



Faculty of Engineering of the University of Porto -Portugal

Department of Mechanical Engineering and Industrial Management

Applied Mechanic Section

Undergraduate Programme in Mechanical Engineering

5th Year – Machine Design

Final Project

**Experimental Comparison of the Mass Balance System
Performance With Carbonitrided Steel Gears And Austempered
Ductile Iron Gears. (PFC2.2003).**

Author:

Henrique Manuel Garcia Pinto

Supervised by:

Prof. Jorge Seabra (FEUP)

Engº Quê Vu –Do (C.A.C.I.A.)

Engº Pedro Reis (C.A.C.I.A.)

2003/2004

**621(047.3)
LEM 2003/PINh**



Faculty of Engineering of the University of Porto -Portugal

Department of Mechanical Engineering and Industrial Management

Applied Mechanic Section

Undergraduate Programme in Mechanical Engineering

5th Year – Machine Design

Final Project

**Experimental Comparison of the Mass Balance System
Performance With Carbonitrided Steel Gears And Austempered
Ductile Iron Gears. (PFC2.2003).**

Author:

Henrique Manuel Garcia Pinto

Supervised by:

Prof. Jorge Seabra (FEUP)

Engº Quê Vu –Do (C.A.C.I.A.)

Engº Pedro Reis (C.A.C.I.A.)

2003/2004

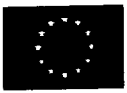
1. The first step in the process is to identify the problem or issue that needs to be addressed. This involves gathering information and understanding the context of the problem.

[illegible][illegible]

1997, 1998, 1999, 2000, 2001, 2002, 2003, 2004, 2005, 2006, 2007, 2008, 2009, 2010, 2011, 2012, 2013, 2014, 2015, 2016, 2017, 2018, 2019, 2020, 2021, 2022, 2023, 2024, 2025, 2026, 2027, 2028, 2029, 2030, 2031, 2032, 2033, 2034, 2035, 2036, 2037, 2038, 2039, 2040, 2041, 2042, 2043, 2044, 2045, 2046, 2047, 2048, 2049, 2050, 2051, 2052, 2053, 2054, 2055, 2056, 2057, 2058, 2059, 2060, 2061, 2062, 2063, 2064, 2065, 2066, 2067, 2068, 2069, 2070, 2071, 2072, 2073, 2074, 2075, 2076, 2077, 2078, 2079, 2080, 2081, 2082, 2083, 2084, 2085, 2086, 2087, 2088, 2089, 2090, 2091, 2092, 2093, 2094, 2095, 2096, 2097, 2098, 2099, 2100, 2101, 2102, 2103, 2104, 2105, 2106, 2107, 2108, 2109, 2110, 2111, 2112, 2113, 2114, 2115, 2116, 2117, 2118, 2119, 2120, 2121, 2122, 2123, 2124, 2125, 2126, 2127, 2128, 2129, 2130, 2131, 2132, 2133, 2134, 2135, 2136, 2137, 2138, 2139, 2140, 2141, 2142, 2143, 2144, 2145, 2146, 2147, 2148, 2149, 2150, 2151, 2152, 2153, 2154, 2155, 2156, 2157, 2158, 2159, 2160, 2161, 2162, 2163, 2164, 2165, 2166, 2167, 2168, 2169, 2170, 2171, 2172, 2173, 2174, 2175, 2176, 2177, 2178, 2179, 2180, 2181, 2182, 2183, 2184, 2185, 2186, 2187, 2188, 2189, 2190, 2191, 2192, 2193, 2194, 2195, 2196, 2197, 2198, 2199, 2200, 2201, 2202, 2203, 2204, 2205, 2206, 2207, 2208, 2209, 2210, 2211, 2212, 2213, 2214, 2215, 2216, 2217, 2218, 2219, 2220, 2221, 2222, 2223, 2224, 2225, 2226, 2227, 2228, 2229, 2230, 2231, 2232, 2233, 2234, 2235, 2236, 2237, 2238, 2239, 2240, 2241, 2242, 2243, 2244, 2245, 2246, 2247, 2248, 2249, 2250, 2251, 2252, 2253, 2254, 2255, 2256, 2257, 2258, 2259, 2260, 2261, 2262, 2263, 2264, 2265, 2266, 2267, 2268, 2269, 2270, 2271, 2272, 2273, 2274, 2275, 2276, 2277, 2278, 2279, 2280, 2281, 2282, 2283, 2284, 2285, 2286, 2287, 2288, 2289, 2290, 2291, 2292, 2293, 2294, 2295, 2296, 2297, 2298, 2299, 2300, 2301, 2302, 2303, 2304, 2305, 2306, 2307, 2308, 2309, 2310, 2311, 2312, 2313, 2314, 2315, 2316, 2317, 2318, 2319, 2320, 2321, 2322, 2323, 2324, 2325, 2326, 2327, 2328, 2329, 2330, 2331, 2332, 2333, 2334, 2335, 2336, 2337, 2338, 2339, 2340, 2341, 2342, 2343, 2344, 2345, 2346, 2347, 2348, 2349, 2350, 2351, 2352, 2353, 2354, 2355, 2356, 2357, 2358, 2359, 2360, 2361, 2362, 2363, 2364, 2365, 2366, 2367, 2368, 2369, 2370, 2371, 2372, 2373, 2374, 2375, 2376, 2377, 2378, 2379, 2380, 2381, 2382, 2383, 2384, 2385, 2386, 2387, 2388, 2389, 2390, 2391, 2392, 2393, 2394, 2395, 2396, 2397, 2398, 2399, 2400, 2401, 2402, 2403, 2404, 2405, 2406, 2407, 2408, 2409, 2410, 2411, 2412, 2413, 2414, 2415, 2416, 2417, 2418, 2419, 2420, 2421, 2422, 2423, 2424, 2425, 2426, 2427, 2428, 2429, 2430, 2431, 2432, 2433, 2434, 2435, 2436, 2437, 2438, 2439, 2440, 2441, 2442, 2443, 2444, 2445, 2446, 2447, 2448, 2449, 2450, 2451, 2452, 2453, 2454, 2455, 2456, 2457, 2458, 2459, 2460, 2461, 2462, 2463, 2464, 2465, 2466, 2467, 2468, 2469, 2470, 2471, 2472, 2473, 2474, 2475, 2476, 2477, 2478, 2479, 2480, 2481, 2482, 2483, 2484, 2485, 2486, 2487, 2488, 2489, 2490, 2491, 2492, 2493, 2494, 2495, 2496, 2497, 2498, 2499, 2500, 2501, 2502, 2503, 2504, 2505, 2506, 2507, 2508, 2509, 2510, 2511, 2512, 2513, 2514, 2515, 2516, 2517, 2518, 2519, 2520, 2521, 2522, 2523, 2524, 2525, 2526, 2527, 2528, 2529, 2530, 2531, 2532, 2533, 2534, 2535, 2536, 2537, 2538, 2539, 2540, 2541, 2542, 2543, 2544, 2545, 2546, 2547, 2548, 2549, 2550, 2551, 2552, 2553, 2554, 2555, 2556, 2557, 2558, 2559, 2560, 2561, 2562, 2563, 2564, 2565, 2566, 2567, 2568, 2569, 2570, 2571, 2572, 2573, 2574, 2575, 2576, 2577, 2578, 2579, 2580, 2581, 2582, 2583, 2584, 2585, 2586, 2587, 2588, 2589, 2590, 2591, 2592, 2593, 2594, 2595, 2596, 2597, 2598, 2599, 2600, 2601, 2602, 2603, 2604, 2605, 2606, 2607, 2608, 2609, 2610, 2611, 2612, 2613, 2614, 2615, 2616, 2617, 2618, 2619, 2620, 2621, 2622, 2623, 2624, 2625, 2626, 2627, 2628, 2629, 2630, 2631, 2632, 2633, 2634, 2635, 2636, 2637, 2638, 2639, 2640, 2641, 2642, 2643, 2644, 2645, 2646, 2647, 2648, 2649, 2650, 2651, 2652, 2653, 2654, 2655, 2656, 2657, 2658, 2659, 2660, 2661, 2662, 2663, 2664, 2665, 2666, 2667, 2668, 2669, 2670, 2671, 2672, 2673, 2674, 2675, 2676, 2677, 2678, 26

621 (047.3)/LEM 2003/PINA

Universidade do Porto
Faculdade de Engenharia
Biblioteca
Nº 87311
CDU _____
Data 23/04/2007



FINAL PROJECT

Experimental Comparison of the Mass Balance System
performance With Carbonitrided Steel Gears and
Austempered Ductile Iron Gears



Acknowledgements

Firstly, I would like to thank Professor Jorge Seabra for his guidance and patience through the project.

Many thanks to Gil Gonçalves and Ricardo Afonso.

Special thanks to Mr. Quê Vu –Do, for his support and for sharing his knowledge with me

Others thanks:

Engineers: Pedro Reis, Rui Amaro, Pedro Rodrigo, António Teixeira, Nuno Merendeiro, Laurence Vieira, Jorge Pereira, Pedro Ribeiro, Carlos Freitas, Marco Messias and João Merendeiro.

Operators: Paulo Pereira and Bruno.

Others: Virgilio Santos, Agostinho Carvalho, Manuel Alves, Victor Amaro



Contents

1. Introduction	5
2. Objectives.....	6
3. Mass Balance System.....	7
4. Materials Characterisation.....	8
4.1. Carbonitrided Steel.....	8
4.2. ADI Characterization	8
4.2.1. Base Material.....	8
4.2.2. ADI production	9
4.2.3. Mechanical Properties	10
4.2.4. ADI Gear Deterioration.....	11
4.2.5. Comparative Advantages of ADI.....	13
5. Tolerance of the Production in Gears.....	15
5.1. Geometry of wheels in external gears with involute profile	15
5.2. Precision system of gears with involute teeth	15
5.3. Addendum modification in wheels of external gears and in wheels of speed increasing gears	16
5.4. Applying standard precision systems to specify teeth backlash in wheels of external gears	17
6. Materials' characteristics	19
6.1. Austempered Ductile Iron (ADI)	19
6.2. Steel – 36 NCD 16 (AFNOR)	20
7. Gear Design and Production	21
7.1. Geometric Characterisation.....	22
7.1.1. M1D Mass Balance System	22
7.1.2. G9T Mass Balance System	24
7.2. Gears and Metal Sleeve Designs.....	27
7.2.1. M1D Mass Balance System Gears	27
7.2.2. G9T Mass Balance System Gears	27
8. Production of gears.....	29
8.1. Turning (pinions G9T and M1D).	29
8.2. Gears G9T	30
8.2.1. Hobbing (G9T).....	30
8.2.2. KLINGELNBERG Control – G9T gears after hobbing.....	32
8.2.3. Heat treatment (gears G9T).....	33
8.2.4. KLINGELNBERG Control - after heat treatment (gears G9T).....	33
8.2.5. Hard turning and grinding (gears G9T).....	33
8.3. Gears M1D	35
8.3.1. Hobbing pinions M1D.....	35



8.3.2. KLINGELNBERG control (gears M1D)	36
8.3.3. Hard Turning and Grinding (gears M1D)	36
8.4. Final conclusions	36
9. Gears assembly	38
9.1. G9T	38
9.2. M1D	39
10. Experimental tests	40
10.1. MBS G9T ORMO tests	40
10.1.1. Tests: preparation and realisation	42
10.1.2. Results analysis	43
10.2. MBS M1D ORMO tests	45
10.2.1. Tests: preparation and realisation	46
10.2.2. Results analysis	47
10.3. Tests conclusions	48
11. AMESim	49
11.1. Models Construction	49
11.2. Require Values	52
11.3. Results	53
11.4. Simulation conclusions	62
12. Final conclusions	63
13 Further work	64
14. Bibliography	65
Appendix A	66
Appendix B	75
Appendix C	81
C.1. Dimensional control after turning	82
Appendix D	82
D.1. M-file to calculate dimension over ball	82
D.2. Approximate expression to calculate dimension under ball	82
Appendix E	82
E.1. Dimensional control	82
Gears G9T	82
Gears M1D	82
E.2. Dimensional control (graphics) of the G9T.gears	82
E.2. Dimensional control (graphics) of the M1D.gears	82
Appendix F	82



Appendix G	82
G9T	82
M1D	82
Appendix H	82
G.1. Inertias	82
G.2. Stiffness	82
G.3. Mesh stiffness	82
G.4. E-mail from AMESim about Mesh Damping.	82



1. Introduction

Austempered Ductile Iron (ADI) is a metal that is going to be used as a substitute to steel in certain mechanical engineering applications. Depending on what application ADI is used, it can bring some advantages. However, to carry out this replacement it is necessary to conduct tests to guarantee that ADI is safe when thus applied.

In this project the application of ADI in gears of the mass balance systems (MBS) will be verified.

As the case under scrutiny certain properties play a major role in attaining good system performance. A mass balance system (MBS) is used to reduce vibration and consequently engine-related noise. The material applied, to the gears in this case, has to exhibit the ability to absorb a given quantity of engine-caused vibration.

To try verifying this ability of the ADI it will be doing experimental tests, with steel gears and ADI gears and both materials will be compared.

The experimental tests will be confronted with of AMESim simulation.

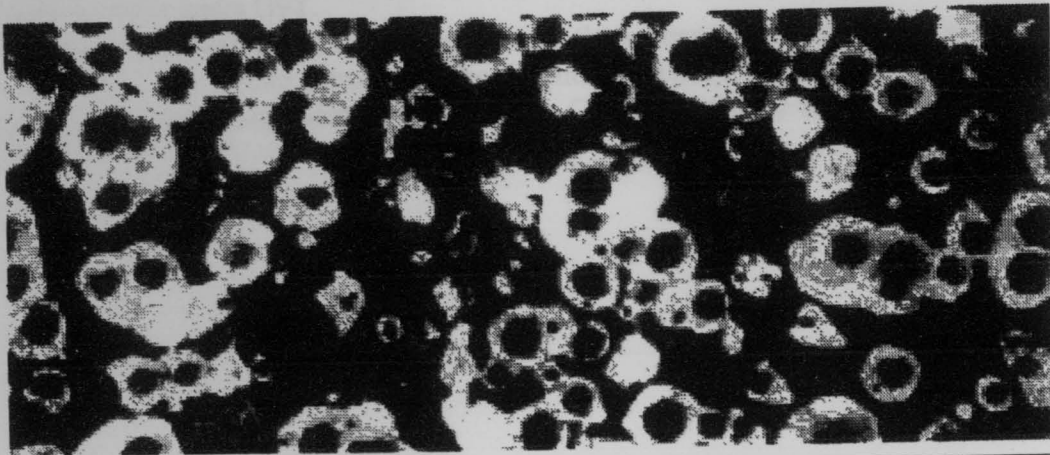
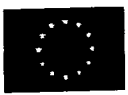


Figure 1 – Nodular graphite cast iron with spheroidal graphite.



2. Objectives

This project's objective is the experimental comparison of the mass balance system performance with carbonitrided steel gears and austempered ductile iron gears.

Here are the various objectives suggested:

- Material characterisation;
- Research into literature on tolerances in gear production;
- Working principle of mass balance system;
- Geometric characterisation;
- Drawings to manufacture the wheels in initial form;
- Production of gears;
- Assembly;
- Make a experimental tests;

However, to carry out the production of gears will be necessary to take some careful, principally with the gears G9T.

At same time will be doing a simulation in the software AMESim. Principal objective of this simulation is to create a model that represents the reality, for in the future studies about the application of the other materials in the MBS. Consequently the simulation results will be compared with the results of experimental tests.



3. Mass Balance System

The mass balance system is a mechanism that will balance 2nd order inertia forces. These forces are generated because of the push rod movement in the internal combustion engine. The movement of the push rod is twice as faster as the crankshaft's. While the crankshaft makes a half turn, the push rod moves from its initial position to a new one and its mass creates a considerable inertia force. These forces are the 2nd order inertia forces.

It is hard to devise a system as close as possible to the ideal one, as one must achieve synchronization between the movement of the crankshaft and that of the mass balance system.

Theoretically, the mass balance system is to make the alternating inertia forces ineffective; in practice though this is a much harder task, since it is difficult to obtain movement synchronization entailing complete ineffectiveness. Nevertheless, when such phenomenon indeed takes place, engine-related vibration and noise are markedly reduced. An engine including this system will hence make car users feel much more comfortable. As this system leads to an increase, even if minimal, in consumption, it is more often applied to high piston displacement engines.

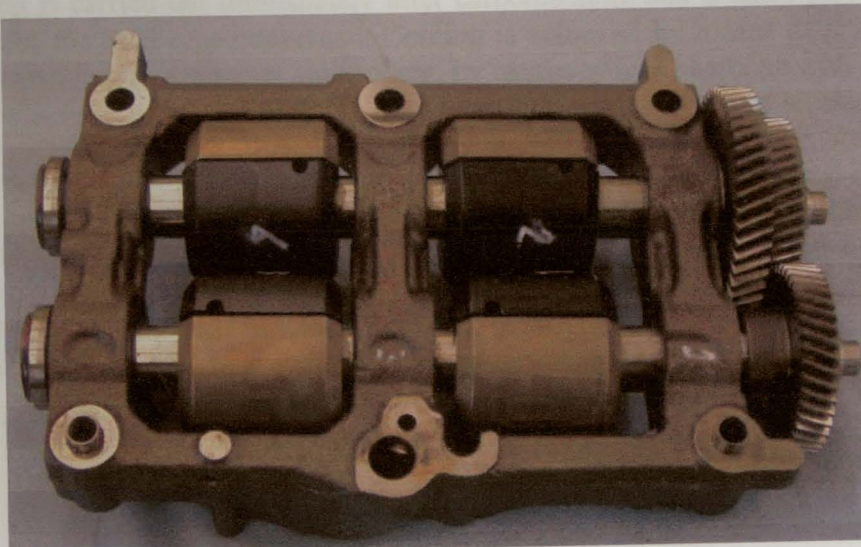


Figure 2 – G9T Mass balance system.



4. Materials Characterisation

4.1. Carbonitrided Steel¹

When the carbonitriding heat treatment is applied to steel carbonitrided steel is obtained.

Carbonitriding combines carbonising and nitriding: very hard surface layers about 0,1 mm depth are thus obtained. This treatment may be applied to any steel.

During carbonitriding carbon and nitrogen are absorbed into the steel's surface. Ensuing nitrogen combines with steel and forms nitrides just as it happens in nitriding. Due to nitrogen absorption transition $\alpha \rightarrow \gamma$ reaches relatively low temperatures – austenite forms about 600° to 750° C. At the same time carbon (dissolves in γ -iron) absorption is favoured so that during cooling there is enough carbon to form martensite. There are two types of carbonitriding taking into consideration transition $\alpha \rightarrow \gamma$ of the workpiece's core. Consequently, there is carbonitriding involving or not transition $\alpha \rightarrow \gamma$ of the core; such transition solely depends on temperature, except for small-sized parts: diffusion is uniform.

Cooling is carried out immediately after carbonitriding and it takes place in the right environment for the steel type concerned. Cooling is achieved by means of nitrogenous salt or, as to carbonitriding in a gaseous atmosphere, by means of hydrocarbons with ammonia.

Carbonising is a treatment applied especially to ferritic-pearlitic steels. The surface layer of the workpiece is carburised: the piece is wrapped in a carbonising atmosphere and heated at a temperature between 870 and 930° C for a given period depending on the type of carbonising atmosphere. The carbonising atmosphere thus passes activated carbon down to the workpiece's surface. The longer the treatment lasts, the higher the depths of carburised surface will be.

Nitriding is a treatment in which nitrogen, in the form of iron nitrides, is introduced into the surface layer of steel hence obtaining great surface hardness.

This process is applied when you wish to obtain maximum hardness or great wear resistance. Volume swell of the nitrided layer gives rise to compressive residual stress that considerably increases fatigue resistance.

4.2. ADI Characterization²

4.2.1. Base Material

ADI is produced from cast iron containing spheroidal graphite (SG iron).

Graphite is spheroidized before the metal is poured by adding an element that contains Mg and rare earths. This material's metallic matrix may range from totally ferritic (17 %

¹ See «Aços – características, tratamentos», Pinto Soares.

² See «Resistência ao desgaste e gripagem de engrenagens em ferro fundido nodular austemperado» - Thesis presented to the Engineering Faculty of Oporto University for a master's degree in Mechanical Engineering – Luis Manuel Martins Leite de Magalhães.



elongation at break) to totally pearlitic (in this case cast iron exhibits greater mechanical strength, up to 800 MPa ultimate stress; its elongation at break is though not higher than 2 per cent).

By means of the austempering heat treatment SG iron's structure is transformed to contain acicular ferrite and austenite; a material with high mechanical properties is thus obtained, its ultimate stress can be between 850 and 1400 MPa and it exhibits excellent wear resistance – the said material is ADI.

Its properties, such as hardenability, are improved adding alloy elements such as copper, manganese, molybdenum and nickel.

Cast iron involves a relatively cheap production process; such castings are of the same or higher quality than steels. The properties of the base material (such as the type of structure, graphite quantity and distribution and type of alloying elements present) are determinant of the final product's quality.

4.2.2. ADI production

ADI is produced by employing a thermal treatment on SG iron – austempering.

Ensuing characteristics depend on the properties of the base material and on the parameters followed during treatment. It is hence important to control parameters such as temperatures and times linked to each one of the stages in the austempering heat treatment.

Such heat treatment comprises three stages:

- austenisation treatment (or solution);
- abrupt cooling followed by a stage in which austenite is isothermally transformed;
- cooling until ambient temperature.

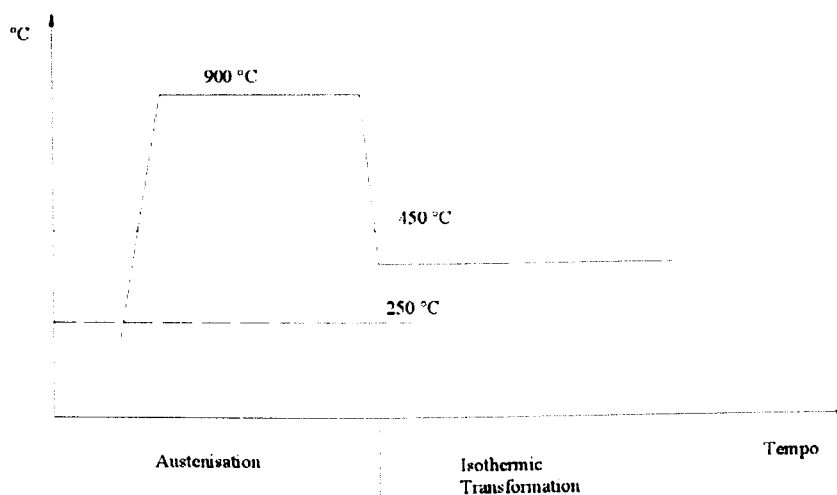


Figure 3 - Heat treatment scheme.

During austenisation the material is heated for a period between 1-4 hours at a temperature between 850-950°C. Time and temperature vary with the base material and expected properties. Carbon homogenizing in the austenite takes place; grain may grow in



size, while carbons dissolve in the material and microsegregation of the eventually present alloying elements decreases.

Prior to isothermal transformation there is an abrupt cooling down to a temperature between 230–450°C, which is kept for ½-5 hours. The final objective is to allow austenite to isothermally transform to bainite. The time required depends both on the quantity of austenite you want to transform and on the quantity of alloying elements present. This transformation plays a decisive role in shaping the material's final properties.

At this stage there are two ranges of temperature that result in obtaining different materials in terms of mechanical properties, for different structures are formed. Consequently, ADI is regarded as a “family” of materials, since the conditions each treatment takes place in lead to materials with quite different properties.

Alloying elements (manganese, molybdenum, copper, nickel) are added to improve the material's hardenability as well as its mechanical properties such as toughness and elongation.

4.2.3. Mechanical Properties

ADI's mechanical properties vary with: the base material's quality, alloying elements, the type of thermal process, mechanical treatments applied to the material and the workpiece's size.

ADI offers mechanical properties inferior to carbonised steel, but it exhibits better performance as to wear resistance than alloy and tempered steels and pearlitic SG iron.

