 with those of geometries 4 and 5. Only the results for immersion factor 0.5 will be used because for immersion factor 0.25 the results for geometry 2 are not reliable. Namely, between the power losses for immersion factor 0.25 and 0.5 for geometry 2 there is a big step. In appendix A the values for δ are given.

A, χ: respectively value of constant and coefficient of V_0/Dp^3

Because V_0 and Dp are two variables that are used many times in other parameters, it is not easy to find an expression to determine the value of χ. For this reason V_0 and also constant A will be chosen when we compare the measured results with the results from the derived formula.

4 Graphical view of new formula

Now most coefficients are known it is possible to compare the measured power losses with the calculated losses. After this comparison it is possible to choose the best values for the constant A and the coefficient χ.

4.1 Comparison between new formula and formula of Macian- Voutay

After comparing the results of the measured power losses and the calculated power loss the constant A and coefficient χ will have the following values: respectively 3.4 and 0.85.

In figure 4.1 and figure 4.2 there will be made a comparison between the old formula for high speeds that was derived by Macian and Voutay, and the formula for high speeds that is derived in this report. In figure 4.1 the results for geometry 1 till 3 will be shown, in figure 4.2 those of geometry 4 and 5. The new formula is given by formula 4.1:

$$Cm = 3.4 \cdot \left(\frac{h}{Rp}\right)^{0.1} \cdot \left(\frac{V_0}{Dp^3}\right)^{-0.85} \cdot Fr^{-0.88} \cdot \left(\frac{b}{Dp}\right)^{0.85} \qquad \text{(Eq. 4.1)}$$

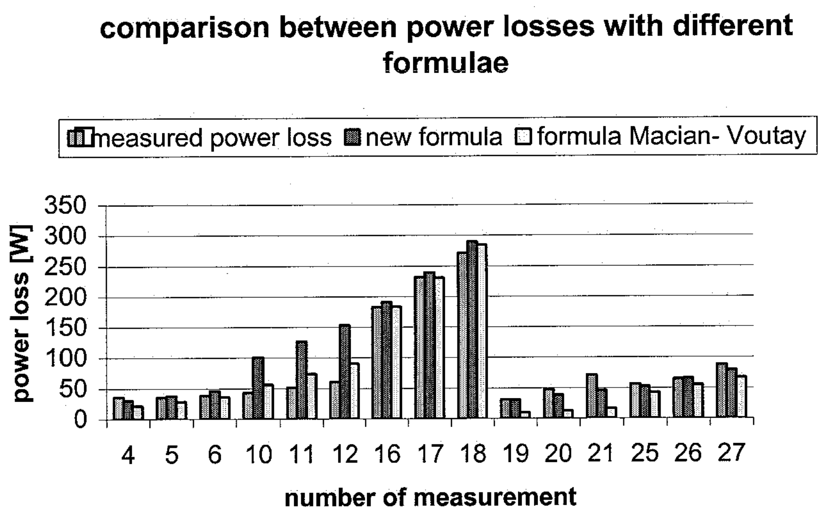

Figure 4.1: comparison between new formula and the formula of Macian- Voutay

It will be clear that for figure 4.1 the results of the new formula are good in comparison with the measured churning power losses. Only for measurement 10- 12 the results are bad. These bad results for these three measurements, geometry 2 with an immersion factor of 0.25, are due to the constant coefficients, which are chosen for the new formula. For geometry 2 the coefficient for the immersion factor (h/Rp) should be near to 1.17 and not have the value of 0.1 like chosen in formula 4.1.

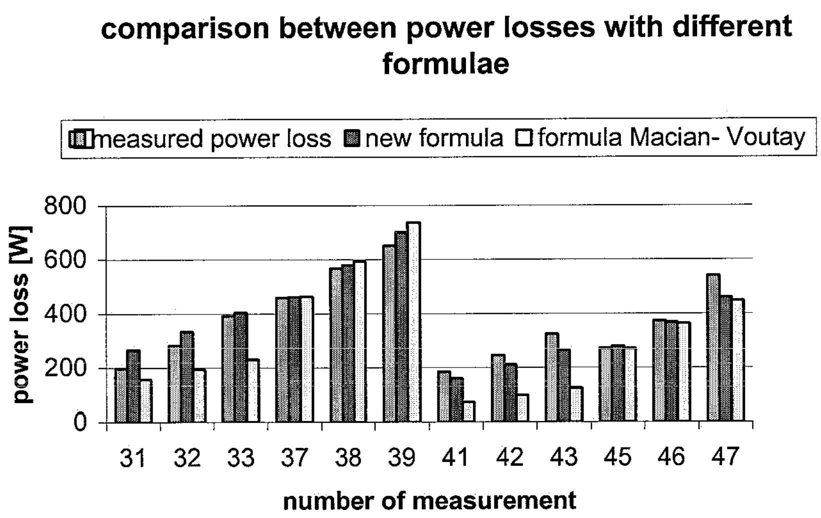

Figure 4.2: comparison between new formula and the formula of Macian- Voutay

In figure 4.2 the results for geometry 4 and 5 are shown. For these geometries the new formula will give better results than the old formula will. Especially for the measurement with a low immersion factor the results a much better.

Because for the derivation of the new formula the results of the measurement with the oil with high viscosity are used, we can look for the results of this measurement in comparison with the new and old formulation. In figure 4.3 the results are shown for geometry 2, immersion factor 0.5, where the first three measurement use oil H6965 and the other three measurement use oil MG632.

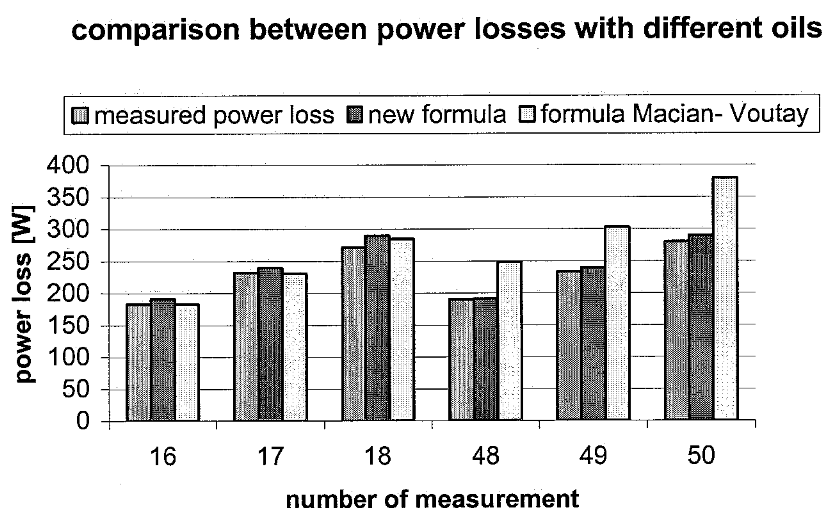

Figure 4.3: comparison between formula derived in this report and the formula of Macian- Voutay when there are used two different kind of oils

Because the ancient formula of Macian- Voutay is dependent of the viscosity of oil at high speed and the new formula is independent of viscosity, there is a difference for the several measurements. For these measurement the new formulation will again give the best results.