In regard to fatigue resistance ADI that has been applied “shot-peening” exhibits a performance close to that of carbonised steel: “shot-peening”, mostly carried out due to existing residual (compressive) stresses and primarily affecting teeth in gears, favours the spread of microcracks. Such phenomenon has major effects on fatigue resistance, namely flex fatigue resistance.

Its ability to resist contact fatigue is only inferior to that of nitrided or carbonised steels. Its bending fatigue resistance is superior to that of steels. Its performance as to bending and contact fatigue is that of a high resistance material; it may even exhibit performance superior to that of the most used high resistance steels.

As far as mechanical treatments are concerned, “shot-peening” is applied to ADI. This treatment improves the material's resistance, namely bending strength. With regard to surface wear resistance there are no conclusive results.

“Shot-peening” has beneficial effects when it leads to the transformation of austenite to martensite on the surface layer of the workpiece under treatment; it has harmful effects when it increases the surface roughness, which may be reduced by means of surface trimming.

When a component made from ADI is used its surface hardens between 500 and 900 HV1; the depth concerned also varies. As regards to 900 HV1 hardness it only affects as far as some microns underneath the surface.

SG iron exhibits good machinability, as it contains graphite: energy is thus spared and the tool life expanded.

Such property changes after austempering – it becomes a hard-to-machine material. Consequently, the project has to be carefully set up so as to obtain pieces as similar to the final version as possible hence preventing post-treatment machining.



It is thus important to stress that IG iron grows about 0,3 per cent in volume during thermal treatment; it may even rise to a maximum of 0,5 per cent.

ADI exhibits greater vibration absorption than steel, mostly due to the presence of graphite. Therefore, gears work more silently. ADI gears produce 6 dB less than steels.

Table 1 - Mechanical properties of the forged steel, nodular pearlitic iron and ADI (ASTM 150/100/7).

Material	σ_r (MPa)	$\sigma_{0,2}$ (MPa)	Rupture elongation (%)	Impact resistance (J)	Hardness (BHV)
Forged Steel	790	520	10	175	262
Nodular pearlitic iron	690	480	3	55	262
ADI (ASTM 150/100/7)	1100	830	10	165	286
ADI*	1433	1178	6	73	(414) [HV]

*ADI used for this study.

4.2.4. ADI Gear Deterioration

Nowadays gear wear is calculated with an acceptable margin of error. Nevertheless, there are situations causing gear to wear. Here are some examples:

- real working conditions are different from the ones projected;
- load abnormally distributed due to an incorrect tooth profile;
- residual stresses;
- inadequate lubrication.

The main wear mechanisms are:

- Bending fatigue – cracks are formed at the tooth root;
- Contact fatigue - ‘pitting’ (pits or micro-fish scaling);
- Adhesion – micro-weldings are formed.

Bending fatigue

When a project is carried out properly this type of fault can be avoided. Under moderate load ADI exhibits excellent performance; under heavy load only the performance of carbonised steels is superior. ADI’s composition conditions its resistance to this type of load.



Contact fatigue

As it contains a significant number of graphite nodules, ADI has a higher tendency to micro-crack formation; these micro-cracks become a centre of stress. When they spread they conduce to delamination; this fact must be taken into consideration when great pressure contacts and great sliding are involved. Volume increase during the transformation of austenite to martensite is, on the other hand, beneficial as it retards the spread of micro-cracks. The carbon that has been released from the graphite nodules due to superficial wear is lubricating; the area that has thus been emptied will later become a place of settling and accumulation of lubricant.

ADI exhibits good resistance to contact fatigue, only being inferior to high hardness steels, namely carbonised.

Adhesion

Adhesion occurs when there is insufficient lubrication hence leading the surfaces or top of any irregularity to touch and thus permitting local material welding and ensuing delamination in the said areas. This mechanism is further worsened if there is increase in contact load, sliding speed, temperature and superficial roughness. But it is ameliorated if you rather increase the lubricant's viscosity, rolling speed, surface hardness and chemical activity in the surrounding environment.

Equal materials tend to adhere. In void when their surfaces are completely clean, almost all materials adhere to each other just by being leaned against.

Adhesion in insoluble materials or materials that are different from each other is not caused by material welding and consequent material delamination. In this case both surfaces are delaminated and the ensuing particles remain between the surfaces; if they are not removed seizing may result. As to gear seizing there is cold seizing relative to low speed and hot seizing relative to gears working at high speed.

This process may be controlled by using lubricants and being particularly cautious when selecting the materials.

ADI gear wear

ADI gear exhibits resistance to adhesion-caused wear similar to that of hardened steels. During tests using low additive oils it exhibited a performance similar to that of carbonised steels.

Predicting failures in gears

The study of these components' life is mainly based on the lubrication regime, the lubricants, materials and their surfaces, the temperatures involved and mechanical requirements.

Seizing is the chief mechanism leading to gear wear and deterioration resulting in the destruction of the said components. Superficial wear caused by fatigue is one of the factors that may conduce to new processes taking place and eventually ending in gear seizing.



Seizing criteria

As to tribology, and for the above-mentioned reasons, gear deterioration and failure is predicted taking into consideration thickness of lubricating film and temperature arising from contact. Seizing displays certain features typical of this phenomenon.

Lubrication regime

There are three types of elastohydrodynamic lubrication:

- Full fluid film lubrication
- Mixed film lubrication
- Boundary lubrication

When pressure equals or is higher than 0.5 GPa we shall speak of elastohydrodynamic lubrication.

Of the existing seizing criteria there are two standards (ISO/DIS 6336/4): the flash temperature criterion and the integral temperature criterion. Their results show reasonable margins of error. Both criteria rest on the temperature arising from contact, since it has been determined that temperature associated with seizing is approximately the same in a given type of gear when it works using a certain type of lubricant.

The flash temperature criterion is founded on temperature variation across the whole line of contact of the gear, that is, across the whole active flank.

The integral temperature criterion makes use of an average figure to calculate flash temperature.

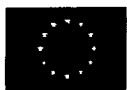
4.2.5. Comparative Advantages of ADI

For ADI is produced from cast iron (an easy-to-found material) the production process of several components can be reassessed. Casting hence plays a fundamental role in machining, as it reduces production costs.

ADI offers mechanical strength similar to that of most wrought or tempered alloy steels. In some cases it can even replace the steels guaranteeing longer component life without failures.

The fact that ADI production is based on a relatively short cycle of heat treatments has been of great influence on reducing production costs: less energy is consumed than that required to thermally treat conventional gear steels. This material may be retempered; it is almost impossible to apply such a process to steels.

ADI evidences a low specific weight rate (10 per cent lighter than steel) and excellent wear resistance (both in dry and lubricated contacts). Furthermore, it exhibits lower superficial wear rates than steels the same hardness (ADI - 0.1 mg/kWh and 0.2 mg/kWh – steels). Superficial roughness on this material quickly disappears after the gear starts working – it is thus not necessary to carry out so thorough a grinding as that applied to superficially hardened steels.



Its wear resistance enables this material to endure extreme lack of lubrication. It presents low friction coefficient hence making it possible to use lubricants of lower quality. When it is working it is even more effective than conventional steels, particularly at low temperatures.

It also shows less notch sensitivity than conventional steels; consequently, it can endure small machining irregularities on the teeth root.

ADI displays great vibration absorbing properties – gears thus work more silently than they do when made from conventional steels.

ADI production does not entail particularly expensive or complex equipment. Technology involved in the said production has been being developed: nowadays it is easy to produce significantly reliable ADI components.

ADI is applied to such components as train wheels, hydraulic pumps, crankshafts and gears – the components are either aimed at resisting wear or are structural components.

Taking into consideration the objective of this project, the most important characteristic, of the ADI, is the great absorbing properties. Nevertheless properties like less notch sensitivity and low friction coefficient are important.



5. Tolerance of the Production in Gears

5.1. Geometry of wheels in external gears with involute profile³

Currently, cogwheel resistance can be calculated by using powerful computer equipment with good graphical functionalities and the codes of the Finite Elements Method (FEM) and the Border Elements Method (BEM). A quite complete analysis of stress on teeth roots is thus carried out. Such analysis cannot though take place without correctly defining the geometry of the tooth's profile.

The geometrical analysis of profiles is developed on a transverse section of a gear (perpendicular to the wheel's axis); it permits a unified assessment of straight and helical teeth and it includes the effects of addendum modification and trimming of protuberances on tooth tops by using pre-shaving tools.

5.2. Precision system of gears with involute teeth⁴

Application of radial mesh Control

Gear system quality is very dependent on the precision of the production processes involved: the former is necessary to guarantee the intended functional characteristics as to working speed, service life, working sound levels and load capacity.

A gear system may contain several types of deviation stemming from different origins and having been caused at different stages of the production process. A gear is produced in the following logical order:

- machining of the body of each wheel;
- shaping teeth in each wheel;
- setting up both cogwheels to start working.

A gear quality control system shall thus focus on:

- control of the body of each wheel;
- control of the teeth of each wheel;
- control of setting-up conditions of the gear.

The complete definition of every standardized precision criterion is contained in the several existing standards.

Controlling the body of each wheel entails checking:

- the dimensions and geometry of the tapping hole or shaft acting as the datum axis;

³ See «Robótica e Automatização», N.º 8, June, 1992, José António dos Santos Almacinha; Pedro de Campos Faria; Joaquim Oliveira Fonseca, Jorge Humberto Oliveira Seabra.

⁴ See «Tecnometal», N.º 82, September and October, 1992 José António dos Santos Almacinha; Jorge Humberto Oliveira Seabra.



- the diameter of the head of a tip cylinder as well as its rotation runout (radial runout);
- rotation runout and warpage (axial runout) of reference surfaces.

In regard to the control of teeth there are precision criteria that may be divided into individual deviations (refer to isolated tooth characteristics) and composite deviations (measuring results from several individual deviations).

Individual deviations may be:

- pitch deviation – caused by the cutting machine;
- profile deviations – total profile deviation can be sorted into a shape deviation and profile slope deviation;
- distortion deviations – total distortion deviations can be sorted into a helix shape deviation and helix slope deviation;
- runout – tooth eccentricity is double as to the wheel's axis when measured perpendicularly to the axis and approximately half way up the tooth; it also refers to irregular tooth backlash resulting from pitch deviations and flank deviations (both tooth flanks);
- tooth thickness tolerance.

Composite deviations may be measured by means of the control of tangential mesh (on one flank) and radial mesh (on both flanks). By controlling these deviations you can ascertain the combined effect of individual geometrical deviations in two working wheels. These deviations may be ascribed to only one of the wheels, if the wheel tested is meshing with a standard wheel. Composite deviation evaluation methods are often used to control gear production.

Mesh control can involve checking:

- distance deviations;
- axis parallelism deviation;
- tooth backlash;
- composite gear deviation.

There are precision standards, which may vary with the standard's origin or its author, prescribing precision types, production processes and their respective applications. The said standards must be regarded as mere guiding lines: as regards to special applications and after producer and customer have agreed so different types of precision may be adopted [3].

5.3. Addendum modification in wheels of external gears and in wheels of speed increasing gears⁵

Although teeth can be of several profiles, production reasons have led to the widespread use of involute profile. The said profile permits perfect functioning even when centre distance varies, while other profiles require normal distance. Involute profile gears

⁵ See «Tecnometal», N.º 78, January and February 1992, José António dos Santos Almacinha, Jorge Humberto Oliveira Seabra.



make it possible to use standard tools to cut wheels with non modified tooth profile, called standard wheels with standard proportion teeth, or wheels with modified tooth profile, the so-called wheels with modified addendum.

Standard teeth are of standard proportion when the reference cutting line of the cutting rack coincides with its datum line during manufacture; otherwise, they are modified.

Centre distance is standard if the teeth in both wheels are of standard proportion. If they have been modified there may be or not distance variation depending on the type of modification.

Addendum modification is positive when the reference line of the cutting rack is exterior to the reference cutting circle of the wheel; otherwise, it is negative.

When teeth are modified it must be borne in mind that such an operation is to change certain important gear characteristics such as contact ratio, specific sliding, gear load capacity and teeth backlash. Nevertheless, when modifying teeth you can improve some of these characteristics and neglect others, which are not as important as the former to the said application.

There are several addendum modification methods conducing to different results: they improve gear characteristics in different ways.

Gears aimed for special applications shall undergo a specific treatment that takes into consideration the characteristics indispensable for them to work properly.

5.4. Applying standard precision systems to specify teeth backlash in wheels of external gears⁶

Dimensional control of tooth thickness

Inevitably, there are differences in gear dimension between the actual model and design projections. On the other hand, while gears are being used, certain dimensions, which are important for them to work properly (centre distance, for instance), may also change. Nonetheless, it must be ensured that the gears will work within certain parameters even if the said dimensional differences indeed occur.

Tooth backlash is defined differently in standard ISO 1328 and standard DIN 3967.

Under standard ISO 1328 there may be a circular or a normal tooth backlash: they are interrelated. Standard DIN 3967 defines the ideal circular tooth backlash in the gear transverse section: it does so by analysing deviations from normal tooth thickness and centre distance deviations.

When specifying nominal figures in draft backlash you can base your projection on previous experience or reference figures in specific literature.

Backlash variation is greatly dependent on tooth thickness tolerance and distance tolerance. Calculation of backlash, tolerance and deviations from tooth thickness has been standardized in the various standards.

Dimension control of tooth thickness may be carried out by means of several methods such as the span measurement over k teeth and the diametrical measurement over spheres or

⁶ «Tecnometal», N.º 79, March and April, 1992, José António dos Santos Almacinha, Jorge Humberto Oliveira Seabra.



rollers. These methods are widely applied during wheel production and control of wheel quality. They are used to indirectly measure tooth thickness; there are mathematical expressions to help us determine actual tooth thickness as to their nominal section.

Tooth backlash results not only from a right choice of deviation from normal tooth thickness in gear wheels but also from the choice of deviation from centre distance and, finally, the choice of tolerance of other individual deviations. Tooth backlash must be calculated as accurately as possible when concerning existing applications.



6. Materials' characteristics

The characteristics of the materials used to make the detachable metal sleeves and the wheels are presented follow.

6.1. Austempered Ductile Iron (ADI)

Table 2 – Nominal chemical composition (weight % + 10 %) of the nodular iron.

C	Si	Cu	Mn	P	S	Cr	Mo	Ni	Mg	Others
3.52	2.39									-

Annealing (preparation for the austempering heat treatment)

Annealing 1000 °C time – 00:30 h;
 Cooling very slow in furnace

Table 3 – Mechanical properties of the nodular iron after annealing.

Tensile strength (N/mm ²)	Yield strength (Rp0.2,N/mm ²)	Elongation (%)	Impact strength (J)	Hardness (HV)
712	427	9	39	226

Austempering heat treatment

Austenization 875 °C;
 Transfer to the salt bath time - < 5 seconds;
 Temperature of isothermal transformation 300 °C;
 Cooling in water;

Table 4 – Mechanical properties of the austempered ductile iron (ADI) after heat treatment.

Tensile strength (N/mm ²)	Yield strength (Rp0.2,N/mm ²)	Elongation (%)	Impact strength (J)	Hardness (HV)
1433	1178	6	73	414



6.2. Steel – 36 NCD 16 (AFNOR)

Table 5 – Nominal chemical composition.

C	Si	Mn	Cr	Mo	Ni
0,36	$\leq 0,40$	0,45	1,8	0,35	3,9

This material it was supply with annealing heat treatment.

Table 6 – Hardness after annealing.

Hardness (HV)
226

To increase the hardness this steel was submitted at one hardening in vacuum.

Table 7 – Hardness after heat treatment.

Hardness (HV)
414



7. Gear Design and Production

Two mass balance systems shall be object of study. They are M1D and G9T.

When the G9T mass balance system was to be produced, a problem with cast iron wheels was found. The central hole in these wheels was larger than necessary. Faced with this problem it was decided to make metal sleeves to obtain the necessary internal diameter.

To produce the gears it was first necessary to find out which production process CACIA uses. Various solutions were suggested to solve this problem. Finally, it was decided to produce detachable sleeves that are going to be introduced in the gears thus providing the necessary diameter to the hole. Metal sleeves were designed by taking into consideration the wheel's process of manufacture at CACIA and the process to transform cast iron into ADI.

Given the gear production process at CACIA and the production process necessary to transform cast iron into ADI, the following operations were performed:

1. FEUP:
 - Wheels manufacture with intermediate dimensions.
 - Metal sleeves manufacture.
 - Assembly of the parts process.
2. CACIA:
 - Wheels and cog manufacture.
3. FEUP:
 - Disassembly the metal sleeves.
 - Gears and metal sleeves heat treatment.
 - Assembly of the parts.
4. CACIA:
 - Rectification of gears.
 - Gears assembly in the shafts.

7.1. Geometric Characterisation

Tables containing the characteristics of the gears of M1D and G9T mass balance systems shall now be presented.

7.1.1. M1D Mass Balance System

Table 8 - Cylindrical Gear Par Cutting Table of the M1D mass balance system.

Cylindrical Gear Par Cutting Table

Gear number	8200385103	8200385231	8200385236
Operating centre distance	74	112	74
Number of gear teeth	1,45	1,4	1,45
Number of teeth	47	49	47
Normal module	1,45	1,4	1,45
Normal pressure angle	20	20	20
Normal helix angle	27,5°	27,5°	27,5°
Helix direction	right	left	left
Reference pitch diameter	76,831	77,338	76,831
Profile theoretical displacement without backlash at radius	-1,24	-0,986	-1,24
Profile maximum finishing displacement at radius	-1,269	-0,992	-1,269
Profile finished minimum displacement at radius	-1,283	-1,045	-1,283



FINAL PROJECT

Experimental Comparison of the Mass Balance System
performance With Carbonitrided Steel Gears and
Austempered Ductile Iron Gears



Universidade do Porto
Faculdade de Engenharia
FEUP



CACTA



RENAULT

Normal base pitch	4,2806	4,33	4,2806
Base helix angle	25,715556°	25,715556°	25,715556°
Base diameter	71,08	71,549	71,08
Tip outside diameter	0,0 78,18 -0,270	0,0 78,97 -0,270	0,0 78,18 -0,270
Tip maximum active diameter	77,78	78,57	77,78
Root maximum theoretical	71,72	72,49	71,72
Root maximum active diameter	71,54	73,23	71,54
Root or inside diameter	0,0 68,44 -0,270	0,0 69,55 -0,270	0,0 68,44 -0,270
Test ball diameter	3	2,6	3
Dimension over 2 balls	-0,057 79,82 -0,087	-0,01 79,48 -0,120	-0,057 79,82 -0,087
Number of teeth for span measurement	k=6	k=6	k=6
Span measurement of k teeth	-0,02 24,04 -0,030	0 23,41 -0,040	-0,02 24,04 -0,030
Base apparent thickness ratio	1	1,1	1
Base transverse thickness (theoretical without backlash)	2,924	3,044	2,924
Operating apparent pressure angle	16,1498	16,6151	16,1498
Operating pitch diameter	74	74,67	74
Operating helix angle on the pitch diameter	26,6885°	26,6833°	26,6885°
Minimum profile contact ratio	2,32	2,27	2,32
Minimum helix contact ratio	1,22	1,25	1,22
Minimum total contact ratio	3,53	3,53	3,53

Checking centre distance	73,88 a 73,92		111,84 a 111,98		73,88 a 73,92	
	After assembly	Before assembly	After assembly	Before assembly	After assembly	Before assembly
Quality as per standard 01-33-001\ -G	8	7	8	7	8	7
Runout(F_r)	7	6	7	6	7	6
Total cumulative pitch deviation (F_p)	7	6	7	6	7	6
Total profile deviation (F_α)	$0 \pm 12 \mu\text{m}$	7	$0 \pm 12 \mu\text{m}$	7	$0 \pm 12 \mu\text{m}$	7
Total helix deviation (F_β)	$0 \pm 15 \mu\text{m}$	7	$0 \pm 15 \mu\text{m}$	7	$0 \pm 15 \mu\text{m}$	7
C_α	$5 \pm 3,5 \mu\text{m}$	$4 \pm 2,5 \mu\text{m}$	$5 \pm 3,5 \mu\text{m}$	$4 \pm 2,5 \mu\text{m}$	$5 \pm 3,5 \mu\text{m}$	$4 \pm 2,5 \mu\text{m}$
C_β	$4,5 \pm 4 \mu\text{m}$	$3,5 \pm 3 \mu\text{m}$	$4,5 \pm 4 \mu\text{m}$	$3,5 \pm 3 \mu\text{m}$	$4,5 \pm 4 \mu\text{m}$	$3,5 \pm 3 \mu\text{m}$

7.1.2. G9T Mass Balance System

Table 9 - Gear characteristics of the G9T mass balance system.

Gear Characteristics

Gear n.º	8200085490	8200085493	8200085476
Operating centre distance	119,4	60	60



FINAL PROJECT

Experimental Comparison of the Mass Balance System
performance With Carbonitrided Steel Gears and
Austempered Ductile Iron Gears



Number of tooth	49	41	41
Normal module	1,55	1,4	1,4
Normal helix angle	23° 0' 0"	20° 3' 0"	20° 3' 0"
Helix direction	Left	Left	Right
Normal pitch diameter	82,509	61,103	61,103
Profile theoretical displacement	-1,119	-0,516	-0,516
Profile maximum finishing displacement	-1,16	-0,558	-0,558
Profile finishing minimum displacement	-1,228	-0,599	-0,599
Base angle helix	21° 32' 28"	18° 47' 38"	18° 47' 38"
Base diameter	76,729	56,976	56,976
Tip outside diameter	0 84,42 -0,27	0 63,16 -0,27	0 63,16 -0,27
Tip maximum active diameter	84,02	62,56	62,56
Root maximum active diameter	77,15	57,83	57,83
Root or inside diameter	0 73,82 -0,27	0 55,81 -0,27	0 55,81 -0,27
Test ball diameter	3	2,5	2,5
Dimension over 2 balls	-0,08 85,31 -0,22	-0,01 63,79 -0,13	-0,01 63,79 -0,13
Reference centre distance	De 119,09 a 119,29		
Hole diameter	21,95 N6	21,55 K6	21,55 K6

Table 10 - Tooth quality criteria of G9T mass balance system.

Teeth quality criteria

Gear n.º	8200085490			8200085493			8200085476		
Criteria	Right side	Left side.	Class	Right side	Left side.	Class	Right side	Left side.	Class
Profile slope deviation ($f_{H\alpha}$)	$0 \pm 6 \mu\text{m}$	$0 \pm 6 \mu\text{m}$	6	$0 \pm 6 \mu\text{m}$	$0 \pm 6 \mu\text{m}$	6	$0 \pm 6 \mu\text{m}$	$0 \pm 6 \mu\text{m}$	6
(C_α)	$3 \pm 2 \mu\text{m}$	$3 \pm 2 \mu\text{m}$	6	$3 \pm 2 \mu\text{m}$	$3 \pm 2 \mu\text{m}$	6	$3 \pm 2 \mu\text{m}$	$3 \pm 2 \mu\text{m}$	6
Helix slope deviation ($f_{H\beta}$)	$0 \pm 7 \mu\text{m}$	$0 \pm 7 \mu\text{m}$	6	$0 \pm 7 \mu\text{m}$	$0 \pm 7 \mu\text{m}$	6	$0 \pm 7 \mu\text{m}$	$0 \pm 7 \mu\text{m}$	6
(C_β)	$3 \pm 2,5 \mu\text{m}$	$3 \pm 2,5 \mu\text{m}$	6	$3 \pm 2,5 \mu\text{m}$	$3 \pm 2,5 \mu\text{m}$	6	$3 \pm 2,5 \mu\text{m}$	$3 \pm 2,5 \mu\text{m}$	6
Total cumulative pitch deviation (F_p)	$27 \mu\text{m}$	$27 \mu\text{m}$	6	$24 \mu\text{m}$	$24 \mu\text{m}$	6	$24 \mu\text{m}$	$24 \mu\text{m}$	6
Runout(F_r)	$21 \mu\text{m}$		6	$19 \mu\text{m}$		6	$19 \mu\text{m}$		6
(f_w)	$3 \mu\text{m}$	$3 \mu\text{m}$	6	$3 \mu\text{m}$	$3 \mu\text{m}$	6	$3 \mu\text{m}$	$3 \mu\text{m}$	6

7.2. Gears and Metal Sleeve Designs

Infra are the schematic drawings of wheels. The detailed drawings of wheels can be found in appendix A.

These drawings represent the final solution to make the wheels.

7.2.1. M1D Mass Balance System Gears

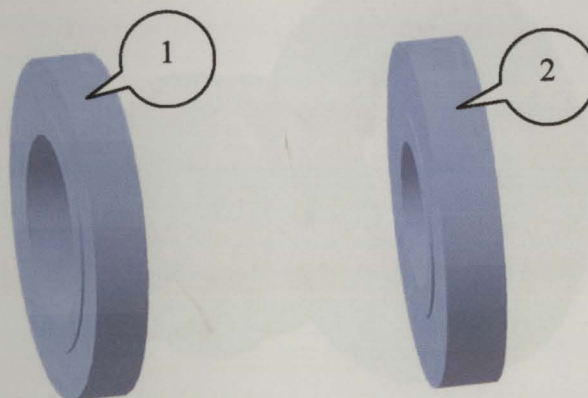


Figure 4 – Wheels M1D 104 (1) and M1D 593* (2)

These wheels belong to mass balance system M1D.

7.2.2. G9T Mass Balance System Gears

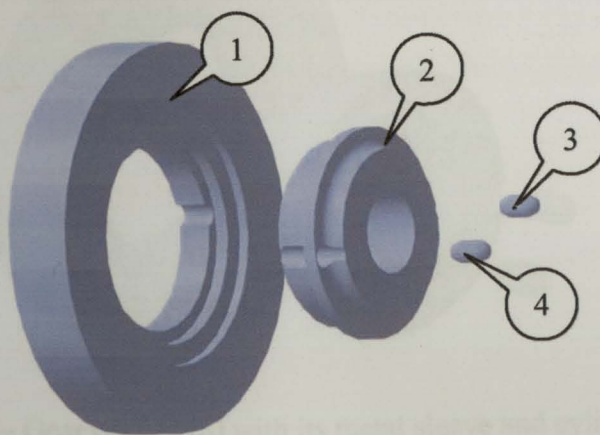


Figure 5 – Gear G9T-(080) with its metal sleeve and cylindrical pins.

* Numbers 104 and 593 correspond to the last three numbers of the intermediate drawings.



In the following figure it is possible to see the wheel G9T with the respective metal sleeve and cylindrical pins. In this figure is also possible to see how the assembly of the parts can be achieved.

For the other two wheels of this mass balance system the construction process was the same. The function of pins is to give the necessary binding allowing the manufacturing of the gears. These pins (tension-pins) make possible the easy extraction, to submit the heat treatment of the gears.

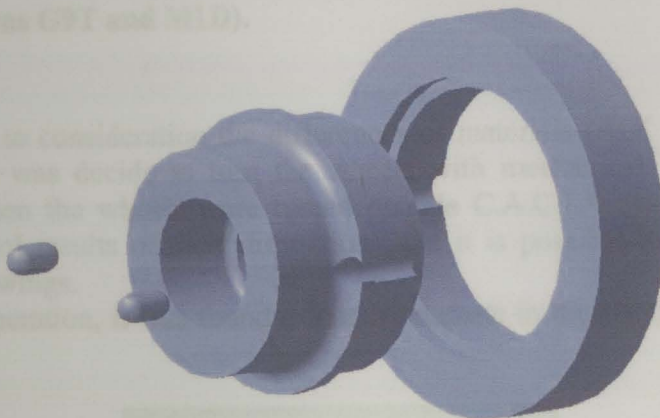


Figure 6 – Gear G9T (079) with its metal sleeve and cylindrical pins.

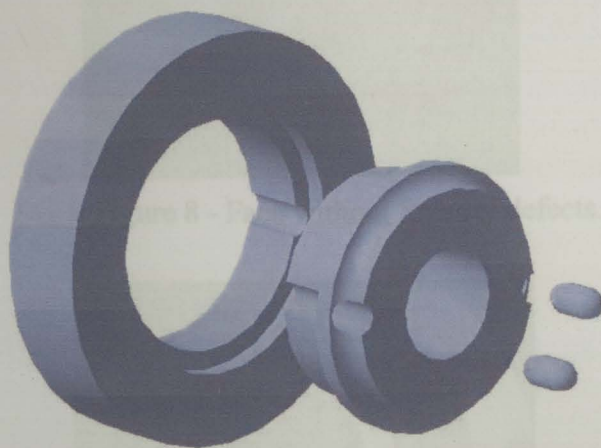


Figure 7 – Gear G9T (078) with its metal sleeve and cylindrical pins.



8. Production of gears

In this stage of the project the ADI wheels have the same dimensions that the steel wheels in the beginning of the production. This way it is possible to introduce the ADI wheels in the production process.

8.1. Turning (pinions G9T and M1D).

Taking into consideration the differences of materials (ADI vs. steel) and due to the metal sleeve it was decided to turn the wheels with mechanical lathe and not in the production line. Then the wheels were turned outside C.A.C.I.A.. In the appendix C are presented the control results of their dimensions and it is possible to verify that they are inside tolerance drawings.

After this operation, it was found a foundry defects in the HP 5 wheel after a visual analysis.



Figure 8 - Face without foundry defects.



Figure 9 - Face with foundry defects.



These defects could be a problem, because they may appear in the teeth after hobbing.

Due to the different behaviour in the heat treatment (different volumes increase) and taking into consideration the value found in the literature (0.4%) and also because of the lack of information on that volume increase in gears, with this dimensions, it was decided hobbing the gears G9T z49. If the final dimension is not inside tolerance, it is maybe possible to correct the mistake in the assembly.

8.2. Gears G9T

This stage was done in the production line following the normal operations, but having more careful in the hobbing.

8.2.1. Hobbing (G9T)

For hobbing gears G9T the radius under ball was calculated. This value is all that the hobbing machine needs to hob the gears. The others characteristics are related to this value.

Table 11 - Hobbing radius under ball.

G9T Mass Balance System				
49	RUB*		41	RUB*
Grinding	39,601		Grinding	29,36
Over Thickness	80 μ m		Over Thickness	80 μ m
Hobbing	With H.T.	Without H.T.	Hobbing	29,594
	39,677	39,835	*Radius Under Ball	

The MatLab file containing the calculus and the expression of the table value is in the appendix D. The values presented in the table have an associated error of 0.1% compared to the MatLab calculated ones.

To hob theses wheels it was considered an over thickness of 80 μ m for all cases (for the wheel that will be treated and for the one that was not treated). In the case of the wheel that will be treated, the volume increase (due to heat treatment) was removed to the over thickness. The maximum value of volume increase is 0.4% [2]. The two wheels were cut with different dimension under ball. The objective of this operation is to try to know the volume increase that the treated wheel will have, so that the remaining wheels be inside of drawings specifications. Also for a safety reason, the hob was accomplished with an approach speed of the gear miller with in 60% of the normal speed, which works at a rotational speed of 480 rpm.

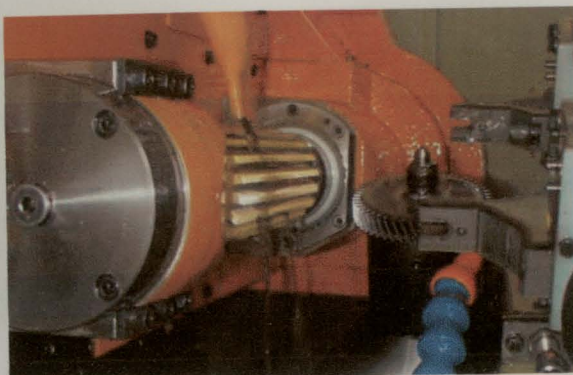


Figure 10 - Hobbing machine.

After hobbing, a gear visual analysis was made; it was possible to verify the emergence of the foundry defects, as it shown in the following figures.



Figure 11 – Gear HP5 with foundry defects.

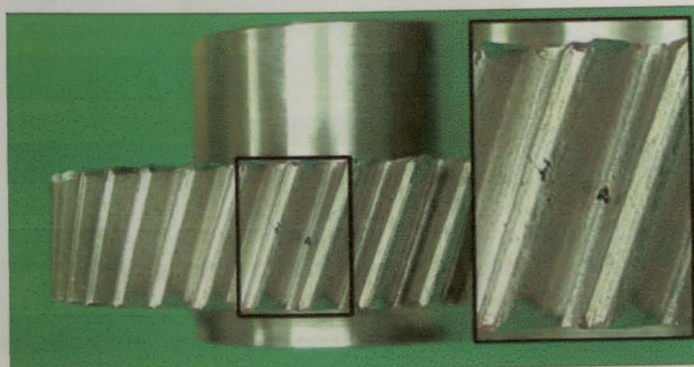


Figure 12 – Gear HP7 with foundry defects.

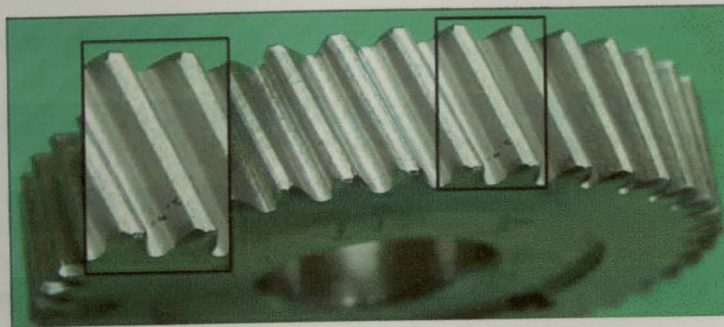


Figure 13 – Gear HP9 with foundry defects.

Foundry defects can provoke failure of a tooth when these are geared.

8.2.2. KLINGELNBERG Control – G9T gears after hobbing.

The cogged control was made in KLINGELNBERG machine. The total cumulative pitch deviation (F_p), total profile deviation (F_α), total helix deviation (F_β) and runout (F_r) were controlled. It was verified that cogged dimensions after hobbing were inside drawings specifications.



Figure 14 - KLINGELNBERG measuring machine.

It is important to say that the control of the radius under ball (RUB) was also made in production line because it is more exact that the value measure in the KLINGELNBERG machine. In the appendix E it is possible to see the F_p , F_a and RUB of the all gears and the others characteristics.

8.2.3. Heat treatment (gears G9T).

Gears heat treatment was making in FEUP (cast iron part) and in F. Ramada, Aços e Industrias, S. A. (steel part). In the appendix are the steel's heat treatment characteristics.

The gears G9T z41 left and right were not suppose to be treated, but due to an error in FEUP these gears were treated. Because of this fact it was impossible to make the planned tests. The number of cast iron gears is inferior to the ones previously expected. Only the HP5 (G9T z49) is cast iron, but it will be assembled and tested with the ADI gears.

8.2.4. KLINGELNBERG Control - after heat treatment (gears G9T).

After heat treatment the cogged dimensions were controlled and it was verified that many dimensions were different. These variations were provoked due to heat treatment and by the assembly of the metal (steel) sleeve. After the heat treatment the sleeve also had dimensional variations. These variations (both the assembly and heat treatment) increase the ADI variation and create a difficult situation, because it is impossible to know which variations are due to the heat treatment.

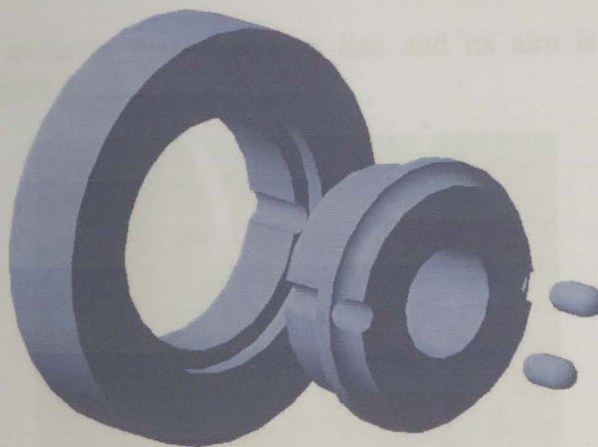


Figure 15 - Example of the assembly.

The treated gear and the non treated gear had variations. The variations in the non treated gear were provoked only by metal sleeve assembly.

8.2.5. Hard turning and grinding (gears G9T).

After heat treatment it was made the hard turning (the hole and one face of the gears) and the grinding of the cogged.



Hard turning is an operation made without lubrication fluid, and its aim is to give at gear bore the final dimension and surface finish. After this operation the cogged was dimensionally controlled and was possible to verify that cogged characteristics were better than after heat treatment. However the cog dimensions are out of drawings specifications.

After this operation the cogged dimension was controlled again, being verified that there was an improvement in all controlled dimensions. Even so it stayed out of the drawings specifications.

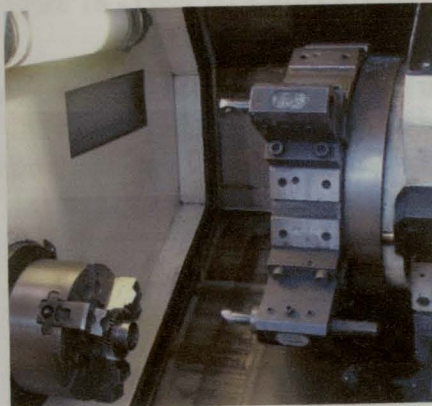


Figure 16 - Hard turning.

The grinding was made in the production line and its aim is to give the final dimensions and surface cogged finish.



Figure 17 - Grinding machine.

After this operation it was verified that the gear HP4 was not completely grinding, because their teeth were not enough thickness. Consequently one flank was not grinding like shows the following figure.

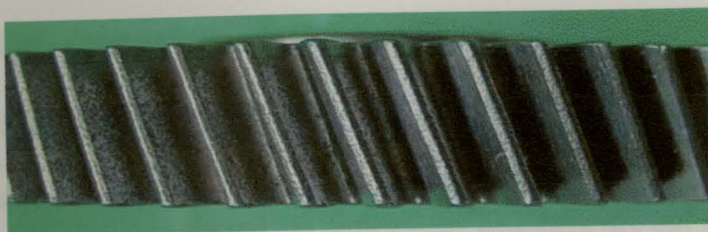


Figure 18 - Gear HP4 after grinding.

This happens due to the radius under ball used to hobbing the gear. In that time it was considered a volume increase of the 0.04%, which it did not happen or it was much lower. Even though the gear HP4 is inside the tolerances, as is possible to observe in the appendix E.

8.3. Gears M1D

Taking into consideration that all gears z41 G9T were previously treated, it has been decided to treat the M1D gears. In this case the tests will be only with ADI gears.

8.3.1. Hobbing pinions M1D

These gears were hobbing in the production line like G9T gears, and it was considerate the results obtained in the previous case. The initial problem to hobbe the gears it was to know that is the volume increase due to the heat treatment. Based in these facts it was calculate the over thickness to hobbe these gears

Table 12 - Hobbing radius under ball.

M1D Mass Balance System		
	RUB [mm]	
	z49 left	z41 left and right
After Grinding	37,108	36,881
Over Thickness - 80 μ m		
After Hobbing	37,36	37,124

This operation was implemented in the same conditions that the series gears, the same approach speed and rotational were used in the gear miller.

8.3.2. KLINGELNBERG control (gears M1D)

In the KLINGELNBERG control machine it was verified that the gears' dimensional characteristics are inside the tolerances.

After heat treatment the gears were controlled and major deformations were noticed in the cogged. These deformations altered the dimensions and some are out of tolerances. In addition to deformations it was found foundry defects that were not visible also, appeared after heat treatment. Its surface has this appearance, because it was clean with sand jet.



Figure 19 - Gear HP3 (M1D z49 left) after heat treatment, with foundry defects.

8.3.3. Hard Turning and Grinding (gears M1D)

These two operations were made in the production line following the normal process. They were made in the same conditions that the series gears.

After this stage it was possible to verify that all dimensions and geometric parameters are into tolerances (appendix E).

8.4. Final conclusions

The ADI shows a different behaviour that the steel used in series gears. The cast iron used has problems with foundry, appears foundry defects.

Heat treatment introduces very deformation although they can be correct in the final operations.

Table 13 - Final dimensions.

			RUB	Fr	Tip Radius	Root Radius
G9T	Nominal	Max.	29,367	0	-	-
		Min.	29,353	19	-	-
	z41 left	HP8	29,336	10	-	-
		HP9	29,363	10	-	-
	z41 right	HP6	29,36	10	-	-



FINAL PROJECT

Experimental Comparison of the Mass Balance System
performance With Carbonitrided Steel Gears and
Austempered Ductile Iron Gears



Universidade do Porto
Faculdade de Engenharia
FEUP



CACTA



RENAULT

prodepiii

	Nominal	HP7	29,362	14	-	-
		Max.	39,636	0	-	-
		Min.	39,566	21	-	-
	z49 left	HP4	39,584	25	-	-
		HP5	39,583	17	-	-
M1D	Nominal	Max.	36,899	0	-	-
		Min.	36,863	21	-	-
	z47 left	HP2	36,887	8	38,978	34,185
	z47 right	HP1	36,874	12	39,009	34,227
	Nominal	Max.	37,153	0	-	-
		Min.	37,063	21	-	-
	z49 left	HP3	37,108	6	39,375	34,673



9. Gears assembly

9.1. G9T

Gears assembly it was doing in the production line in the same conditions which the series gears. First it was assembly the gears in the shafts as show the next figure.

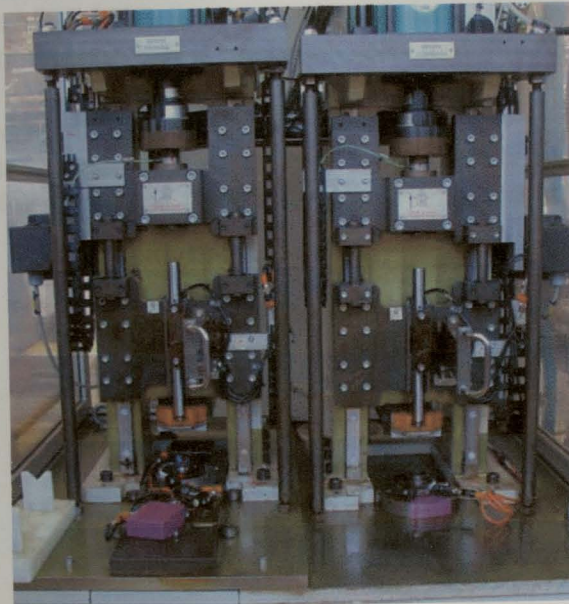


Figure 20 - Assembly the gears G9T in the shafts.

After assembly the gears in the shafts it was assembly the shafts in the crankshaft as show the next figure.

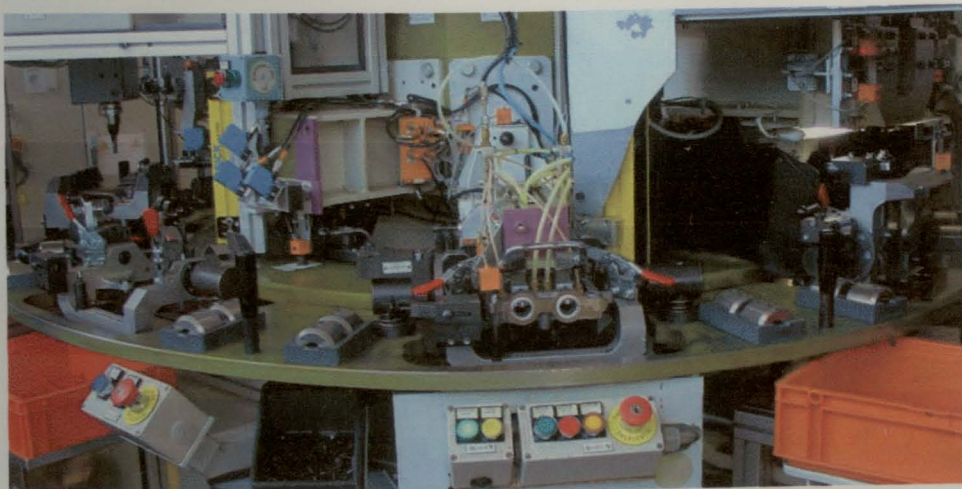
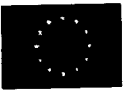


Figure 21 - Assembly the shafts in the crankshaft (G9T).

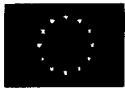


During these stages only the assembly of the series shafts in the crankshaft it was not possible. Consequently the assembly it was doing manually. The others shaft they were assembly normally without problems.

9.2. M1D

When the MBS M1D was assembly the production process it was in the test stage, because at this moment the MBS M1D is not a series product. However the assembly was made following the production process. It is important refer that this process is different that the MBS G9T process.

The assembly flowed normally and when the gears were assembly in the shafts it was possible to see that ADI gears make possible the process to flow faster. Gears were heating up before the assembly in the shafts and the ADI gears heated up must faster than the steel gears.



10. Experimental tests

The experimental tests was realise in the ORMO tests banc. This tests banc needs two MBS to equilibrate the forces generated by eccentric masses rotation. Tests aim is to try measure the meshing vibration to know which material has better behaviour.

10.1. MBS G9T ORMO tests

It was tested three MBS, one with ADI gears, one with ADI and cast iron gears and last one with steel gears. How the time is not enough it was decide to make these tests with the maximum backlash only, because it was considered the worse work situation. In the next table were the values of backlash and the position of MBS.

Table 14 - Backlash

66E + 63D				63E + Serie_2						
A Side		B Side		A Side		B Side				
Backlash				Backlash						
66E			63D		1	Serie_2		2	63E	
z41 - z41	z49 - z98		z41 - z41	z49 - z98		z41 - z41	z49 - z98		z41 - z41	z49 - z98
0,09	0,075		0,011	0,07			0,06		0,04	0,08
0,07	0,07		0,09	0,07			0,06		0,04	0,09
0,09	0,09		0,011	0,08			0,06		0,05	0,08
0,08	0,08		0,01	0,07			0,04		0,05	0,09

Table 15 - Backlash tolerances.

Tolerances	
z49 - z98	0,08
	0,02
z41 - z41	0,184
	0,019

Analysing the previous tables it is possible verify that the meshing z41 – z41 are into the tolerances and that the MBS that were assembly together are a similar backlash, but the series MBS have a backlash smaller than the others. This is important, because it is a parameter that influences the vibrations behaviour.

Other parameter that is important it is the lateral play that its values are presented in the next table with their tolerances.



Table 16 - Lateral play.

Lateral play [μm]				
	63E (Series)	63D (ADI + cast iron)	66E (ADI)	Tolerance
Primary shaft	110	156	146	100
Secondary shaft	120	142	134	200

All lateral plays are into the tolerances and the MBS that will go be assembly together have a similar lateral play.



Figure 22 - Backlash control (G9T).

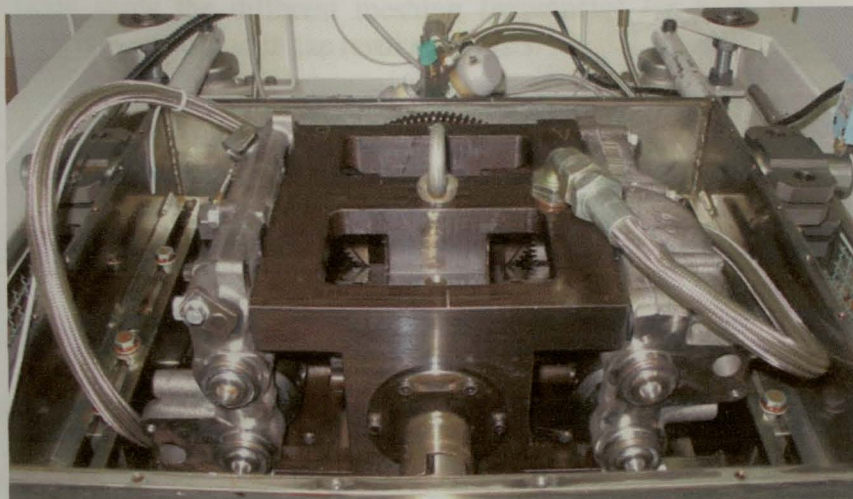


Figure 23 - ORMO tests banc (G9T).