4.2 Demand for use of formula low or high speed

From measurement before it becomes clear that oil churning losses depend on the speed and geometry of gear. In figure 4.4 three modules are shown with their tangential speed versus the drag torque coefficient. For small speeds the drag torque coefficient are about equal, but for higher speeds there is a difference between the drag torque coefficient of module 1.5 and those of module 3 and module 5. Because for these three modules the pitch diameter is almost equal, this difference can be explained by the difference in face width.

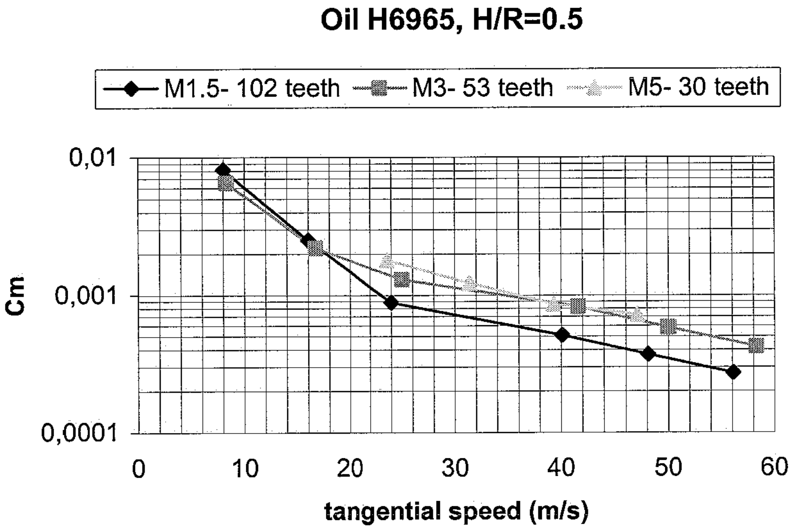

Figure 4.4: tangential speed versus drag torque coefficient for three modules

It is also known that at low speeds the viscosity has a big influence on the oil churning losses, because the oil is still cold and so the viscosity is high. High viscosity means of course a lot of resistance and so at low speeds viscosity is an important parameter. In figure 4.5 two series for the dimensionless drag torque coefficients are drawn. It becomes plain in this figure that for low speeds viscosity has a big influence.

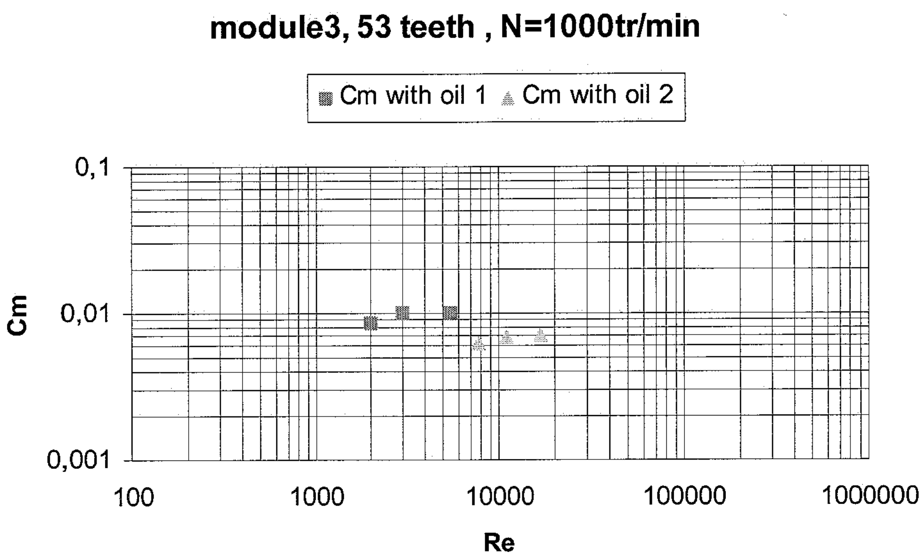

Figure 4.5: dimensionless drag torque coefficient versus Re for two different oils

Using all these parameters it is possible to choose a dimensionless number that will ascertain the switch point between the use of the formula for low speeds (given in equation 2.12) and the one for high speeds (equation 4.1). This new Reynolds number Re^x will be defined in equation 4.2:

$$Re^x = \frac{v_t \cdot b}{\nu} \qquad \text{(Eq. 4.2)}$$

Where v_t: rotational speed = ω.Rp [m/s]
b: face width [m]
ν: viscosity at oil temperature [m^2/s]

In figure 4.6 there are six measurements at six speeds, namely: 1000 till 3000 and 5000 till 7000 rpm. For each measurement there is looked which formula (low or high speed) gives the best results with respect to the measured power loss. A – means the low speed gives the best result, a + means the high speed formulation gives the best result and a ± means both formulations are good. These signs are given in table 4.1 and table 4.2.