10.1.1. Tests: preparation and realisation

To measure the vibration, initially it was used an accelerometer triaxial, but the unique accelerometer available can not be used, because it was not ground isolate. Consequently it was decide to use two accelerometers uniaxial and mounting as show the next figure.



Figure 24 - Accelerometer mounting and position.

With this mounting it is possible to measure the vibration in two directions one perpendicular at MBS and the other one parallel. The accelerometer characteristics were in the appendix F.

The next table show the different meshing frequencies for both meshing at the different speeds that it is hope to find in the results analysis.

Table 17 - Meshing frequencies

rpm	Meshing frequencies [Hz]	
	49	41
850	1388	1162
2000	3267	2733
4000	6533	5467

Taking into consideration the meshing frequencies and the characteristics of the acquisition system (PROSIG) it was define the following parameters:

- Acquisition time – 1 s;
- Acquisition frequency – 40 000 Hz;
- Filter low pass – 10 000Hz.



These values were choosing taking into consideration the error of accelerometer (to the filter low pass) and the meshing frequencies. In the appendix F are the accelerometer characteristics.

10.1.2. Results analysis

To analyse the temporal response it was calculated its FFT. This way it is possible to know what is the response with bigger frequency amplitude.

After an analysis of the results (the graphs are in the appendix G) it was possible to verify that any meshing frequency is present in the graphs (FFT). The graphics present many frequencies which characterise all MBS and not only the meshing as it was waited.

Analysing the different rotational speed, at 850 rpm both measure directions (X and Y) have the same behaviour, appearing at a frequency of the 8000 Hz the natural frequency with the bigger magnitude value. This frequency is from MBS with steel gears in one situation and from MBS with cast iron gear.

Each colour corresponds to:



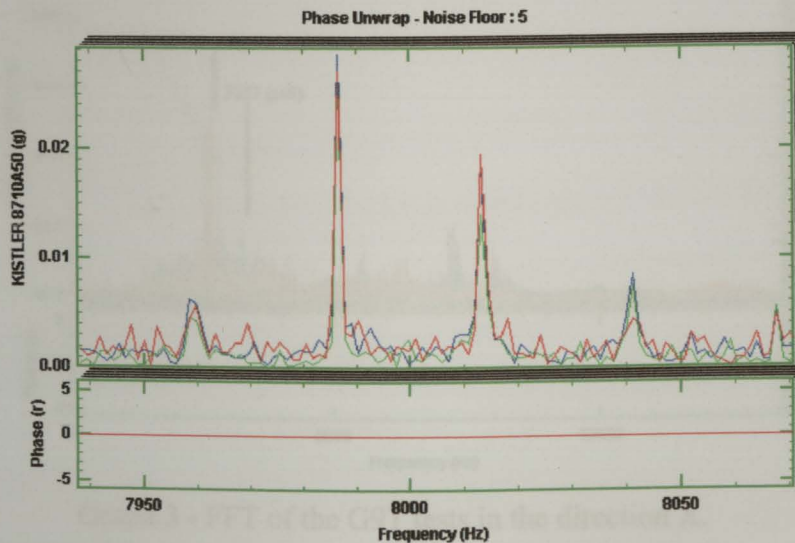
Steel



ADI



Cast Iron + ADI

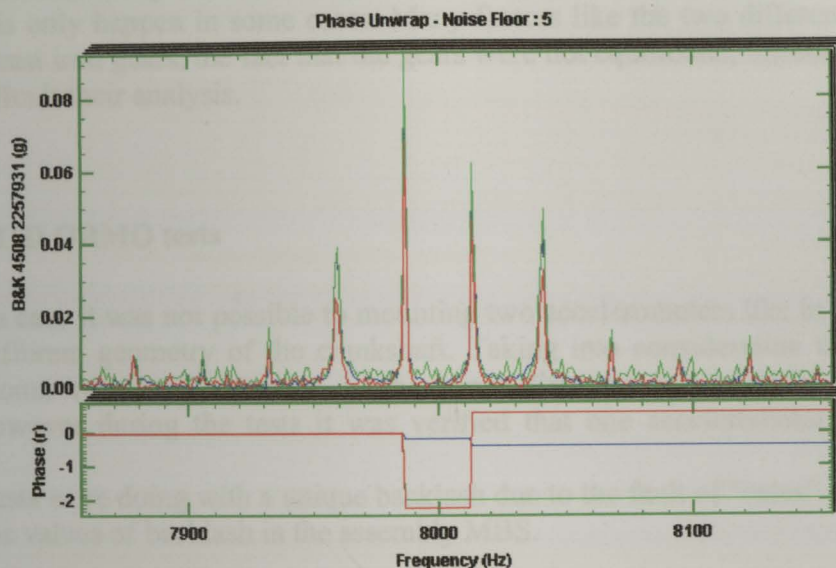


Graph 1 – FFT of the G9T at 850 rpm in the direction X.

Analysing the different rotational speed, at 850 rpm both measure directions (X and Y) have the same behaviour, appearing at a frequency of the 8000 Hz the natural frequency

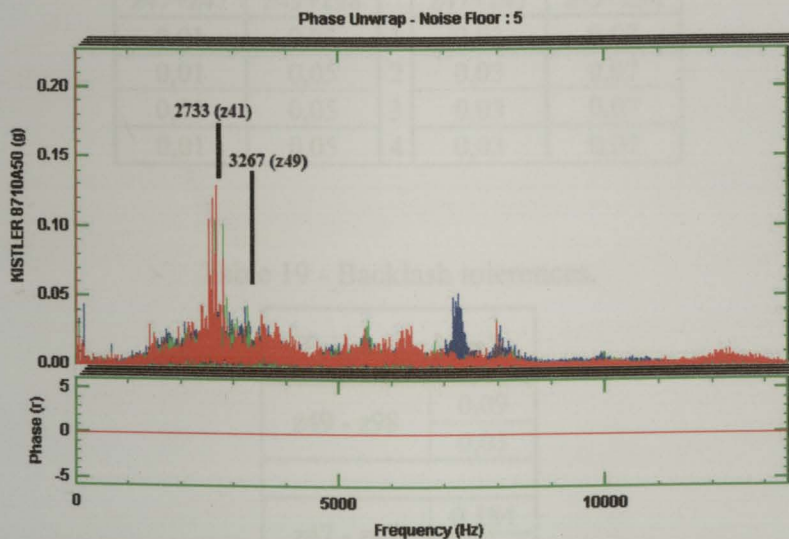


with the bigger magnitude value. This frequency is from MBS with steel gears in one situation and from MBS with cast iron gear.



Graph 2 - FFT of the MBS G9T at 850 rpm in the direction Y.

At 2000 rpm the MBS with ADI show a worse behaviour, because with the natural frequencies at the same frequencies it has bigger magnitude.



Graph 3 - FFT of the G9T tests in the direction X.

With a rotational speed of the 4000 rpm the MBS with one gear in cast iron and in the direction Y has more natural frequencies and with bigger magnitude. Analysing the signals of the others tow MBS, it is verified that the MBS with steel has the same natural frequencies



but bigger than MBS with ADI. In the direction X the MBS with ADI gears has a worse behaviour.

It is possible to say that the ADI give at MBS a better behaviour than the steel, but in these tests this only happen in some cases. Many factors like the two different materials in the ADI and cast iron gears, the fact that the gears were not equilibrate, influence the results, and make difficult their analysis.

10.2. MBS M1D ORMO tests

In this case it was not possible to mounting two accelerometers like in the G9T tests, due to the different geometry of the crankshaft. Taking into consideration this fact it was decide to mount two accelerometers in the same direction but in different crankshaft positions. However during the tests it was verified that one accelerometer did not work correctly.

The tests were doing with a unique backlash due to the fault of "cales". The following table show the values of backlash in the assembly MBS.

Table 18 - Backlash

A Side		B Side		
Backlash [mm]				
A 102 (Series)		1 2 3 4	A 101 (ADI)	
z47+z47	z49+z98		z47+z47	z49+z98
0,01	0,05		0,03	0,07
0,01	0,05		0,03	0,07
0,01	0,05		0,03	0,07
0,01	0,05	0,03	0,07	

Table 19 - Backlash tolerances.

Tolerances [mm]	
z49 - z98	0,09
	0,03
z47 - z47	0,184
	0,019

As it is possible to see in the previous table only the meshing z47 – z47 of the series MBS have a backlash that is out of tolerances and it is very different of the ADI MBS. The meshing z98 – z49 have a different backlash too. This data are important, because they will have a big influence over the tests and they make impossible the comparison.

Table 20 – Lateral play and tolerances.

Lateral play [μm]			
	A 102 (Series)	A 101 (ADI)	Tolerance
Primary shaft	200	216	35
Secondary shaft	206	190	320

The previous table show the lateral play of the MBS M1D and it is possible to verify that the clearance is into the tolerances.

10.2.1. Tests: preparation and realisation

As it was already said, it was mounting tow accelerometer as show the next figure, but only one works. The configuration of the acquisition it was the same used in the G9T tests.

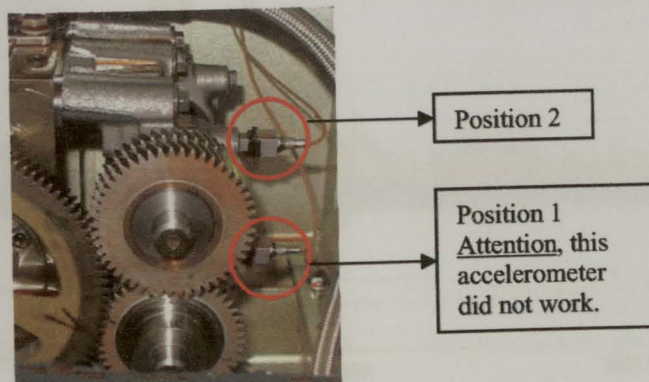


Figure 25 - Accelerometers position.



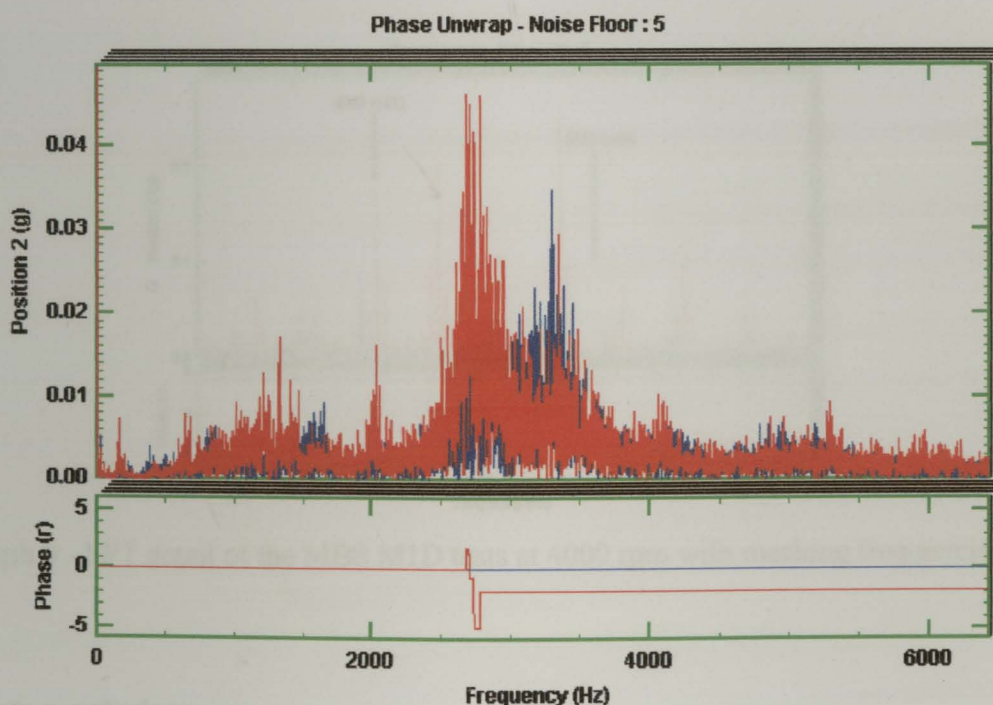
Figure 26 - Assembly in tests banc.

10.2.2. Results analysis

Table 21 - Meshing frequencies.

rpm	Meshing frequencies [Hz]	
	49	47
850	1388	1332
2000	3267	3133
4000	6533	6267

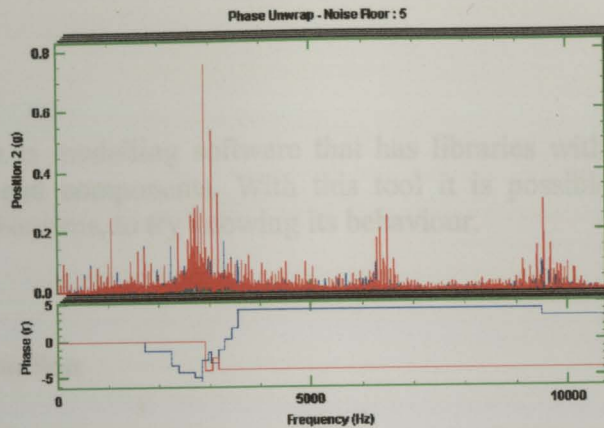
Analyse the results like happen in the G9T tests the meshing frequencies was not visible in the FFT graphs.



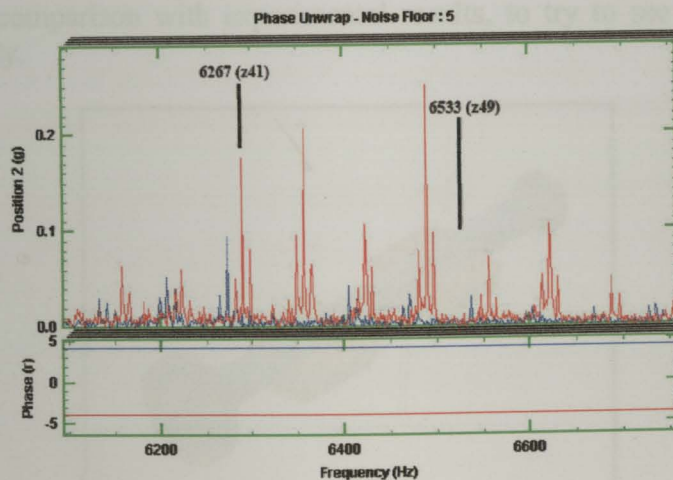
Graph 4 - FFT of the MBS M1D at 850 rpm.

For these MBS were found natural frequencies with great magnitude in the frequencies interval from 2500 to 3500 Hz, but the MBS with ADI show much time a bigger magnitude. This can be explained, due to the fact that the MBS with steel gears has smaller backlash in both meshing than the ADI. Each meshing of steel gears has less 20 μm than the ADI gears. Consequently the comparison of these MBS it is impossible because they are not in the same conditions. The shocks in the MBS with ADI are stronger than the MBS with steel, what provoke more and more strong vibrations.

In this case it is necessary to make more tests with other series MBS and if is possible with the same backlash or the closest possible of MBS with ADI gears



Graph 5 - FFT of the MBS M1D tests at 4000 rpm.



Graph 6 - FFT detail of the MBS M1D tests at 4000 rpm with meshing frequencies.

10.3. Tests conclusions

In the MBS G9T with ADI it is possible to see that the ADI give to the MBS a better behaviour in some situations, principally at 850 rpm and 4000 rpm. At 2000 rpm the Steel show a better behaviour than ADI. The MBS used in these tests are the backlash near one of the others, but not exactly the same as well as the lateral play. The variation of these values is very important because they influence directly the system behaviour.

However these tests are not enough to obtain a good conclusion, because the tests with MBS M1D are not valid for comparison due to the different backlash.

In the MBS G9T tests are tow rotational speed where the ADI are better and other one where the steel is better. It was necessary to make others tests with plus rotational speeds and maybe thinking in others types of tests.



11. AMESim

The AMESim is modelling software that has libraries with mathematic models of mechanical systems and components. With this tool it is possible to create models that simulate the real mechanisms, to try knowing its behaviour.

11.1. Models Construction

To run the AMESim simulation there were created two models for mass balance system G9T and one for mass balance system M1D. The two models of mass balance system G9T were built to clarify some doubts in relationship to the best division of the part. Later will be making a comparison with experimental results, to try to see which model better represents the reality.

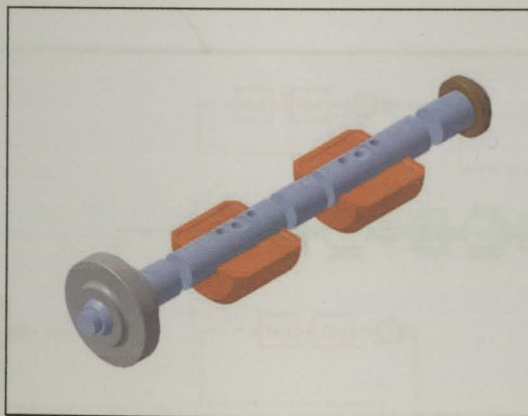


Figure 27 – G9T model 1.

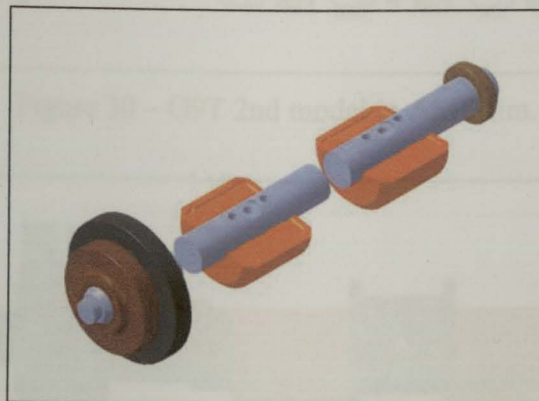


Figure 28 – G9T model 2.

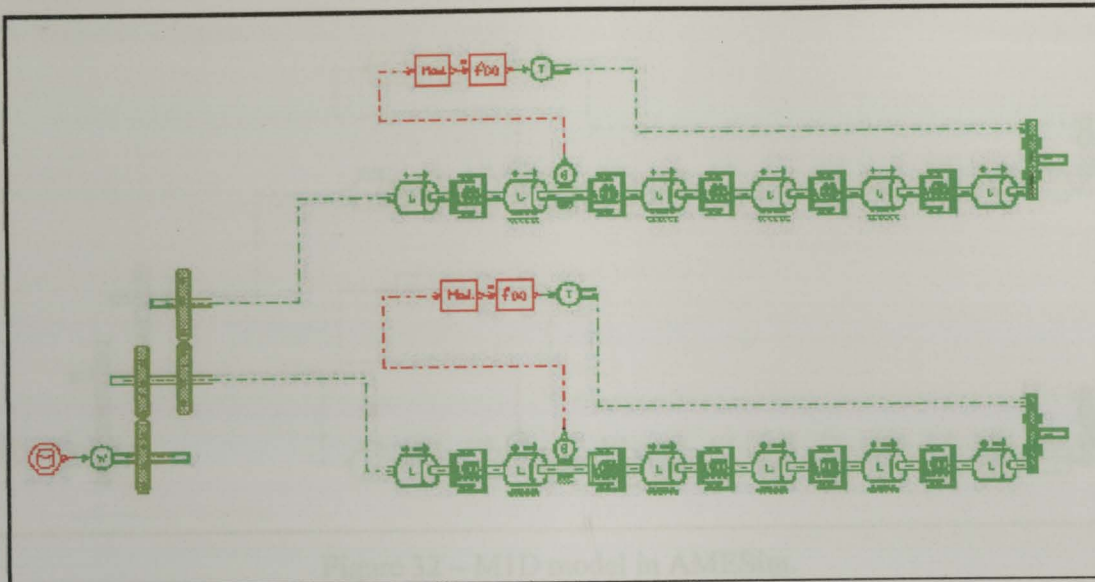


Figure 29 – G9T, 1st model in AMESim.

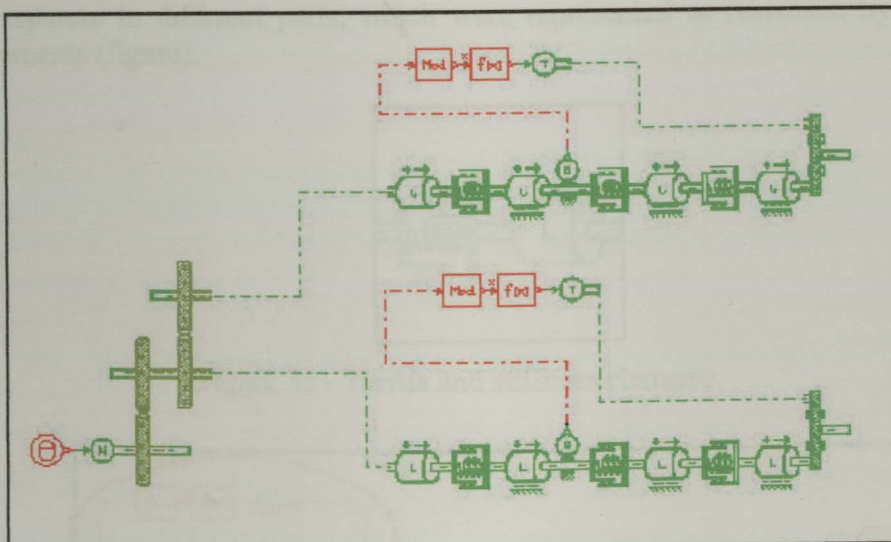


Figure 30 – G9T 2nd model in AMESim.

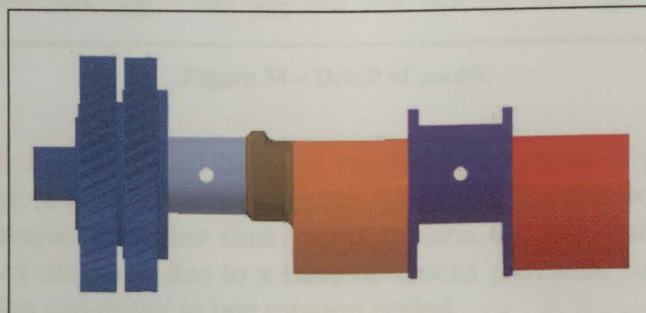


Figure 31 – M1D model.

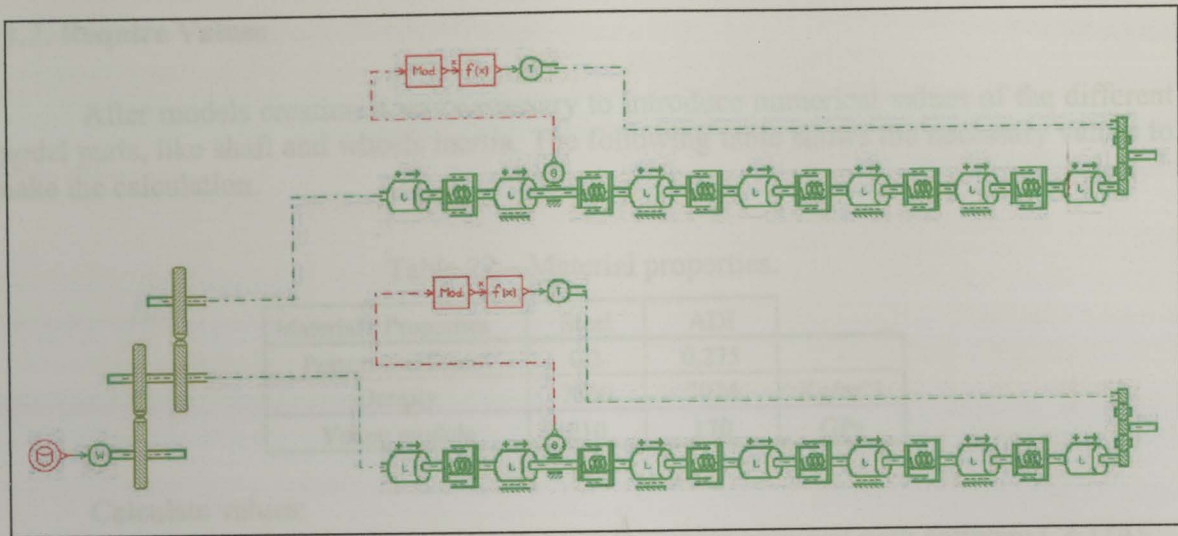


Figure 32 – M1D model in AMESim.

The models in AMESim represent all masses and inertias of shafts and wheels. The shaft was separate in different parts, which were represented in AMESim by inertia and stiffness elements (figure).

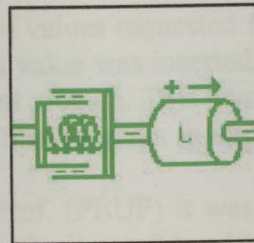


Figure 33 – Inertia and stiffness elements

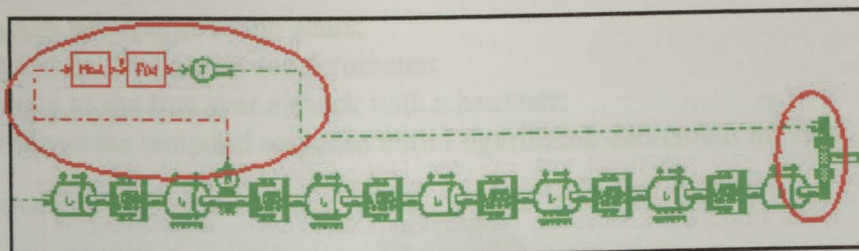


Figure 34 – Detail of model.

The elements inside the red balloons represent the effect of gravity over eccentric masses and they exist in all models constructed. While rotating this eccentric mass one time it favours the movement and other time works thwarts the movement. This phenomenon increases the noise of meshing, due to a raise of shocks provoked for active flank change. The gear can be driven and driver in one rotation period.



11.2. Require Values

After models creation it was necessary to introduce numerical values of the different model parts, like shaft and wheels inertia. The following table shows the necessary values to make the calculation.

Table 22 – Material properties.

Material's Properties	Steel	ADI	
Poison coefficient	0,3	0,275	-
Density	7860	7074	Kg/m ³
Young module	210	170	GPa

Calculate values:

- Shaft and gears inertias – these values were obtained with software CATIA;
- Shaft stiffness;
- Mesh stiffness.

In the appendix H they are the calculate values.

The values like geometric characteristics and teeth number are introduced in the mesh stiffness calculation.

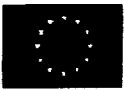
It is important to refer that one of values requested for the program was not obtained by calculations, in this case a predefined value was inserted. This value is the mesh damping and only can be obtained by experimental process. This was verified after a small research in the WEB and after contacts with AMESim (e-mails in appendix). This value depending of gears' geometric characteristics too.

However with the help of one Prof. (FEUP) it was elaborated a possible method to obtain the mesh damping value. But to make it possible with the existing equipment it will be necessary to destroy almost one gear. The procedure is:

- Mesh two gears, one is fix and the other is free;
- Apply pre-charge to the gears;
- Put in the fix gear an accelerometer;
- Apply to the free gear a shock with a hammer;
- Analyse the temporal response with Logarithmic decrement method.



Figure 35 – Accelerometer position.



Before its implementation, this procedure needs to be studied better, or find other method (the calculation of the mesh damping can give another project). In this project due to the short time available this value was impossible to obtain. The simulation was run taking into account the same value of the mesh damping for both materials. The used value was 1000 N/(m/s).

11.3. Results

The simulation was running for different values of the backlash (maximum and minimum value), to compare the behaviour in the both situations.

Table 23 – Backlash values for the G9T MBS.

		G9T	
		z41	z49
Min.	19	20	[μm]
Max.	184	80	

Table 24 – Backlash values for the M1D MBS.

		M1D
		z47 and z49
Min.	30	[μm]
Max	90	

Maybe, the most important result that AMESim provides, for this study, is the relative linear displacement. These values represent the distance between the active flanks of the mesh tooth. By analysing these values, when exist a big oscillation it is possible to say that there will be more shocks leading to an increase of noise. Based in this fact and analysing the graphs of relative linear displacement, the noise quantification can be observed.

The input signal is a rotational velocity that simulates the acyclic movement which is a motor characteristic. This figure represents the input signal.



FINAL PROJECT

Experimental Comparison of the Mass Balance System
performance With Carbonitrided Steel Gears and
Austempered Ductile Iron Gears



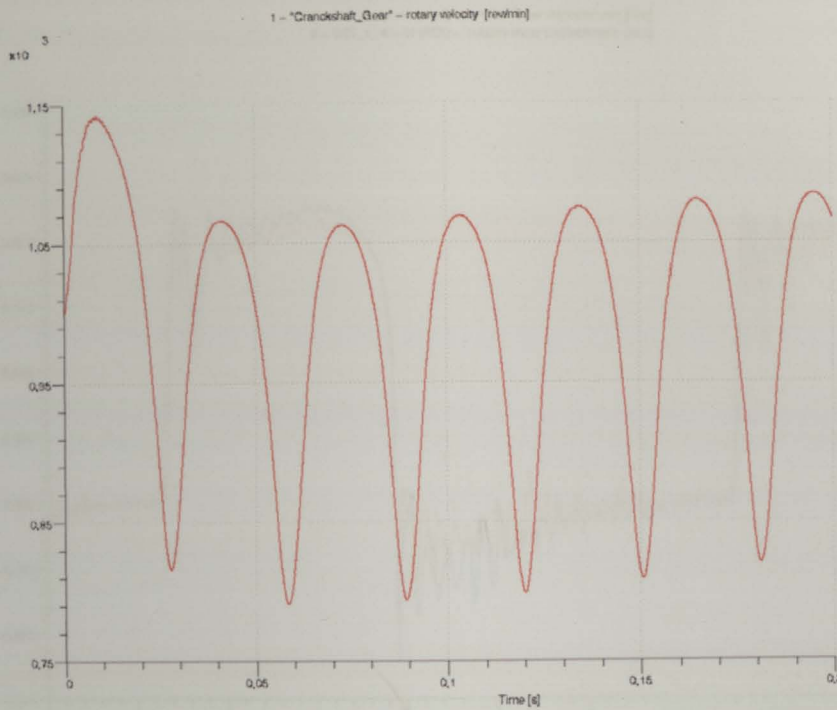
Universidade do Porto
Faculdade de Engenharia
FEUP



CACIA

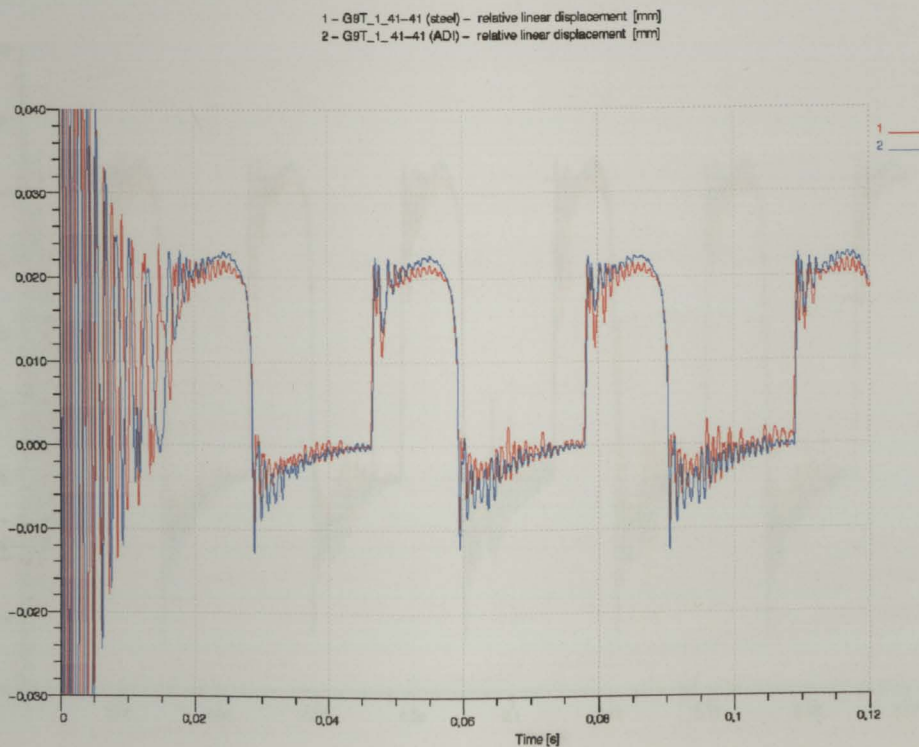


RENAULT



Graph 7 – Motor velocity at 1000 rpm.

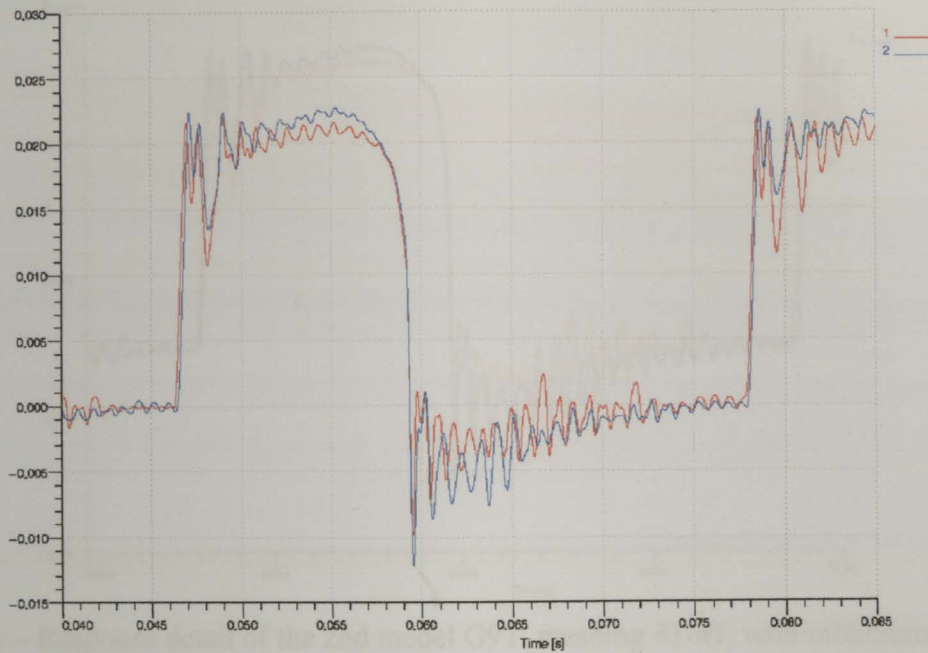
The next graphs show the relative linear displacement for the different models and different values of backlash (minimum and maximum project value).



Graph 8 – Response of the 1st model G9T, meshing 41-41, with minimum backlash.

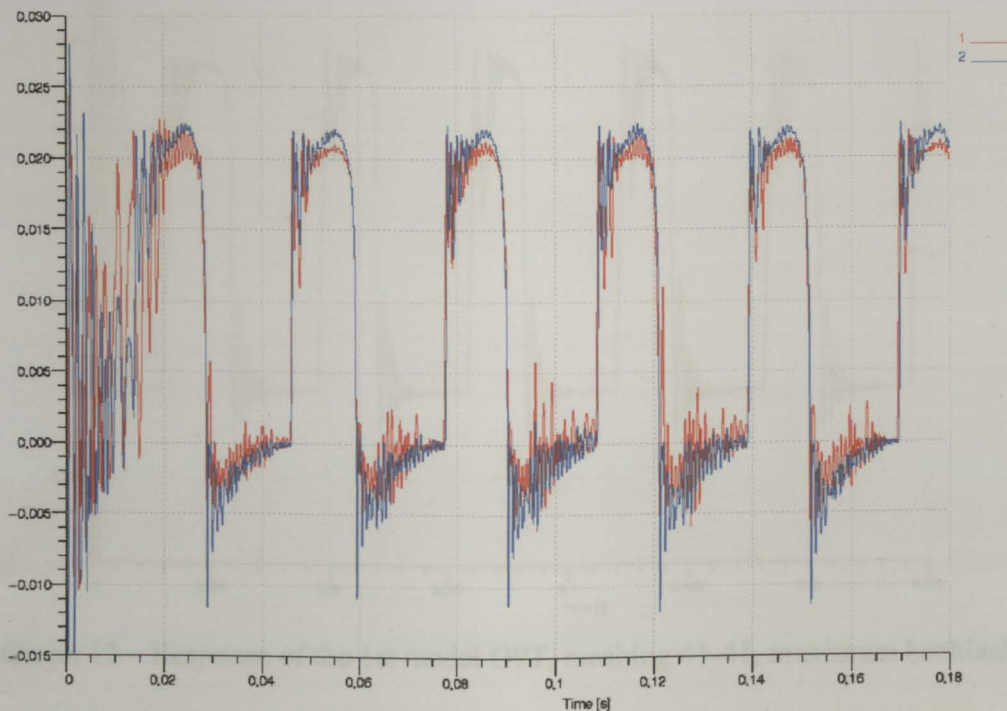


1 - G9T_1_41-41 (steel) - relative linear displacement [mm]
2 - G9T_1_41-41 (ADI) - relative linear displacement [mm]



Graph 9 - Detail of the 1st model G9T, meshing 41-41, with minimum backlash.

1 - G9T2_1_41-41 (steel) - relative linear displacement [mm]
2 - G9T2_1_41-41 (ADI) - relative linear displacement [mm]



Graph 10 - Response of the 2nd model G9T, meshing 41-41, with minimum backlash.



FINAL PROJECT

Experimental Comparison of the Mass Balance System
performance With Carbonitrided Steel Gears and
Austempered Ductile Iron Gears



Universidade do Porto
Faculdade de Engenharia
FEUP



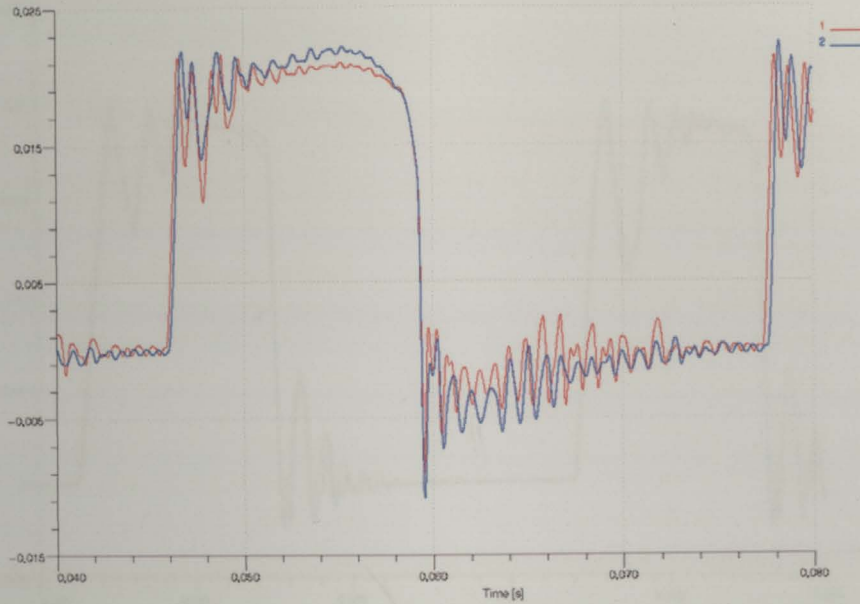
CACIA



RENAULT

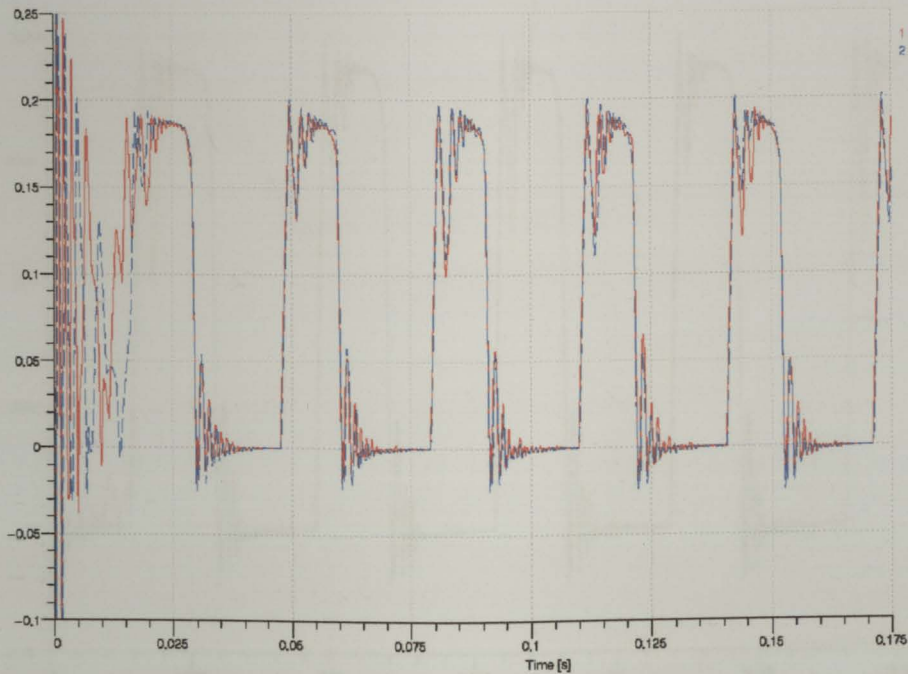


1 - G9T2_1_41-41 (steel) - relative linear displacement [mm]
2 - G9T2_1_41-41 (ADI) - relative linear displacement [mm]



Graph 11 - Response detail of the 2nd model G9T, meshing 41-41, with minimum backlash

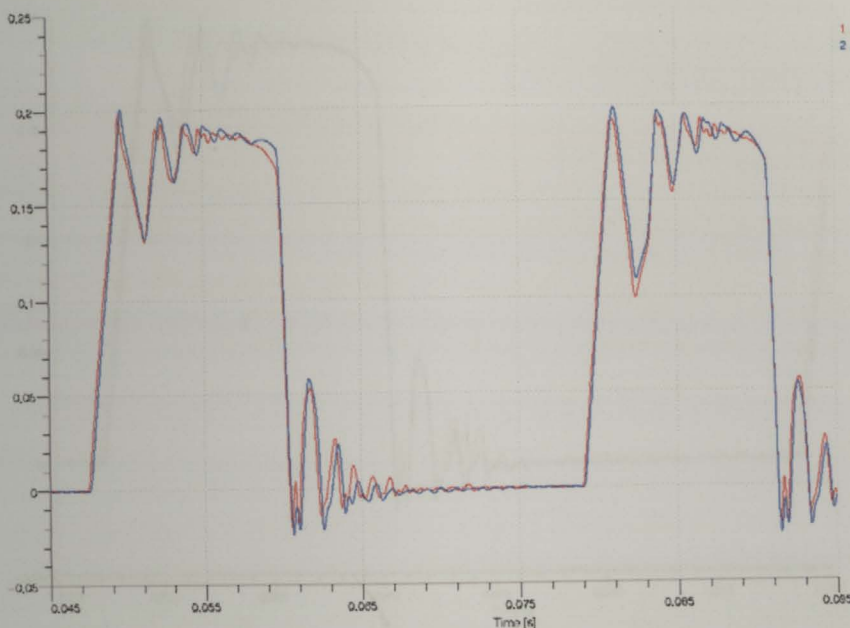
1 - G9T_2_41-41 (steel) - relative linear displacement [mm]
2 - G9T_2_41-41 (ADI) - relative linear displacement [mm]



Graph 12 - Response of the 1st model G9T, meshing 41-41, maximum backlash.

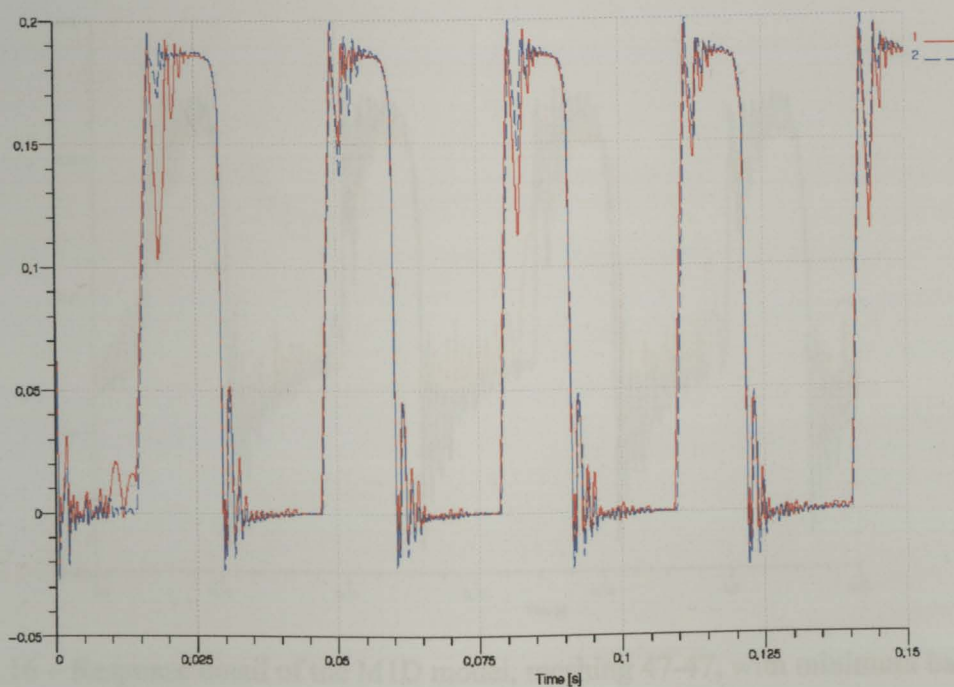


1 - G9T_2_41-41 (steel) - relative linear displacement [mm]
2 - G9T_2_41-41 (ADI) - relative linear displacement [mm]



Graph 13 - Response detail of the 1st model G9T, meshing 41-41, with maximum backlash.

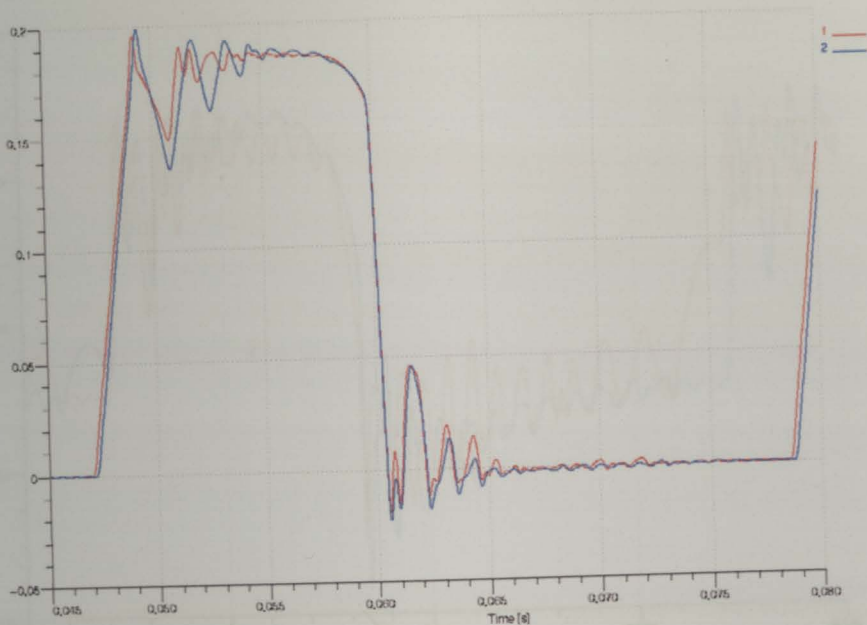
1 - G9T2_2_41-41 (steel) - relative linear displacement [mm]
2 - G9T2_2_41-41 (ADI) - relative linear displacement [mm]



Graph 14 - Response of the 2nd model G9T, meshing 41-41, with maximum backlash.