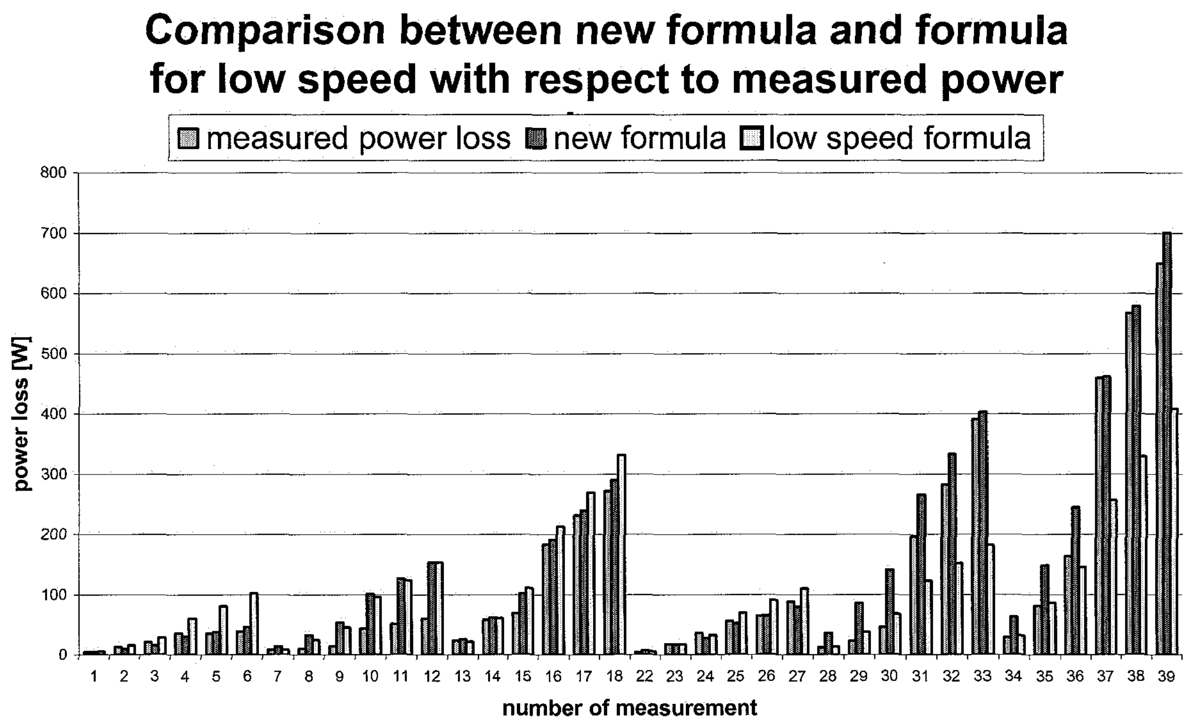

Figure 4.6: comparison between low and high speed formula

Table 4.1: +, - and ± for comparing formulations and signs < and > for demand

Nr.	1	2	3	4	5	6	7	8	9	10	11	12
Sign	±	±	±	+	+	+	-	-	-	±	±	±
Re^x	703	1674	2939	6298	7689	9361	1108	2466	4283	9865	13450	18535
Demand	<	<	<	>	>	>	<	<	<	>	>	>
Nr.	13	14	15	16	17	18	22	23	24	25	26	27
Sign	-	-	+	+	+	+	-	-	-	+	+	+
Re^x	1373	4138	7482	26793	42281	58487	1153	2757	4835	10610	14021	20758
Demand	<	<	>	>	>	>	<	<	<	>	>	>

Table 4.2: +, - and ± for comparing formulations and signs < and > for demand

Nr.	28	29	30	31	32	33
Sign	-	-	-	+	+	+
Re^x	2037	5087	10094	47880	83021	132891
Demand	<	<	>	>	>	>
Nr.	34	35	36	37	38	39
Sign	-	-	-	+	+	+
Re^x	2642	8743	23417	127151	185012	248690
Demand	<	>	>	>	>	>

The value, for which the new Reynolds number will switch from the low speed formula to the new formula for high speed, will be chosen equal to 6000. To check this value for each measurement from figure 4.6 there is given a sign in table 4.1 and 4.2 for the demand. When the value of the demand is under 6000 than the sign will be a < and if the value is bigger the sign will be a >. For a good result the – should match with the < and the + should match with the >. This because a – and < both means the low speed formula has the best result and + and > both means the new formula for high speeds has the best result.

Table 4.1 shows most of the results match with the two signs. Only for geometry 4 some signs (measurement 30, 35 and 36) are wrong. So this new Reynolds number is reliable for most of the measurement. This wills able people to choose their own geometry and speed, and the Reynolds number determines which formula should be used.

4.3 Simulation for churning losses for arbitrary geometries

After finding a new Reynolds number that will ascertain which formula will be used, there can be made two simulations. One simulation will be with a constant viscosity and with a varying speed. Here the speed will consist of low and high speeds. Another simulation will be simulation with a chosen constant speed, but with a varying viscosity. So after both simulations can be ascertained what the influences of speed and viscosity are.

4.3.1 Simulation with constant viscosity at variating speeds

When we choose an arbitrary geometry we will be able to draw the churning losses versus speed. To know which formula for the churning losses should be used the new Reynolds number should be calculated. When the value of this Reynolds number will pass the border of 6000 the flow will change his regime. For the next three gears, given in table 4.2, there are chosen constant values of viscosity and the speed varies from 1000 rpm till 7000 rpm.

Table 4.2: three arbitrary chosen geometries

Geometry	Module (mm)	Pitch diameter (mm)	Face width (mm)	Viscosity (cst)
6	1	100	20	20
7	2	200	15	60
8	1	50	15	40

Figures 4.7, 4.8 and 4.9 give the power losses versus speeds for the three geometries.

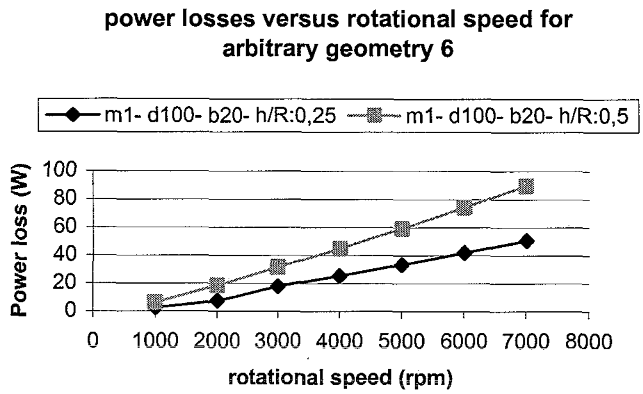

Figure 4.7: power losses versus speed for geometry 6

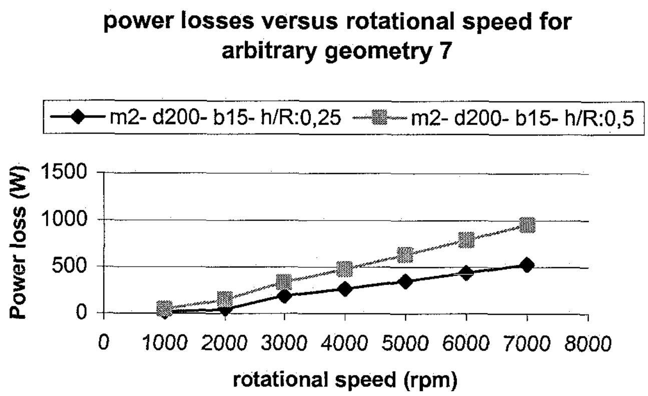

Figure 4.8: power losses versus speed for geometry 7

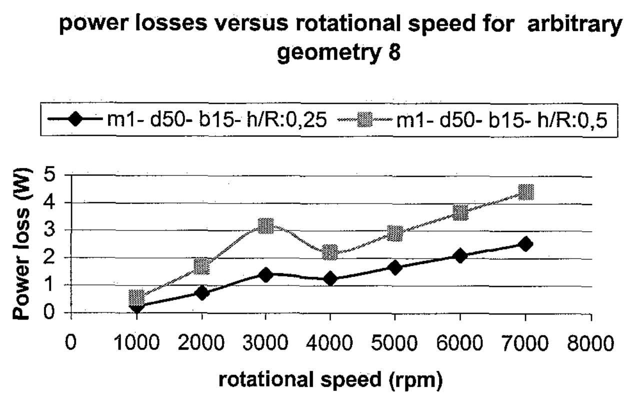

Figure 4.9: power losses versus speed for geometry 8

From this figures it can be seen that due to our new Reynolds number Re^x it is possible to choose your own geometry and plot the churning losses against the speed. Only in figure 4.9 there is a drop in the power losses between 3000 and 4000 rpm. Here the gear changes from the low speed regime to the high speed regime.

In fluid flows this drop of power losses or drag torque is familiar. When we look at figure 4.10 we see the same drop for the dimensionless drag torque coefficient for a disk. For low speeds the viscous forces acting on the gear are the most important forces, but for higher speeds the internal forces are the most important one. An example of an internal force is the force due to inertia. The switch of these forces could explain the drop in figure 4.9. Furthermore, as expected, a higher speed will give higher oil churning losses.

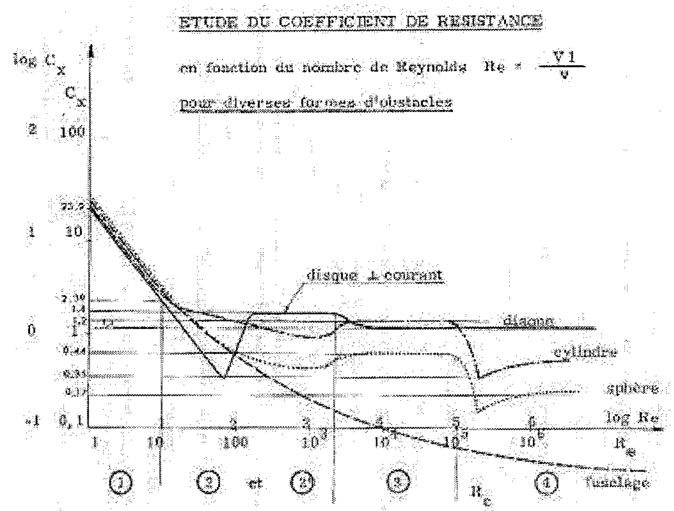

Figure 4.10: drag torque coefficient versus Reynolds number for different kind of geometries

4.3.2 Simulation with constant speed at variating viscosity

Because in section 4.3.1 the viscosity was taken as a constant, now we will see what the influence of a variating viscosity will be. In this section the viscosity will vary from 440 cst till 4 cst and there has been chosen a constant speed. Again two arbitrary geometries are chosen and now the dimensionless drag torque versus Reynolds number are drawn. The geometries that have been chosen are given in table 4.3.

Table 4.3: two arbitrary chosen geometries

Geometry	Immersion factor	Module	Pitch diameter	Face width	Speed
9	50%	5	200	20	2000
10	25%	1	100	10	6000

The dimensionless drag torque versus speed for these two measurement are drawn in figure 4.11.

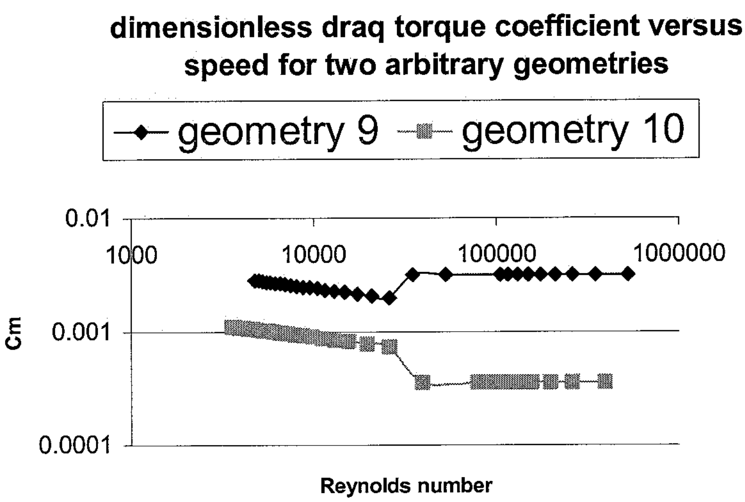

Figure 4.11: dimensionless drag torque coefficient versus Reynolds number for two arbitrary chosen geometries

To make a comparison between figure 4.10 and figure 4.11 also figure 4.11 is drawn on logarithmic scale. For geometry 10 figure 4.11 shows comprehensible results. Because for a low Reynolds number and so a high viscosity the resistance of oil will be high. The high resistance of oil will give a high churning loss. After geometry 10 changes of regime there is no more influence of viscosity and the churning losses will be constant.

Geometry 9 shows almost the same result as in figure 4.9 was shown. First the value of the dimensionless drag torque will decrease for higher Reynolds numbers. After a step in the dimensionless drag torque coefficient again there is no influence of viscosity and the churning losses will be constant. From figure 4.11 it appears the influence of viscosity on the oil churning losses depends on the geometry that is chosen.

From figure 4.7 till 4.9 and figure 4.11 it appears to be that the new formula in connection with the low speed formula and the new Reynolds number can give a good estimation of the oil churning losses.

Conclusion

After studying the previous results and making the necessary measurement it becomes plain the oil churning losses, which is in opposition to the old formula derived by Macian- Voutay, are not dependent of the viscosity of oil. This independency of viscosity for high speed will result in the removal of the Reynolds number in the formula that was derived by a dimensionless analysis before.

With the results of the made measurement the coefficients for the new formula are derived. After the two formulae for high speeds are compared with the measured power losses at high speed, it appears the new formula has better results for the measurement.

Because of the coefficients in the new formula are not dependent of the geometry, not for all measurement the new formula gives the best results. For some measurement with immersion factor 0.25 and low oil churning losses the new formula gives worse results.