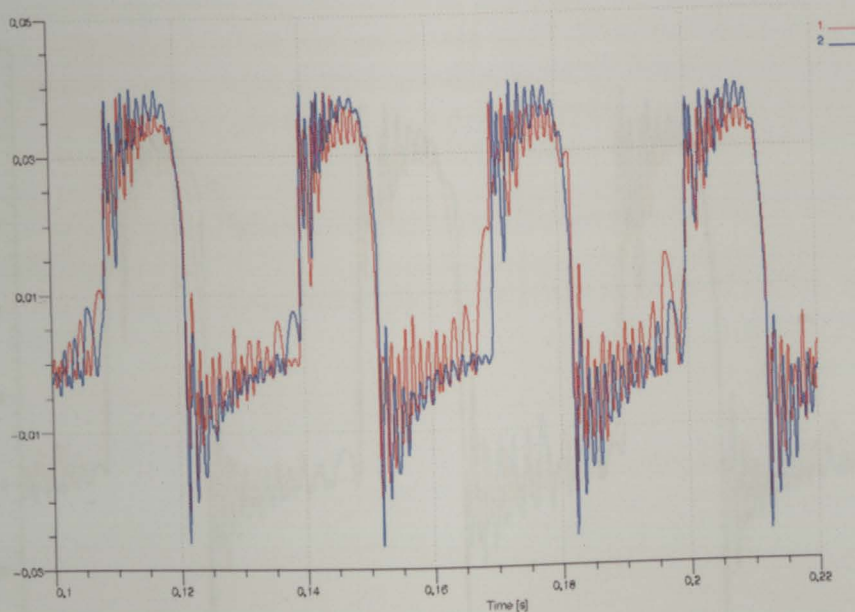


1 - G8T2_2_41-41 (steel) - relative linear displacement [mm]
2 - G8T2_2_41-41 (ADI) - relative linear displacement [mm]



Graph 15 – Response detail of the 2nd model G9T, meshing 41-41, with maximum backlash.

1 - M1D_1_47-47 (steel) - relative linear displacement [mm]
2 - M1D_1_47-47 (ADI) - relative linear displacement [mm]



Graph 16 – Response detail of the M1D model, meshing 47-47, with minimum backlash.



FINAL PROJECT

Experimental Comparison of the Mass Balance System
performance With Carbonitrided Steel Gears and
Austempered Ductile Iron Gears



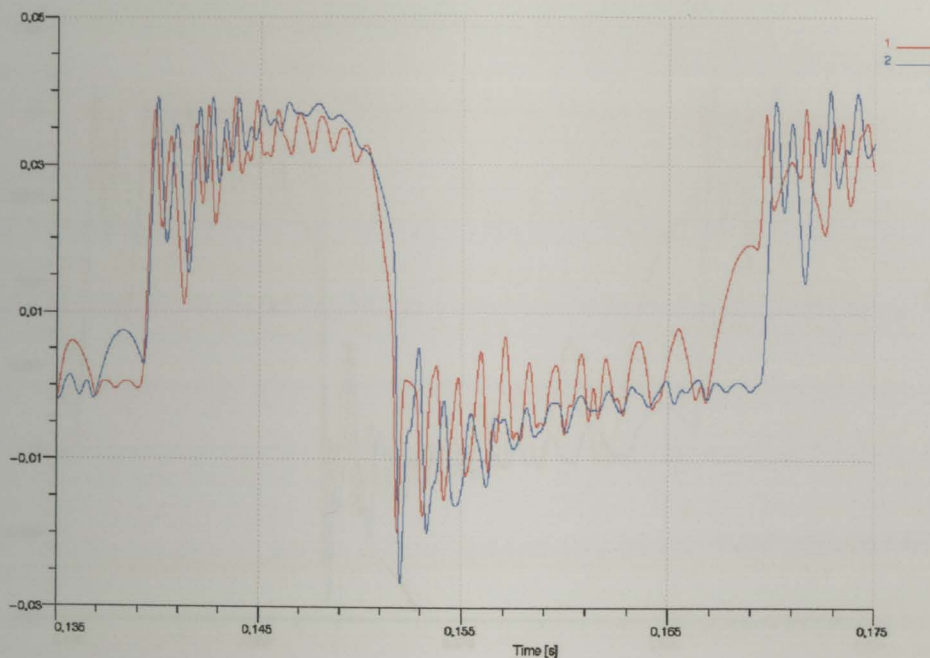
Universidade do Porto
Faculdade de Engenharia
FEUP



RENAULT

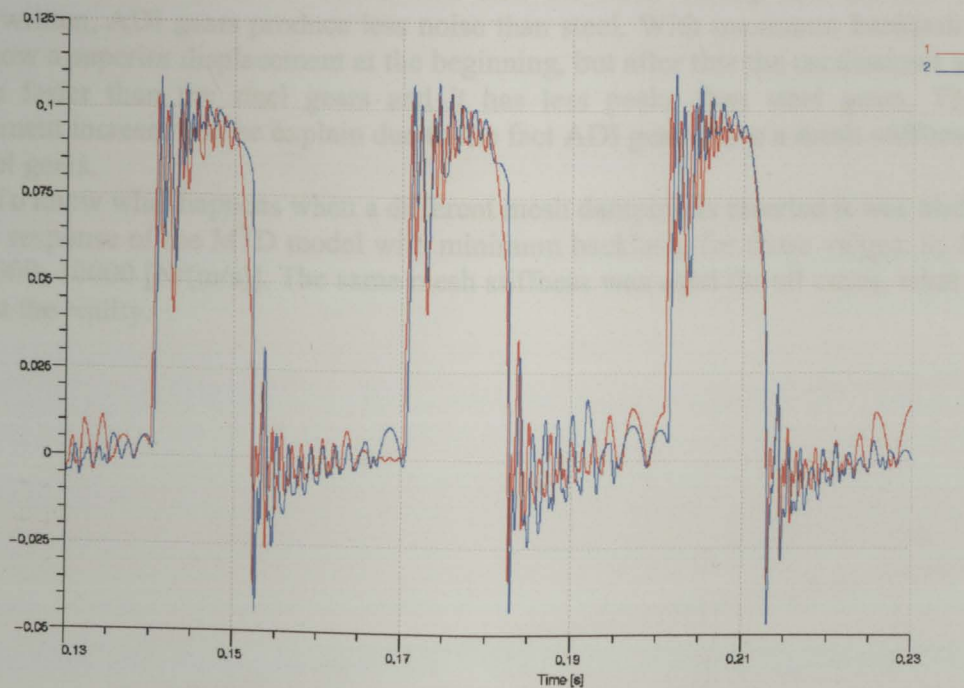


1 - M1D_1_47-47 (steel) - relative linear displacement [mm]
2 - M1D_1_47-47 (ADI) - relative linear displacement [mm]



Graph 17 - Response detail of the M1D model, meshing 47-47, with minimum backlash.

1 - M1D_2_47-47 (steel) - relative linear displacement [mm]
2 - M1D_2_47-47 (ADI) - relative linear displacement [mm]



Graph 18 - Response of the M1D model, meshing 47-47, with maximum backlash.

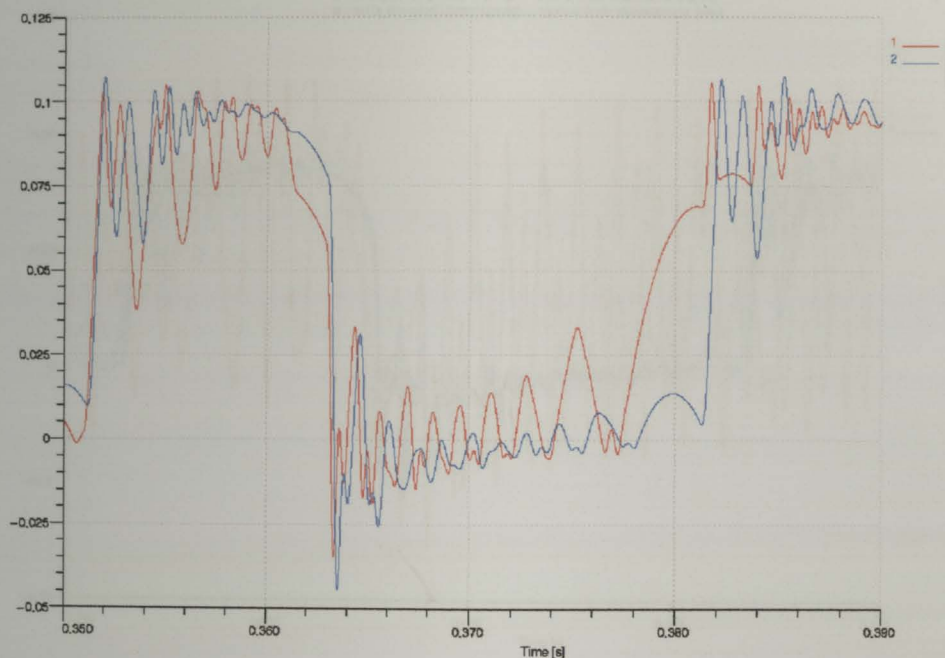


FINAL PROJECT

Experimental Comparison of the Mass Balance System
performance With Carbonitrided Steel Gears and
Austempered Ductile Iron Gears



1 - M1D_2_47-47 (steel) - relative linear displacement [mm]
2 - M1D_2_47-47 (ADI) - relative linear displacement [mm]

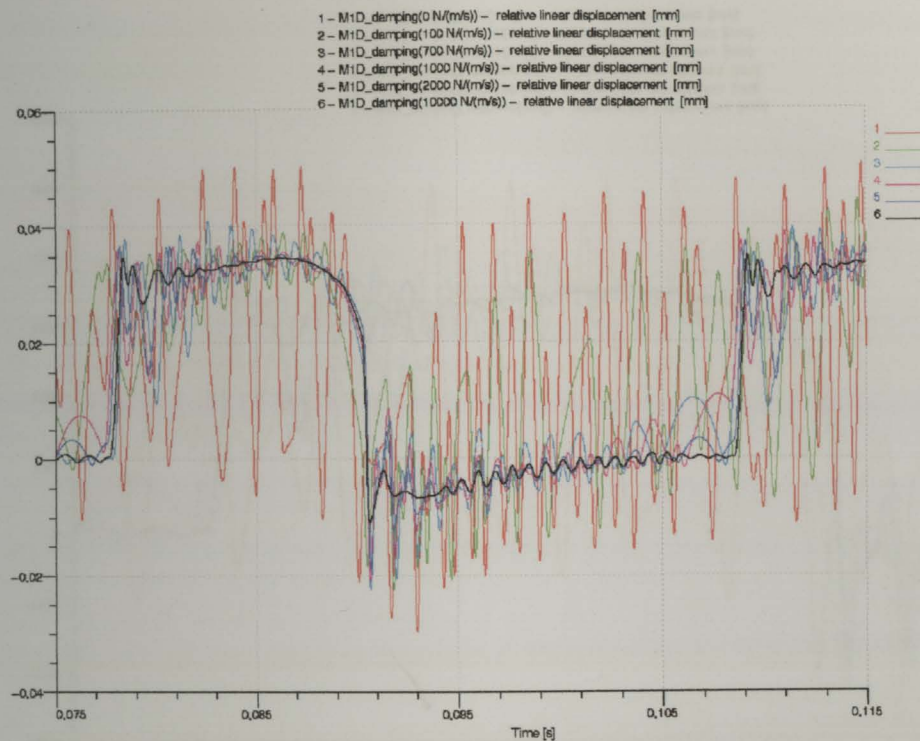


Graph 19 – Response detail of the M1D model, meshing 47-47, with maximum backlash.

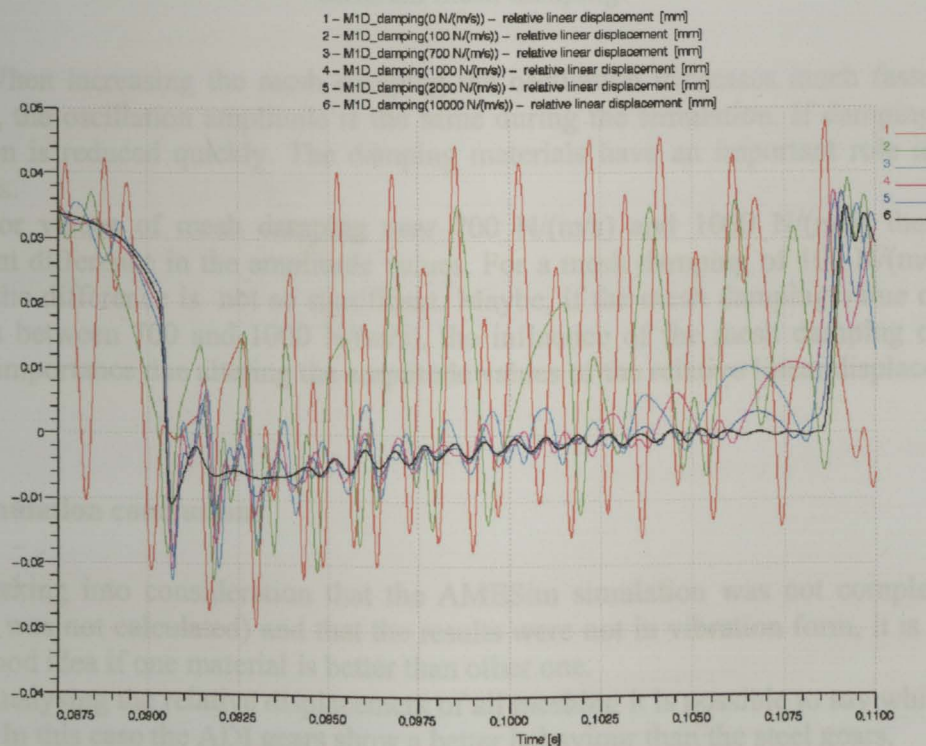
By analysing these graphs it is possible to reach the conclusion that ADI has a better behaviour because it has fewer oscillations than the steel. Taking into account what was already written, ADI gears produce less noise than steel. With maximum backlash the ADI gears show a superior displacement at the beginning, but after this the oscillation's amplitude decrease faster than the steel gears and it has less peaks than steel gears. This initial displacement increase can be explain due to the fact ADI gears have a mesh stiffness smaller than steel gears.

To know what happens when a different mesh damping is inserted it was bluit a graph with the response of the M1D model with minimum backlash, for these values: 0; 100; 700; 1000; 2000; 10000 [N/(m/s)]. The same mesh stiffness was used for all cases, what does not represent the reality.

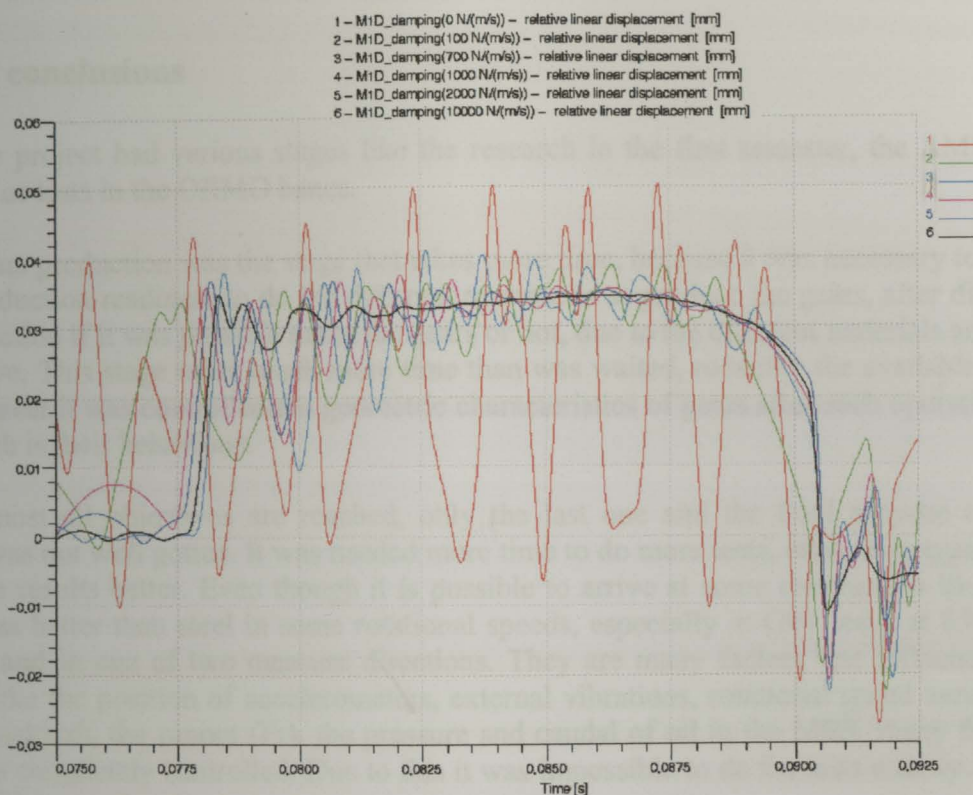




Graph 20 – Response of the M1D model, meshing 47-47, with minimum backlash and different mesh



Graph 21 – Response detail of the M1D model, meshing 47-47, with minimum backlash and different mesh damping.



Graph 22 – Response detail of the M1D model, meshing 47-47, with minimum backlash and different mesh damping.

When increasing the mesh damping the oscillation decreases much faster. Without damping, the oscillation amplitude is the same during the simulation. If damping is big the oscillation is reduced quickly. The damping materials have an important role in the gears vibrations.

For values of mesh damping near 700 N/(m/s) and 1000 N/(m/s) there exists a significant difference in the amplitude values. For a mesh damping of 100 N/(m/s) and 700 N/(m/s) the difference is not so significant. Maybe, if the mesh damping value of steel and ADI was between 700 and 1000 N/(m/s), the influence of the mesh damping could be of extreme importance due altering the amplitude values of the relative linear displacement.

11.4. Simulation conclusions

Taking into consideration that the AMESim simulation was not completely (mesh damping was not calculated) and that the results were not in vibration form, it is possible to have a good idea if one material is better than other one.

Analysing the relative displacement of all meshing it is possible to say which material is better. In this case the ADI gears show a better behaviour than the steel gears.



12. Final conclusions

The project had various stages like the research in the first semester, the AMESim simulation or tests in the ORMO bancs.

Gears production was the stage that takes more time, because it was necessary to wait for the production readiness to do the different operations to produce the gears, after discuss with production if it was possible make the gears or not, due to the different materials and the metal sleeve. This stage takes much more time than was waited, reducing the available tests time. However it was controlled the geometric characteristics of gears after each operation to know which is their behaviour.

Almost all objectives are reached, only the last one and the final propose of the project it was not well gotten. It was needed more time to do more tests, other tests types and analyse the results better. Even though it is possible to arrive at some conclusions like that the ADI has better than steel in some rotational speeds, especially in G9T tests, at 850 and 4000 rpm and in one of two measure directions. They are many factors that influence the measures like the position of accelerometers, external vibrations, rotational speed variation, different backlash, the runout (Fr), the pressure and caudal of oil in the MBS, many factors that are not completely controlled. Due to this it was impossible to do the tests exactly in the same condition.

In the AMESim simulation it was possible to verify that the ADI has a better behaviour than steel, taking into consideration that the meshing damping was not calculated and it was used the same value for both material, that do not represent the reality. It is important too, to say that AMESim results are not vibrations values, they are displacements and supposed that more variations in the displacement provoke more vibrations. It was in this way that the results were analysing.

As the experimental results were not as it was expected, the results comparison was not possible, because the simulation was only run for 1000 rpm, were not made tests at this speed. There is still a lot of work to do to obtain a complete comparison, like the fatigue tests.



FINAL PROJECT

Experimental Comparison of the Mass Balance System
performance With Carbonitrided Steel Gears and
Austempered Ductile Iron Gears



13 Further work

Exist yet some work that can be doing in the future:

- Calculate the meshing damping.
- Make more tests, like fatigue tests and tests in the motor

After to analyse the new results and if the ADI is much better than the steel, can be study too:

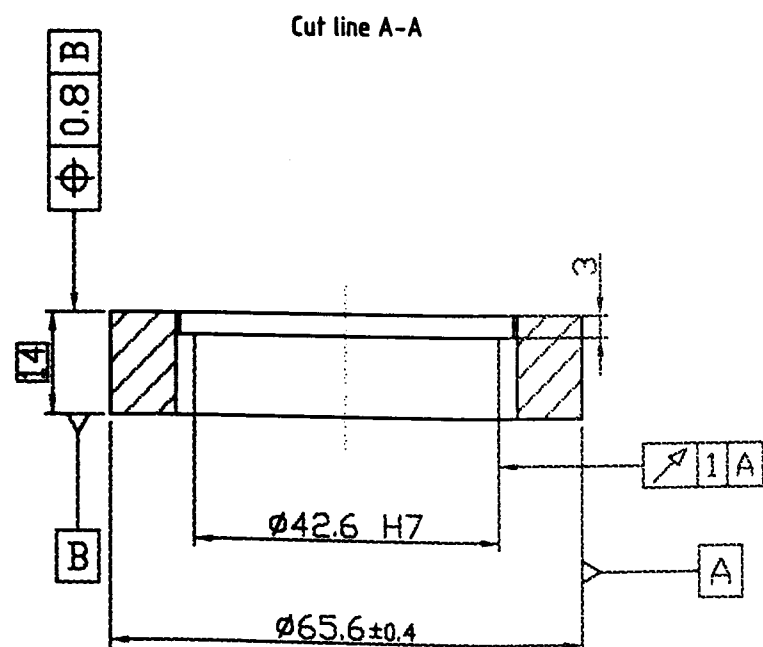
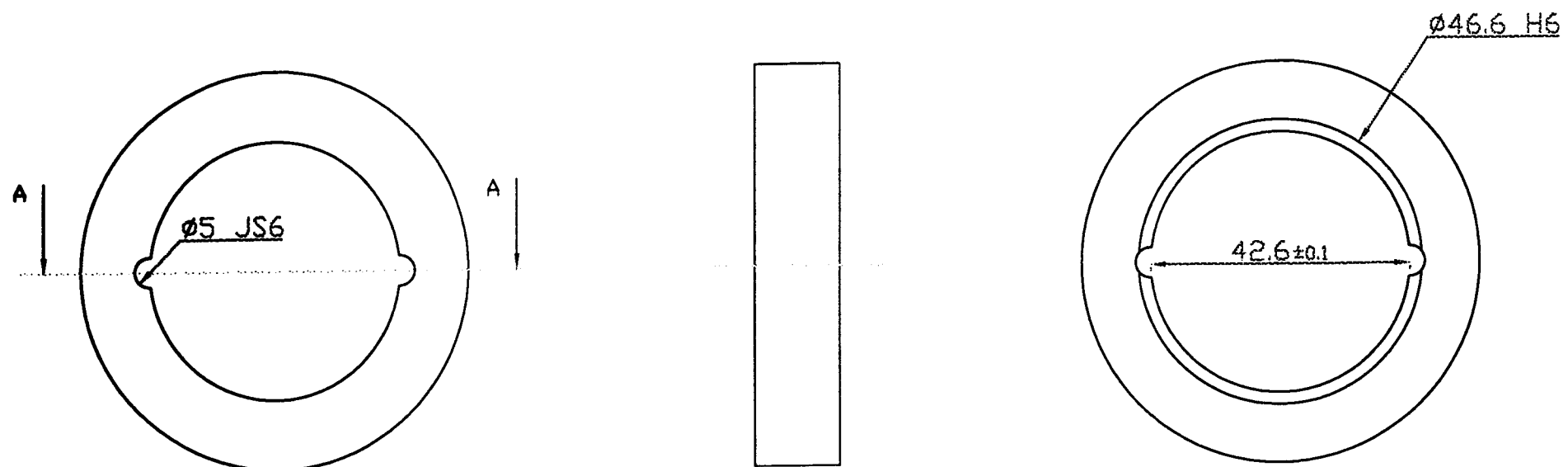
- Characterise the ADI behaviour during the production process and analyse if this is the better process or not.
- Make an analysis of costs with the change material, verify if exist benefits.



FINAL PROJECT
Experimental Comparison of the Mass Balance System
performance With Carbonitrided Steel Gears and
Austempered Ductile Iron Gears

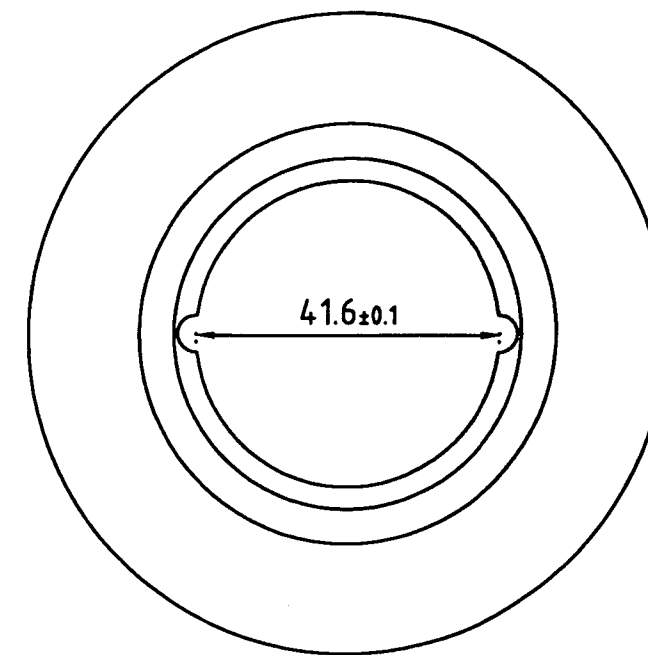
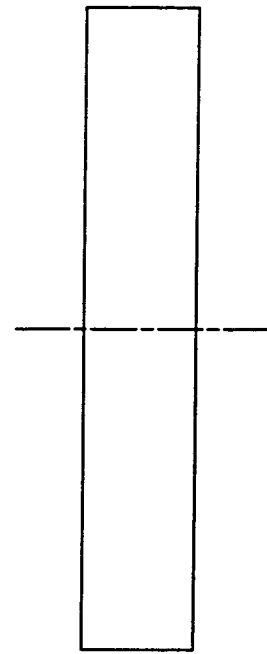
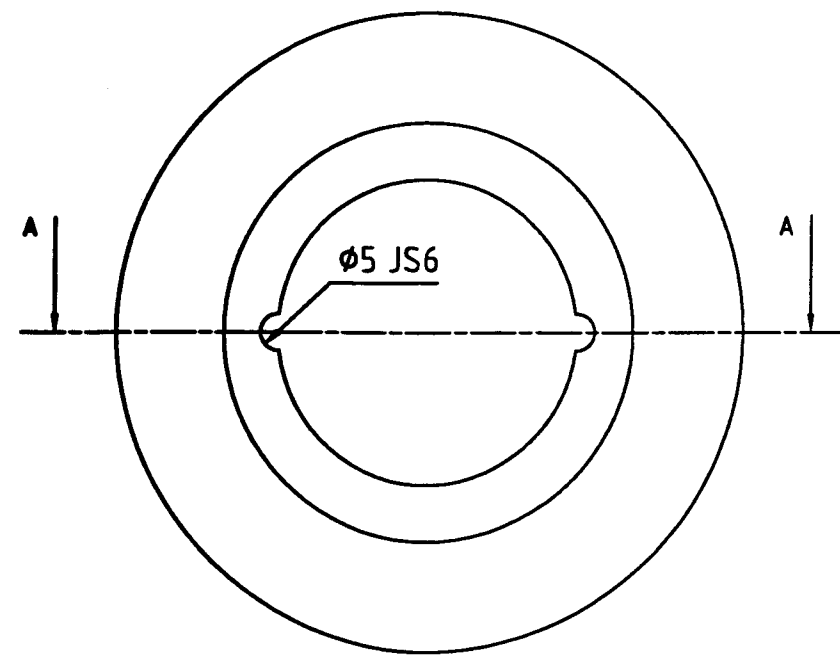


Appendix A

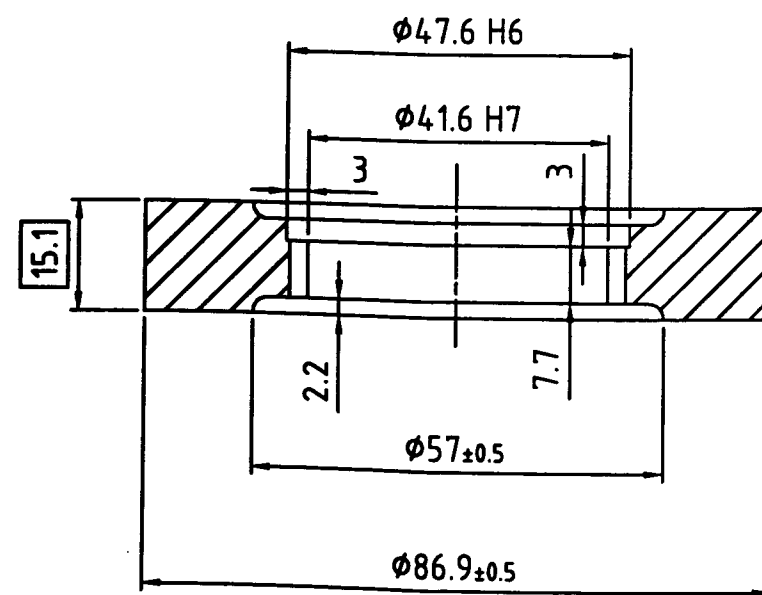


	Dates	Names	Mass Balance System Wheel G9T - 079	Henrique Pinto
Designed by				
Checked by				
Approved by				FEUP
Scale 1:1				N.º 001

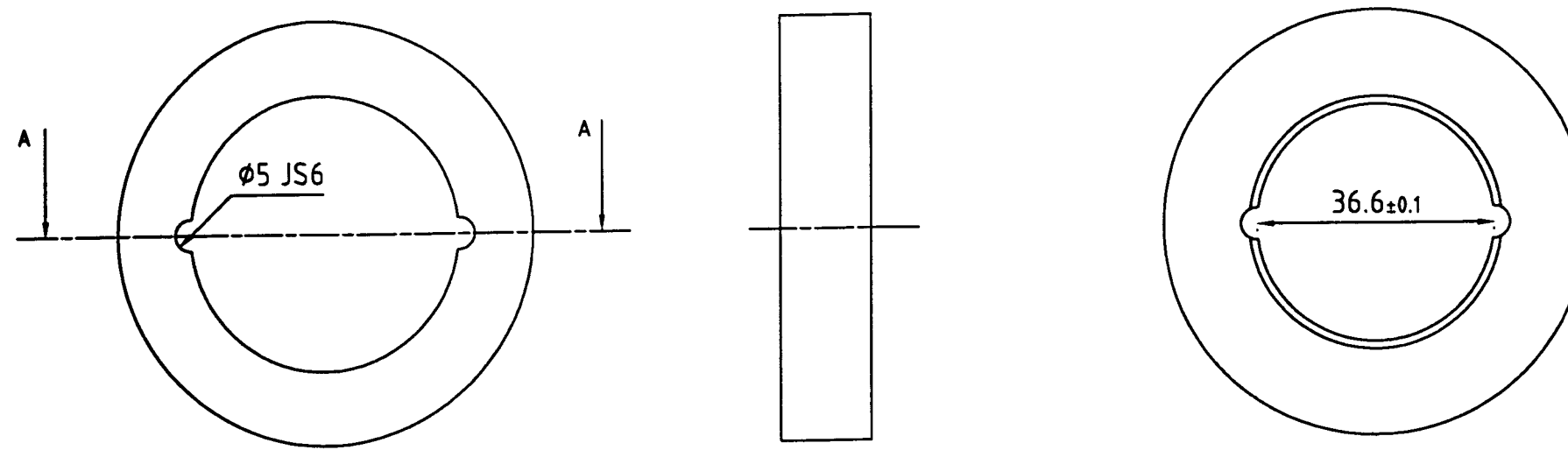
Final Project
Machine Design



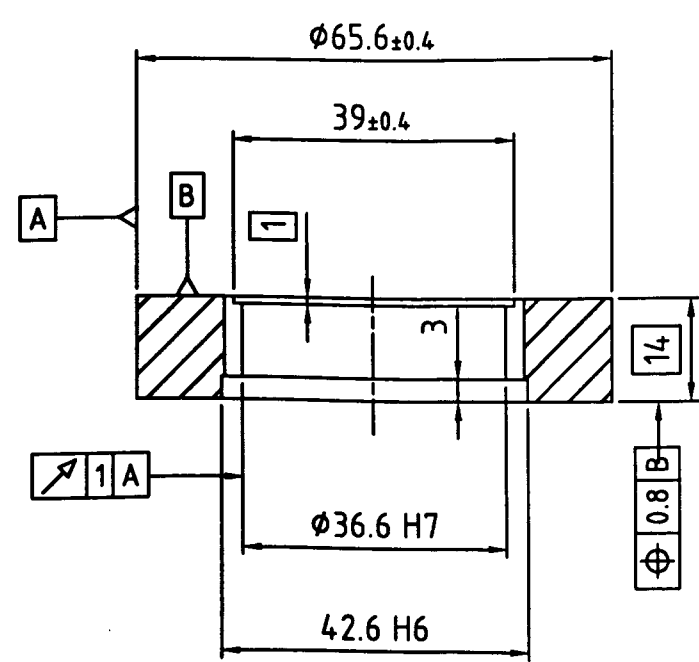
Cut line A-A



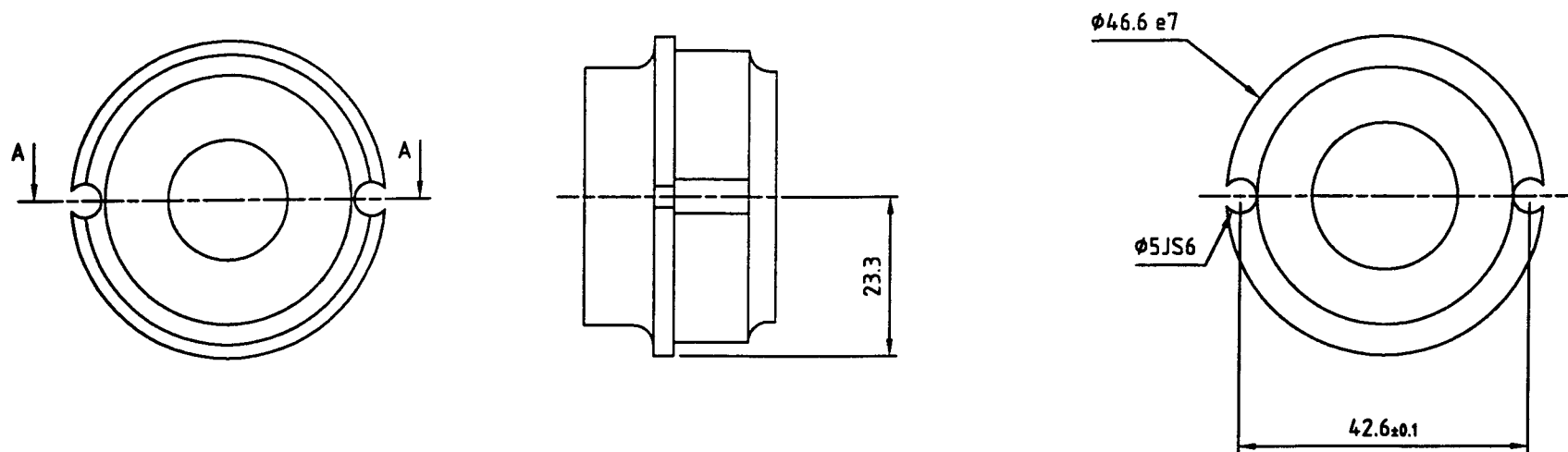
	Date	Names	Mass Balance System Wheel G9T - 080	Henrique Pinto
Designed by				
Checked by				
Approved by			Final Project Machine Design	FEUP
Escala 1:1				N.º 002



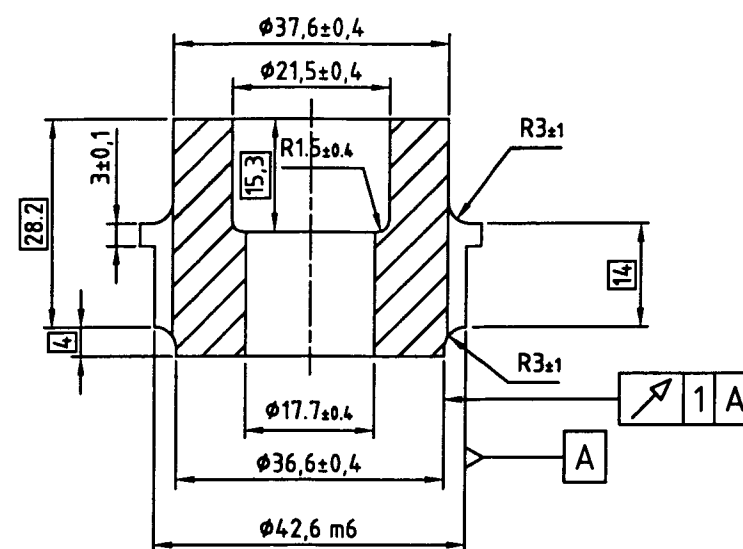
Cut line A-A



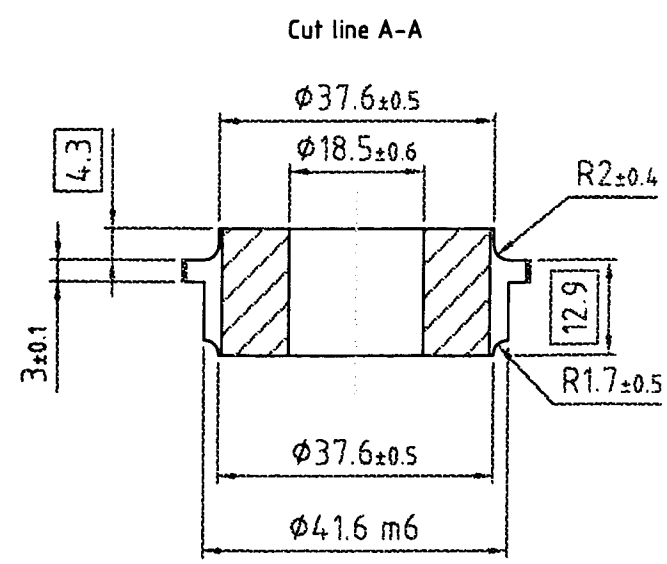
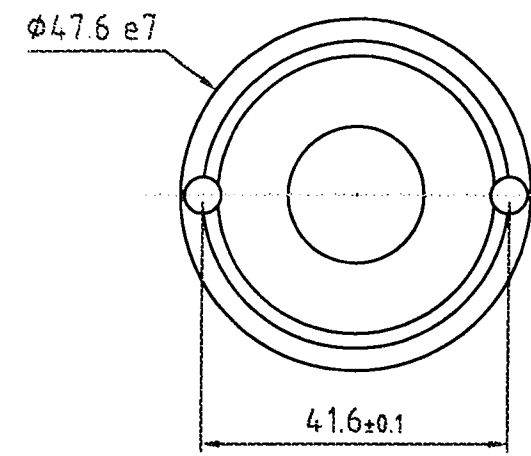
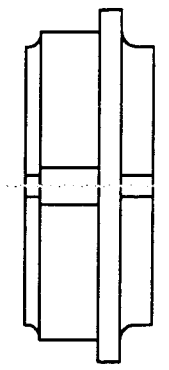
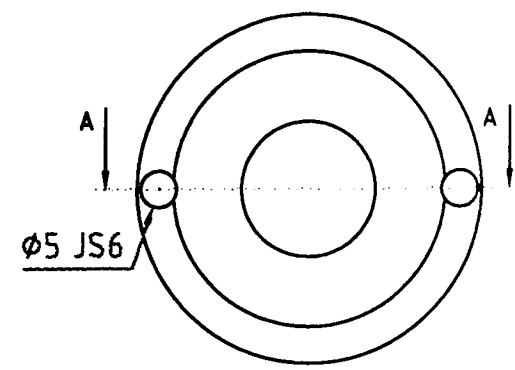
	Date	Names	Mass Balance System Wheel G9T - 079	Henrique Pinto
Designed by				
Checked by				FEUP
Approved by			Final Project Machine Design	N.º 003
Scale 1:1				



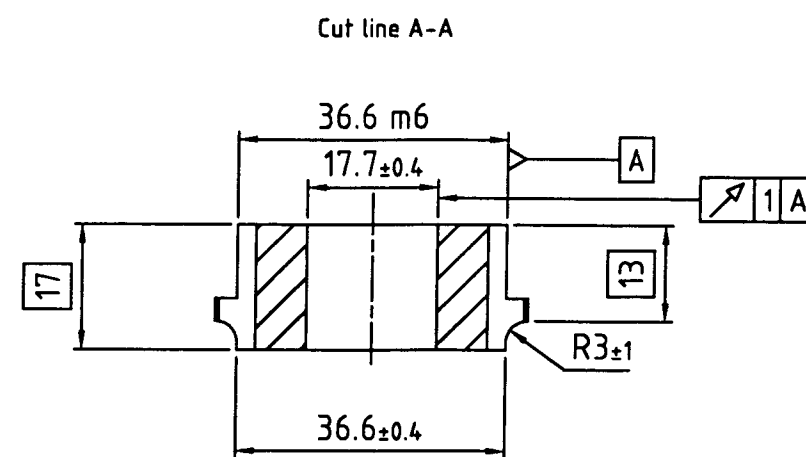
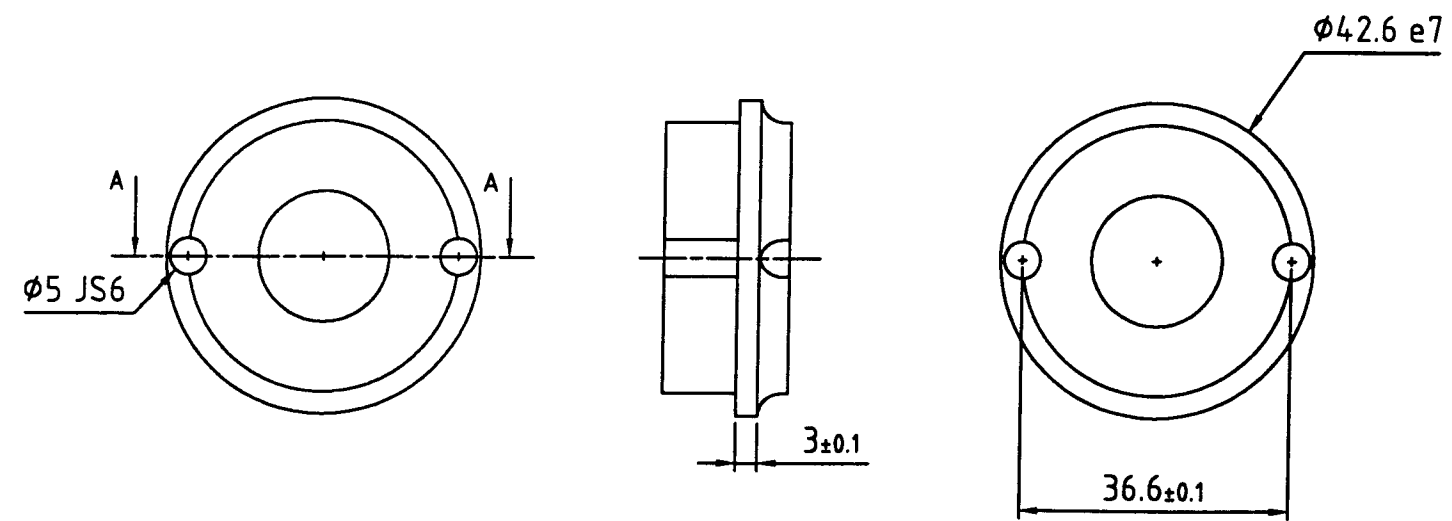
Cut line A-A



	Date	Names	Mass Balance System Metal Sleeve G9T - 079	Henrique Pinto
Designed by				
Checked by				
Approved by				FEUP
Scale 1:1	Final Project Machine Design			N.º 004

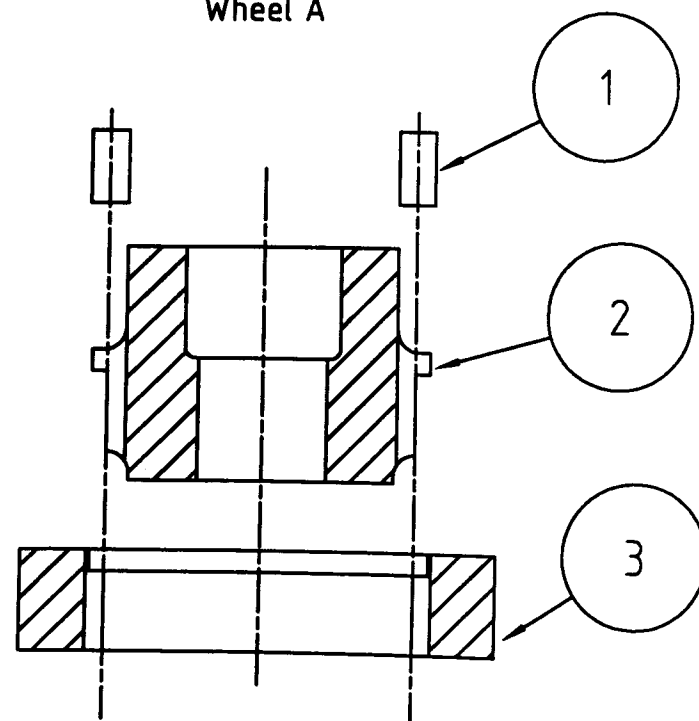


	Date	Names	Mass Balance System Metal Sleeve G9T - 080	Henrique Pinto
Designed by				FEUP
Checked by				N.º 005
Approved by				
Scale 1:1	Final Project Machine Design			
				

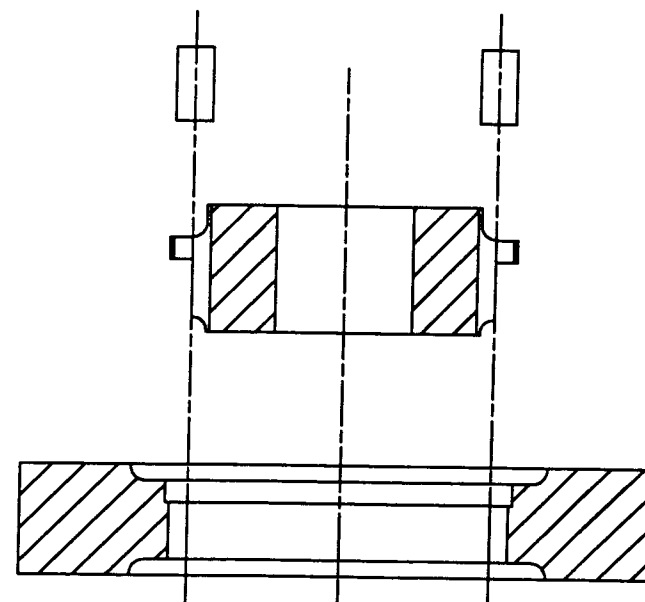


	Date	Names	Mass Balance System Metal Sleeve G9T - 078	Henrique Pinto
Designed by				
Checked by				
Approved by				FEUP
Scale 1:1	Final Project Machine Design			N.º 006

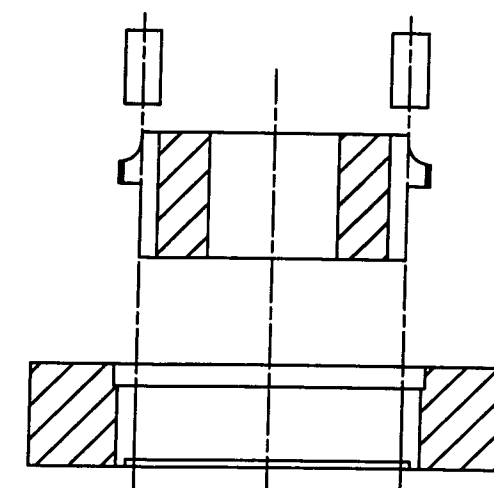
Wheel A



Wheel B



Wheel C



Parts List			
Name	Item	Qty	Material
Pin	1	6	Steel
Metal sleeve	2	3	Cast Iron
Wheel	3	3	Steel