To find a connection between the two formulae for low and high speed a new Reynolds number has derived. This number determines which formula under certain conditions will be used. When this Reynolds number was applied to arbitrary geometries it appears this number gives correct results.

Some recommendations for the future are:

- Apply a torque sensor to the test bench, so instead of an electric power loss the mechanical power loss can be measured. For measuring the power losses now two big values are substracted from each other that can give a deviation in the real value for the power losses.
- For the best value of power losses it is recommended to use a digital analysis. This analysis will give an average value for the measurement.
- Make the coefficients that will be used in the new formula dependent of the chosen geometry. So for all measurement the best value of coefficients will be used.

List of symbols

Symbols	Description	[Unit]
A, B	Constants to determine viscosity	[-]
a, b, c	Constant to calculate windage loss	[-]
α	Pressure angle tooth	[°]
b	Face width	[m]
Cm	Drag torque coefficient	[-]
Dp	Pitch diameter	[m]
Fr	Froude number	[-]
g	Gravitation acceleration	[$m.s^{-2}$]
h_{dent}	Tooth depth	[m]
h	Dynamic immersion height of gear	[m]
h_{stat}	Static level of oil bath	[m]
I	Immersion factor	[-]
I1	Current of the inducer with load	[V]
$I1_0$	Current of the inducer without load	[V]
m	Module	[m]
N	Rotational speed	[$tr.min^{-1}$]
ν	Kinematics viscosity	[$m^2.s^{-1}$] = [10^6 cst]
P	Power	[W]
R1	Resistance of DC motor	[Ω]
Re	Reynolds number	[-]
Rp	Pitch radius	[m]
ρ	Density of oil	[$kg.m^{-3}$]
Sm	Wetted surface area	[m^2]
Smd	Wetted surface teeth	[m^2]
Smf	Wetted surface flanks	[m^2]
T	Temperature of oil bath	[°C]
T_{amb}	Temperature of surroundings	[°C]
U1	Tension of the inducer with load	[V]
$U1_0$	Tension of the inducer without load	[V]
Ubal	Contact tension for DC motor	[V]
V_0	Oil volume of carter	[m^3]
V_1	Taken volume after test with air	[m^3]
V_2	Taken volume after test without air	[m^3]
Ω	Rotational speed	[$rad.s^{-1}$]
Z	Number of teeth	[-]

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[2] – Changenet C., Denis M. and Nast J.D.: "Thermal study of railway reduction gears". VDI- Berichte n° 1230, 1996

[3] – Changenet C. and Heckly A. : "Rendement et comportement thermique d'une boîte de vitesse à cing rapports". 4th world congress on gearing and power transmission, Paris, March 1999

[4] – N. Canitrot: "Pertes par barbotage dans les boîte de vitesses : formulation et simulation dans les boîtes MA, BE, ML". TFE ECAM Lyon 1998

Appendix A Calculation of the coefficients α, β, γ, δ and χ

Calculation of α

Measurement	Kind of oil	Power loss [W]	Cm	Viscosity [cst]	Re	α
Geometry 2– I0.5 – N5000	H6965	183	0.00050708	20.93	146406	-8.10^{-3}
	MG632	190	0.00051249	78.51	39029	
Geometry 2– I0.5 – N6000	H6965	232	0.00037179	15.92	231034	19.10^{-3}
	MG632	233	0.00036370	50.58	72701	
Geometry 2– I0.5 – N7000	H6965	271	0.00027426	13.42	319590	$-2.8.10^{-3}$
	MG632	280	0.00027524	46.05	93164	
					average	$2.7.10^{-3}$

Calculation of γ

Measurement	Kind of oil	Rotational speed [rpm]	Power loss [W]	Cm	Fr	γ
Geometry 2– I0.5	H6965	5000	183	0.000507	2139	-0.85
		6000	232	0.000372	3080	
Geometry 2– I0.5	H6965	6000	232	0.000372	3080	-0.99
		7000	271	0.000274	4192	
Geometry 2– I0.5	H6965	5000	183	0.000507	2139	-0.91
		7000	271	0.000274	4192	
Geometry 1– I0.5	H6965	5000	36	0.000825	1342	-1.51
		6000	35	0.000463	1932	
Geometry 1– I0.5	H6965	6000	35	0.000463	1932	-1.24
		7000	38	0.000316	2630	
Geometry 1– I0.5	H6965	5000	36	0.000825	1342	-1.39
		7000	38	0.000316	2630	
Geometry 3– I0.25	H6965	5000	34	0.001117	1258	-0.30
		6000	48	0.001001	1811	
Geometry 3– I0.25	H6965	6000	48	0.001001	1811	-0.23
		7000	71	0.000932	2466	
Geometry 3– I0.25	H6965	5000	34	0.001117	1258	-0.27
		7000	71	0.000932	2466	
Geometry 2– I0.5	MG632	5000	190	0.000513	2139	-0.90
		6000	233	0.000364	3080	
Geometry 2– I0.5	MG632	6000	233	0.000364	3080	-0.94
		7000	280	0.000275	4192	
Geometry 2– I0.5	MG632	5000	190	0.000513	2139	-0.92
		7000	280	0.000275	4192	
					average	-0.87

Calculation of β

Measurement	Immersion factor	Power loss [W]	Cm	β
Geometry 1 – 7000 rpm	25%	23	0.000341	-0.11
	50%	38	0.000316	
Geometry 2 – 5000 rpm	25%	43	0.000234	1.12
	50%	183	0.000507	
Geometry 2 – 6000 rpm	25%	51	0.000162	1.20
	50%	232	0.000372	
Geometry 2 – 7000 rpm	25%	60	0.000119	1.20
	50%	271	0.000274	
			average geometry 2	1.17
Geometry 3 – 5000 rpm	25%	31	0.001120	0.10
	50%	56	0.001196	
Geometry 3 – 6000 rpm	25%	48	0.001211	-0.60
	50%	65	0.000800	
Geometry 3 – 7000 rpm	25%	71	0.001010	-0.56
	50%	88	0.000687	
			average geometry 3	-0.35
Geometry 4 – 5000 rpm	25%	196	0.000635	0.36
	50%	459	0.000817	
Geometry 4 – 6000 rpm	25%	282	0.000528	0.15
	50%	567	0.000584	
Geometry 4 – 7000 rpm	25%	391	0.000460	-0.13
	50%	650	0.000421	
			average geometry 4	0.13
Geometry 5 – 5000 rpm	25%	245	0.001017	-0.25
	50%	372	0.000857	
Geometry 5 – 6000 rpm	25%	323	0.000775	-0.11
	50%	539	0.000719	
			average geometry 5	-0.18
			overall average	0.132

Calculation of δ

Measurement	Geometry	Power loss	Cm	b [m]	Dp [m]	δ
N 5000- I 0.5	2	183	0.000507	0.014	0.153	0.95
	4	459	0.000817	0.024	0.159	
N 6000- I 0.5	2	232	0.000372	0.014	0.153	0.90
	4	567	0.000584	0.024	0.159	
N 7000- I 0.5	2	271	0.000274	0.014	0.153	0.86
	4	650	0.000421	0.024	0.159	
N 5000- I 0.5	2	183	0.000507	0.014	0.153	0.94
	5	372	0.000857	0.024	0.150	
N 6000- I 0.5	2	232	0.000372	0.014	0.153	1.18
	5	540	0.000719	0.024	0.150	
					average	0.97

Appendix B1 measurement 1 t/m 27

number of measure ment	used oil	rota tional speed (rpm)	Immersion factor I	geo metry	Module (m)	number of teeth	tooth depth h dent (m)	pressure angle (°)	face width (m)	Diameter (m)	churning losses without ventilation (W)	churning losses with ventilation (W)	temperature at stable level (°C)	Tooth (m^2)	Flank (m^2)	wetted surface area (m^2)	Cm	churning losses measured (W)
1	H6965	1000	50%	1	0,0015	64	0,003375	20	0,014	0,096	4,4	4,4	24	0,00355	0,00283	0,00638	0,01243	4,4
2	H6965	2000	50%	1	0,0015	64	0,003375	20	0,014	0,096	12,7	12,7	27,5	0,00355	0,00283	0,00638	0,00449	12,7
3	H6965	3000	50%	1	0,0015	64	0,003375	20	0,014	0,096	20,72	20,72	30,8	0,00355	0,00283	0,00638	0,00217	20,72
4	H6965	5000	50%	1	0,0015	64	0,003375	20	0,014	0,096	27,8	35,6	36,4	0,00355	0,00283	0,00638	0,00080	35,6
5	H6965	6000	50%	1	0,0015	64	0,003375	20	0,014	0,096	23,6	35,4	36,8	0,00355	0,00283	0,00638	0,00046	35,4
6	H6965	7000	50%	1	0,0015	64	0,003375	20	0,014	0,096	25,2	38,4	37,8	0,00355	0,00283	0,00638	0,00032	38,4
7	H6965	1000	25%	2	0,0015	102	0,003375	20	0,014	0,153	8,25	8,25	23,8	0,00391	0,00265	0,00656	0,00560	8,25
8	H6965	2000	25%	2	0,0015	102	0,003375	20	0,014	0,153	9,96	9,96	25,9	0,00391	0,00265	0,00656	0,00085	9,96
9	H6965	3000	25%	2	0,0015	102	0,003375	20	0,014	0,153	14,47	14,47	28,9	0,00391	0,00265	0,00656	0,00036	14,47
10	H6965	5000	25%	2	0,0015	102	0,003375	20	0,014	0,153	25,3	43,1	36	0,00391	0,00265	0,00656	0,00023	43,1
11	H6965	6000	25%	2	0,0015	102	0,003375	20	0,014	0,153	29,8	51,4	39	0,00391	0,00265	0,00656	0,00016	51,4
12	H6965	7000	25%	2	0,0015	102	0,003375	20	0,014	0,153	34,8	60,1	43,1	0,00391	0,00265	0,00656	0,00012	60,1
13	H6965	1000	50%	2	0,0015	102	0,003375	20	0,014	0,153	23,62	23,62	28,1	0,00566	0,00719	0,01285	0,00819	23,62
14	H6965	2000	50%	2	0,0015	102	0,003375	20	0,014	0,153	58,3	58,3	37,1	0,00566	0,00719	0,01285	0,00253	58,3
15	H6965	3000	50%	2	0,0015	102	0,003375	20	0,014	0,153	68,6	68,6	41,6	0,00566	0,00719	0,01285	0,00088	68,6
16	H6965	5000	50%	2	0,0015	102	0,003375	20	0,014	0,153	165	182,8	63,3	0,00566	0,00719	0,01285	0,00051	182,8
17	H6965	6000	50%	2	0,0015	102	0,003375	20	0,014	0,153	210	231,6	72,7	0,00566	0,00719	0,01285	0,00037	231,6
18	H6965	7000	50%	2	0,0015	102	0,003375	20	0,014	0,153	246	271,3	79,1	0,00566	0,00719	0,01285	0,00027	271,3
19	H6965	5000	25%	3	0,003	30	0,00675	20	0,024	0,09	17	31	44,5	0,00394	0,00092	0,00486	0,00112	31
20	H6965	6000	25%	3	0,003	30	0,00675	20	0,024	0,09	30	48	50,7	0,00394	0,00092	0,00486	0,00100	48
21	H6965	7000	25%	3	0,003	30	0,00675	20	0,024	0,09	46	71	53,4	0,00394	0,00092	0,00486	0,00093	71
22	H6965	1000	50%	3	0,003	30	0,00675	20	0,024	0,09	4,49	4,49	24,4	0,00571	0,00249	0,00820	0,01199	4,49
23	H6965	2000	50%	3	0,003	30	0,00675	20	0,024	0,09	17,81	17,81	28	0,00571	0,00249	0,00820	0,00595	17,81
24	H6965	3000	50%	3	0,003	30	0,00675	20	0,024	0,09	27,57	36,52	31,3	0,00571	0,00249	0,00820	0,00361	36,52
25	H6965	5000	50%	3	0,003	30	0,00675	20	0,024	0,09	42	55,99	37,5	0,00571	0,00249	0,00820	0,00120	55,99
26	H6965	6000	50%	3	0,003	30	0,00675	20	0,024	0,09	46	64,66	39,8	0,00571	0,00249	0,00820	0,00080	64,66
27	H6965	7000	50%	3	0,003	30	0,00675	20	0,024	0,09	63	88,18	45,8	0,00571	0,00249	0,00820	0,00069	88,18

Continuation appendix B1

viscosity at 40° (cst)	viscosity at 100° (cst)	viscosity at oil temperature (cst)	A	B	Re	Fr	V0 (m^3)	Theoretical Cm model Macian-Voutay	Theoretical losses Macian Voutay (W)	Measured power loss (W)	Cm new model	Power loss new model (W)	Cm low speed formula	Power loss with low speed formula (W)	Re^x
48	8,3	100,11	-3,28	8,40	2410	54	0,00338	0,00529	1,9	4,4	0,01161	4,1	0,01495	5,3	703,0
48	8,3	84,05	-3,28	8,40	5741	215	0,00338	0,00192	5,4	12,7	0,00343	9,7	0,00542	15,4	1674,6
48	8,3	71,82	-3,28	8,40	10079	483	0,00338	0,00105	10,0	20,7	0,00168	16,0	0,00296	28,3	2939,6
48	8,3	55,87	-3,28	8,40	21593	1342	0,00338	0,00048	21,4	35,6	0,00068	30,2	0,00137	60,5	6298,1
48	8,3	54,91	-3,28	8,40	26362	1932	0,00338	0,00037	28,5	35,4	0,00050	37,9	0,00105	80,5	7689,0
48	8,3	52,62	-3,28	8,40	32097	2630	0,00338	0,00030	36,1	38,4	0,00038	45,9	0,00084	102,0	9361,5
48	8,3	101,14	-3,28	8,40	6060	86	0,00210	0,00334	4,9	8,3	0,00932	13,7	0,00565	8,3	1108,9
48	8,3	90,94	-3,28	8,40	13477	342	0,00210	0,00123	14,5	10,0	0,00275	32,4	0,00208	24,5	2466,4
48	8,3	78,56	-3,28	8,40	23404	770	0,00210	0,00067	26,8	14,5	0,00135	53,6	0,00114	45,3	4283,2
48	8,3	56,84	-3,28	8,40	53906	2139	0,00210	0,00031	56,3	43,1	0,00055	101,0	0,00052	95,3	9865,2
48	8,3	50,03	-3,28	8,40	73496	3080	0,00210	0,00023	73,3	51,4	0,00040	126,6	0,00039	124,0	13450,3
48	8,3	42,36	-3,28	8,40	101279	4192	0,00210	0,00018	90,4	60,1	0,00030	153,3	0,00030	152,9	18534,8
48	8,3	81,64	-3,28	8,40	7507	86	0,00283	0,00653	18,8	23,6	0,00899	25,9	0,00761	21,9	1373,9
48	8,3	54,21	-3,28	8,40	22609	342	0,00283	0,00226	52,0	58,3	0,00266	61,3	0,00263	60,6	4137,7
48	8,3	44,97	-3,28	8,40	40883	770	0,00283	0,00122	95,3	68,6	0,00130	101,3	0,00143	111,0	7481,8
48	8,3	20,93	-3,28	8,40	146406	2139	0,00283	0,00051	182,9	182,8	0,00053	190,8	0,00059	213,0	26793,3
48	8,3	15,92	-3,28	8,40	231035	3080	0,00283	0,00037	230,7	231,6	0,00038	239,3	0,00043	268,7	42280,9
48	8,3	13,42	-3,28	8,40	319591	4192	0,00283	0,00029	284,5	271,3	0,00029	289,6	0,00033	331,3	58487,2
48	8,3	40,10	-3,28	8,40	26443	1258	0,00300	0,00036	10,1	31,0	0,00110	30,4	0,00100	27,9	14103,0
48	8,3	31,81	-3,28	8,40	39994	1812	0,00300	0,00027	12,8	48,0	0,00080	38,2	0,00074	35,5	21330,1
48	8,3	28,92	-3,28	8,40	51323	2466	0,00300	0,00021	16,0	71,0	0,00061	46,2	0,00058	44,4	27372,1
48	8,3	98,08	-3,28	8,40	2162	50	0,00344	0,00993	3,7	4,5	0,01906	7,1	0,01624	6,1	1153,1
48	8,3	82,03	-3,28	8,40	5170	201	0,00344	0,00360	10,8	17,8	0,00563	16,9	0,00589	17,6	2757,4
48	8,3	70,17	-3,28	8,40	9066	453	0,00344	0,00197	19,9	36,5	0,00276	27,9	0,00322	32,5	4835,2
48	8,3	53,29	-3,28	8,40	19895	1258	0,00344	0,00090	42,3	56,0	0,00112	52,5	0,00148	69,1	10610,6
48	8,3	48,40	-3,28	8,40	26290	1812	0,00344	0,00068	55,3	64,7	0,00081	65,8	0,00112	90,5	14021,1
48	8,3	38,14	-3,28	8,40	38920	2466	0,00344	0,00052	67,3	88,2	0,00062	79,7	0,00086	110,0	20757,6

Appendix B2 measurement 28 t/m 50

number of measure ment	used oil	rota tional speed (rpm)	Immersion factor I	geo metry	Module (m)	number of teeth	tooth depth h dent (m)	pressure angle (°)	face width (m)	Diameter (m)	churning losses without ventilation (W)	churning losses with ventilation (W)	temperature at stable level (°C)	Tooth (m^2)	Flank (m^2)	wetted surface area (m^2)	Cm	churning losses measured (W)
28	H6965	1000	25%	4	0,003	53	0,00675	20	0,024	0,159	12,08	12,08	24,4	0,00696	0,00287	0,00983	0,00488	12,08
29	H6965	2000	25%	4	0,003	53	0,00675	20	0,024	0,159	23,09	23,09	28,9	0,00696	0,00287	0,00983	0,00117	23,09
30	H6965	3000	25%	4	0,003	53	0,00675	20	0,024	0,159	37,34	45,90	35	0,00696	0,00287	0,00983	0,00069	45,90
31	H6965	5000	25%	4	0,003	53	0,00675	20	0,024	0,159	157	196,43	63,4	0,00696	0,00287	0,00983	0,00063	196,43
32	H6965	6000	25%	4	0,003	53	0,00675	20	0,024	0,159	214	282,02	76,3	0,00696	0,00287	0,00983	0,00053	282,02
33	H6965	7000	25%	4	0,003	53	0,00675	20	0,024	0,159	283	390,84	89,1	0,00696	0,00287	0,00983	0,00046	390,84
34	H6965	1000	50%	4	0,003	53	0,00675	20	0,024	0,159	29,04	29,04	29,7	0,01009	0,00776	0,01785	0,00646	29,04
35	H6965	2000	50%	4	0,003	53	0,00675	20	0,024	0,159	80,92	80,92	41,2	0,01009	0,00776	0,01785	0,00225	80,92
36	H6965	3000	50%	4	0,003	53	0,00675	20	0,024	0,159	154,97	163,53	57	0,01009	0,00776	0,01785	0,00135	163,53
37	H6965	5000	50%	4	0,003	53	0,00675	20	0,024	0,159	420	459,43	102,7	0,01009	0,00776	0,01785	0,00082	459,43
38	H6965	6000	50%	4	0,003	53	0,00675	20	0,024	0,159	499	567,02	112,8	0,01009	0,00776	0,01785	0,00058	567,02
39	H6965	7000	50%	4	0,003	53	0,00675	20	0,024	0,159	542	649,84	120,9	0,01009	0,00776	0,01785	0,00042	649,84
40	H6965	3000	25%	5	0,005	30	0,01125	20	0,024	0,15	108,321	136,3	67,7	0,00657	0,00255	0,00912	0,00262	136,3
41	H6965	4000	25%	5	0,005	30	0,01125	20	0,024	0,15	139,076	182,7	83,3	0,00657	0,00255	0,00912	0,00148	182,7
42	H6965	5000	25%	5	0,005	30	0,01125	20	0,024	0,15	175,716	245,1	91,5	0,00657	0,00255	0,00912	0,00102	245,1
43	H6965	6000	25%	5	0,005	30	0,01125	20	0,024	0,15	215,068	322,8	101,8	0,00657	0,00255	0,00912	0,00078	322,8
44	H6965	3000	50%	5	0,005	30	0,01125	20	0,024	0,15	140,63	168,43	59,9	0,00952	0,00691	0,01643	0,00180	168,43
45	H6965	4000	50%	5	0,005	30	0,01125	20	0,024	0,15	227,64	270,7	78,9	0,00952	0,00691	0,01643	0,00122	270,7
46	H6965	5000	50%	5	0,005	30	0,01125	20	0,024	0,15	303,9	372,08	90,8	0,00952	0,00691	0,01643	0,00086	372,08
47	H6965	6000	50%	5	0,005	30	0,01125	20	0,024	0,15	434,15	539,8	108,4	0,00952	0,00691	0,01643	0,00072	539,8
48	MG632	5000	50%	2	0,0015	102	0,003375	20	0,014	0,153	171,8	190	67	0,00566	0,00719	0,01285	0,00053	190
49	MG632	6000	50%	2	0,0015	102	0,003375	20	0,014	0,153	212,2	233	77,8	0,00566	0,00719	0,01285	0,00037	233
50	MG632	7000	50%	2	0,0015	102	0,003375	20	0,014	0,153	254,7	280	80,3	0,00566	0,00719	0,01285	0,00028	280

Continuation appendix B2

viscosity at 40° (cst)	viscosity at 100° (cst)	viscosity at oil temperature (cst)	A	B	Re	Fr	V0 (m^3)	Theoretical Cm model Macian-Voutay	Theoretical losses Macian Voutay (W)	Measured power loss (W)	Cm new model	Power loss new model (W)	Cm low speed formula	Power loss with low speed formula (W)	Re^X
48	8,3	98,08	-3,28	8,40	6748	89	0,00201	0,00673	16,7	12,1	0,01457	36,1	0,00532	13,2	2037,1
48	8,3	78,56	-3,28	8,40	16851	356	0,00201	0,00242	47,9	23,1	0,00430	85,2	0,00191	37,8	5087,0
48	8,3	59,38	-3,28	8,40	33437	800	0,00201	0,00129	86,0	45,9	0,00211	140,8	0,00102	67,9	10094,1
48	8,3	20,87	-3,28	8,40	158601	2223	0,00201	0,00050	155,5	196,4	0,00086	265,3	0,00040	122,8	47879,6
48	8,3	14,44	-3,28	8,40	275010	3200	0,00201	0,00036	192,4	282,0	0,00062	332,7	0,00028	151,9	83021,9
48	8,3	10,52	-3,28	8,40	440202	4356	0,00201	0,00027	230,0	390,8	0,00047	402,7	0,00021	181,7	132891,2
48	8,3	75,62	-3,28	8,40	8752	89	0,00277	0,01280	57,5	29,0	0,01395	62,7	0,00710	31,9	2642,1
48	8,3	45,70	-3,28	8,40	28963	356	0,00277	0,00433	155,9	80,9	0,00412	148,2	0,00240	86,5	8743,4
48	8,3	25,60	-3,28	8,40	77570	800	0,00277	0,00217	262,9	163,5	0,00202	245,0	0,00120	145,9	23417,2
48	8,3	7,86	-3,28	8,40	421189	2223	0,00277	0,00082	462,2	459,4	0,00082	461,6	0,00046	256,4	127151,5
48	8,3	6,48	-3,28	8,40	612852	3200	0,00277	0,00061	593,2	567,0	0,00060	578,7	0,00034	329,0	185011,9
48	8,3	5,62	-3,28	8,40	823786	4356	0,00277	0,00048	735,7	649,8	0,00045	700,6	0,00026	408,1	248690,2
48	8,3	18,34	-3,28	8,40	96339	755	0,00214	0,00094	49,0	136,3	0,00215	111,7	0,00086	44,9	30828,4
48	8,3	12,08	-3,28	8,40	195040	1342	0,00214	0,00058	71,0	182,7	0,00129	159,6	0,00053	65,0	62412,7
48	8,3	9,97	-3,28	8,40	295550	2097	0,00214	0,00040	97,2	245,1	0,00087	210,4	0,00037	89,0	94576,0
48	8,3	8,00	-3,28	8,40	441751	3019	0,00214	0,00030	124,0	322,8	0,00063	263,8	0,00027	113,6	141360,3
48	8,3	23,29	-3,28	8,40	75884	755	0,00286	0,00204	191,6	168,4	0,00208	194,8	0,00128	119,7	24282,9
48	8,3	13,49	-3,28	8,40	174632	1342	0,00286	0,00121	269,9	270,7	0,00125	278,3	0,00076	168,6	55882,2
48	8,3	10,12	-3,28	8,40	290921	2097	0,00286	0,00083	362,3	372,1	0,00085	367,1	0,00052	226,3	93094,6
48	8,3	7,03	-3,28	8,40	502680	3019	0,00286	0,00060	448,4	539,8	0,00061	460,2	0,00037	280,1	160857,7
320	24	78,51	-3,36	8,78	39029	2139	0,00283	0,00067	248,3	190,0	0,00053	190,8	0,00078	281,2	7142,6
320	24	50,58	-3,36	8,78	72701	3080	0,00283	0,00047	302,5	233,0	0,00038	239,3	0,00055	342,6	13304,8
320	24	46,05	-3,36	8,78	93164	4192	0,00283	0,00037	379,0	280,0	0,00029	289,6	0,00043	429,2	17049,7