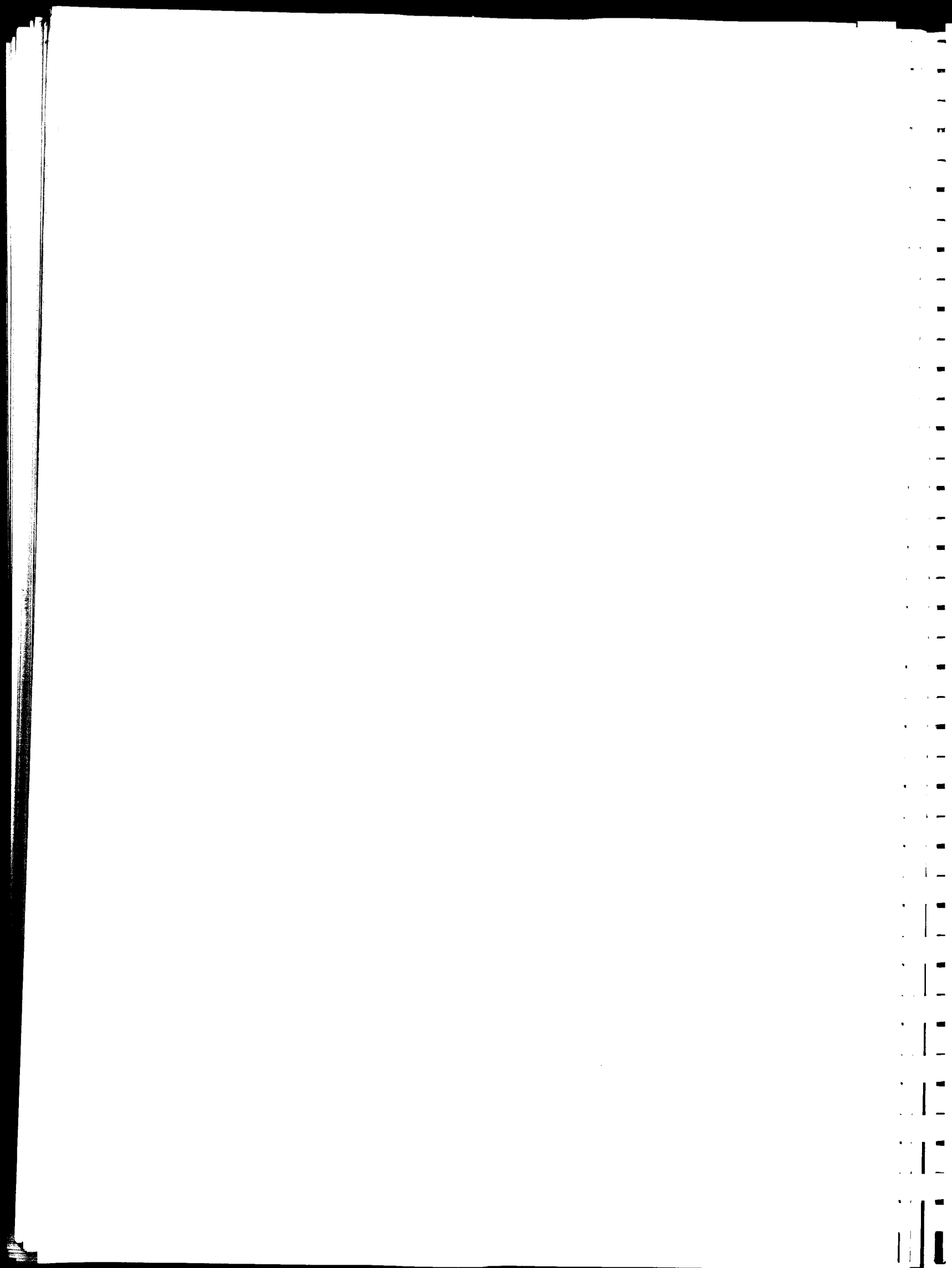
	Date	Names	Mass balance System G9T Metal sleeve assembly	Henrique Pinto	
Desegined by					
Checked by					
Approved by					FEUP
Scale 1:1	Final Project Machine Disegn			N.º 007	
					

[illegible]

Technical drawing of a circular part. The drawing shows three concentric circles. A horizontal dimension line with arrows at both ends passes through the center of the circles. The dimension value is $\phi 81,4 \pm 0,4$. The letter 'B' is written above the left arrow of the dimension line. The letter 'B' is also written above the right arrow of the dimension line. The dimension line is labeled with the number 1.

[illegible]

	Date	Names	Masss Balance System - M1D Wheels	Henrique Pinto
Designed by				
Checked by				
Approved by				
Scale 1:1	Final Project Machine Design			N.º 008
				





FINAL PROJECT

Experimental Comparison of the Mass Balance System
performance With Carbonitrided Steel Gears and
Austempered Ductile Iron Gears



Universidade do Porto
Faculdade de Engenharia

FEUP



CACIA

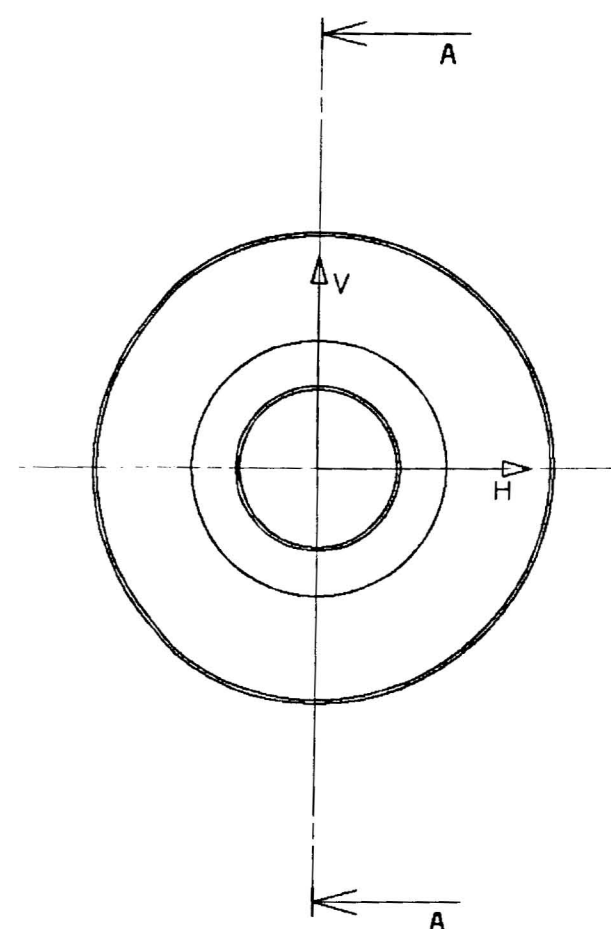
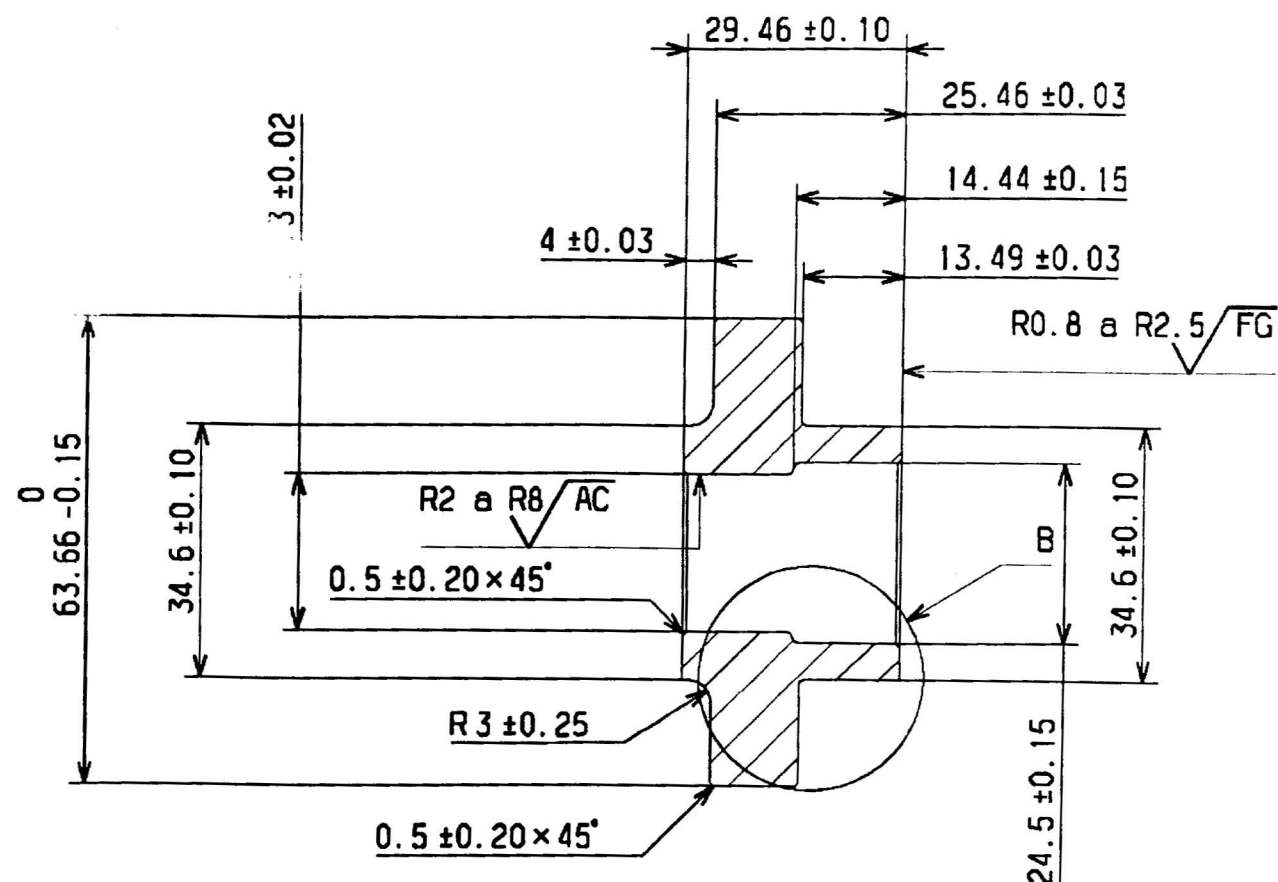


RENAULT



Appendix B

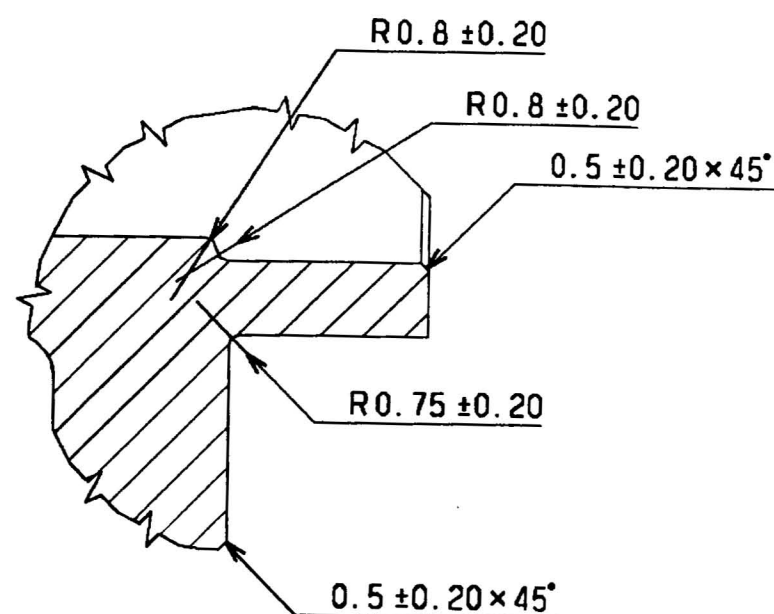
A-A



NOTA 1: MATIERE: FONTE NODULAIRE
DURETE EN SURFACE: 226 HV

NOTA 2: RUGOSITE GENERALE $R 10/\sqrt{\quad}$
SAUF INDICATIONS CONTRAIRES

DETAIL : B
ECHELLE : 2:1



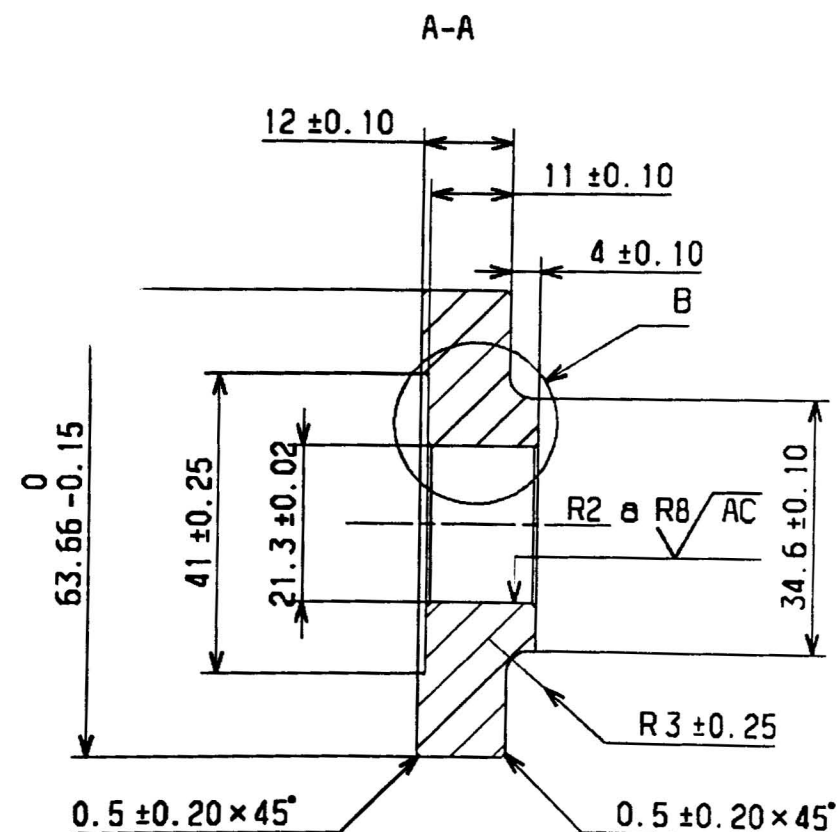
Objectif fonctionnel PLAN DE SPECIFICATION / Scope SPECIFICATION DRAW			
Etat de la planche Sheet state	Echelle Scale	Dessiné le par Date of Issue by	PIGNON ARBRE EQUILIBRAGE + PINION-BAL SHAFT
+	1:1	11/03/2004 H. Pinto	
Symbole S/R S/R Symbol		Service Department	Numero de plan / Drawing number
		DTC/BEE	1020105001
		Utilisateurs Users	Modif planche Sheet revision level
			--
			Planche Sheet
			1/1
			Type planche Sheet type
			+
			Format
			A3
			RENAULT

Ce plan est la propriété de RENAULT - 92109 Boulogne Billancourt Cedex (France) et ne peut être reproduit ou communiqué sans son autorisation.
This drawing is the property of RENAULT - 92109 Boulogne Billancourt Cedex (France) and cannot be duplicated or circulated without its permission.

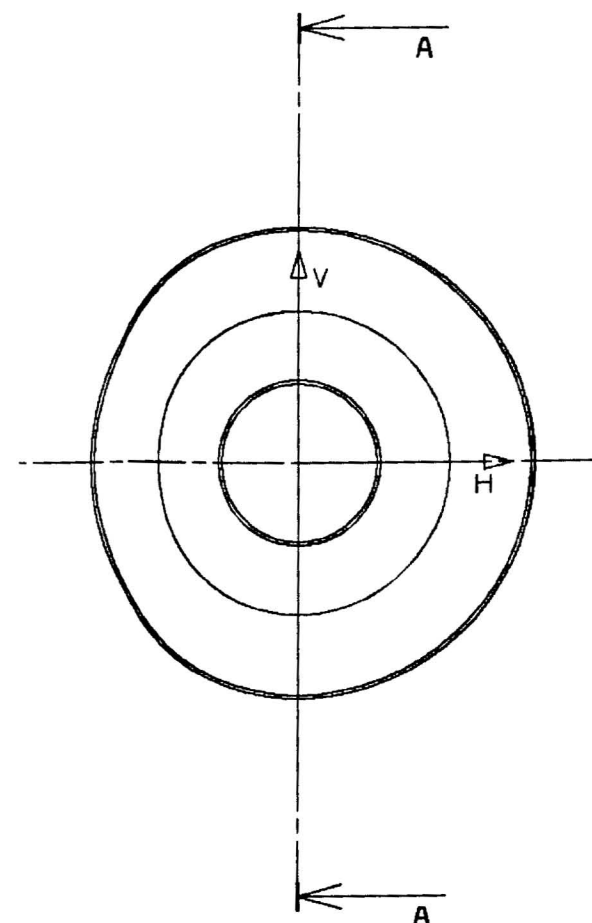
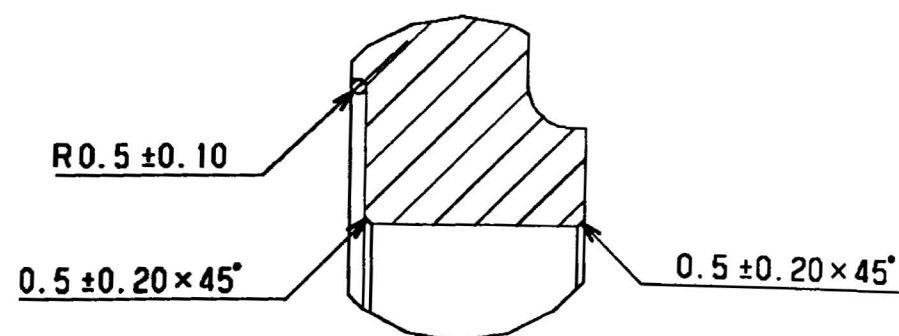


NOTA 1: MATIERE: FONTE NODULAIRE
DURETE EN SURFACE: 226 HV

NOTA 2: RUGOSITE GENERALE $R 10 \sqrt{\quad}$
SAUF INDICATIONS CONTRAIRES



DETAIL : B
ECHELLE : 2:1



Objectif fonctionnel PLAN DE SPECIFICATION / Scope SPECIFICATION DRAW			
Etat de la planche Sheet state	Echelle Scale	Dessiné le par Date of Issue by	PIGNON ARBRE EQUILIBRAGE + PINION-BAL SHAFT
+	1:1	12/03/2004 H. PINTO	
Symbole S/R S/R Symbol		Service Department	Numero de plan / Drawing number 1020105002
		Utilisateurs Users	
			Modif planche Sheet revision level

			Planche Sheet
			+
			Type planche Sheet type
			+
			Format A3



Ce plan est la propriété de RENAULT - 92109 Boulogne Billancourt Cedex (France) et ne peut être reproduit ou communiqué sans son autorisation.
This drawing is the property of RENAULT - 92109 Boulogne Billancourt Cedex (France) and cannot be duplicated or circulated without its permission.

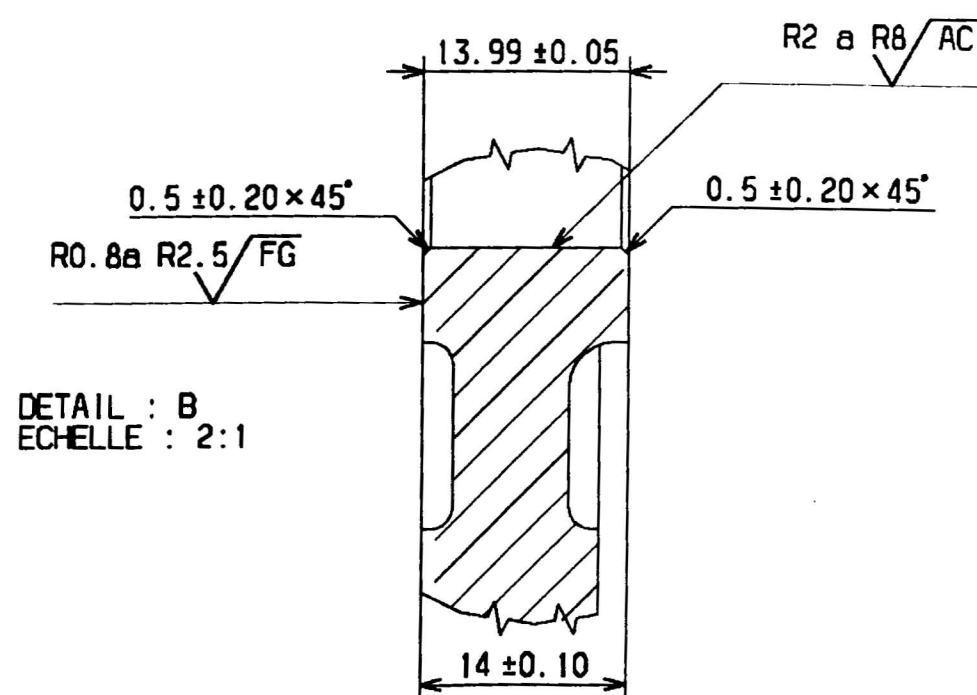
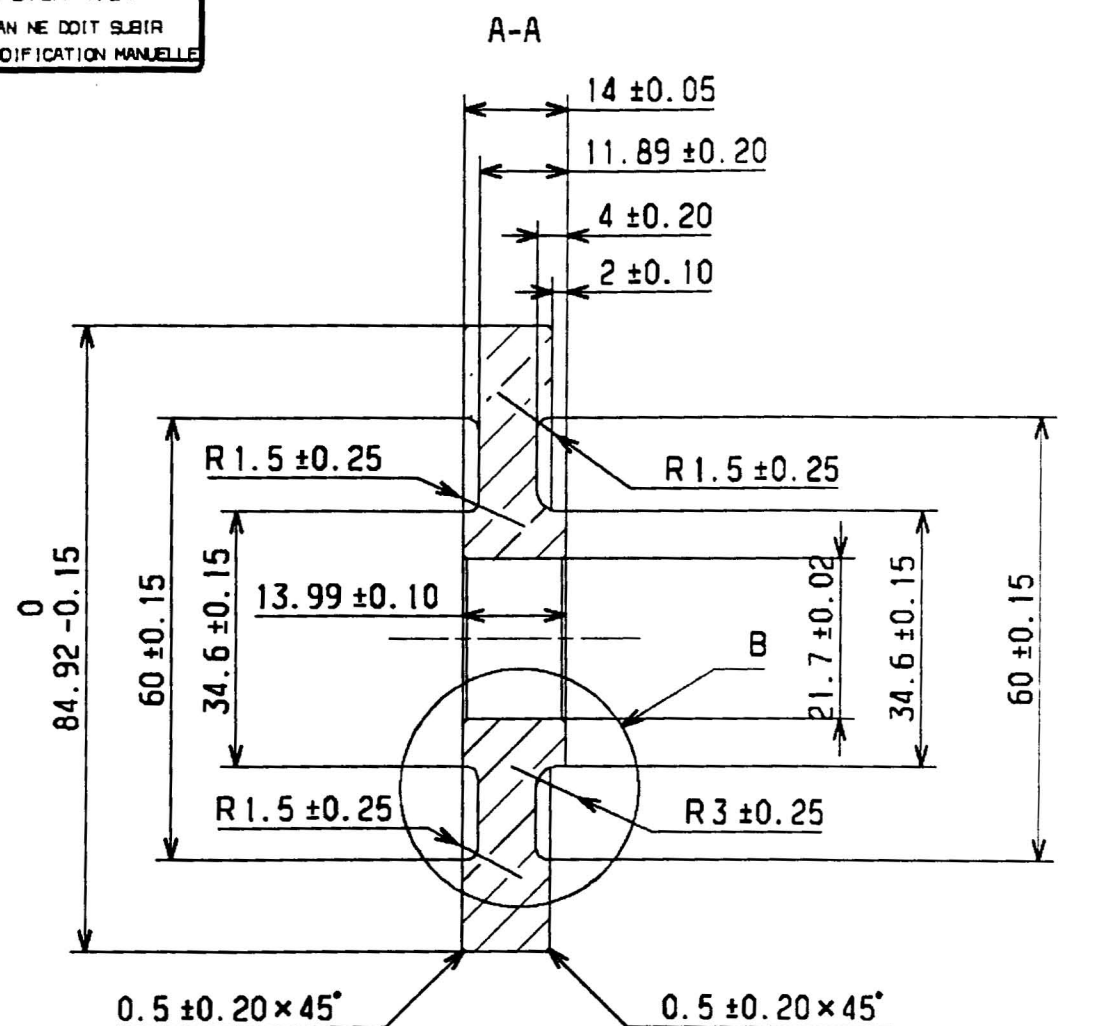
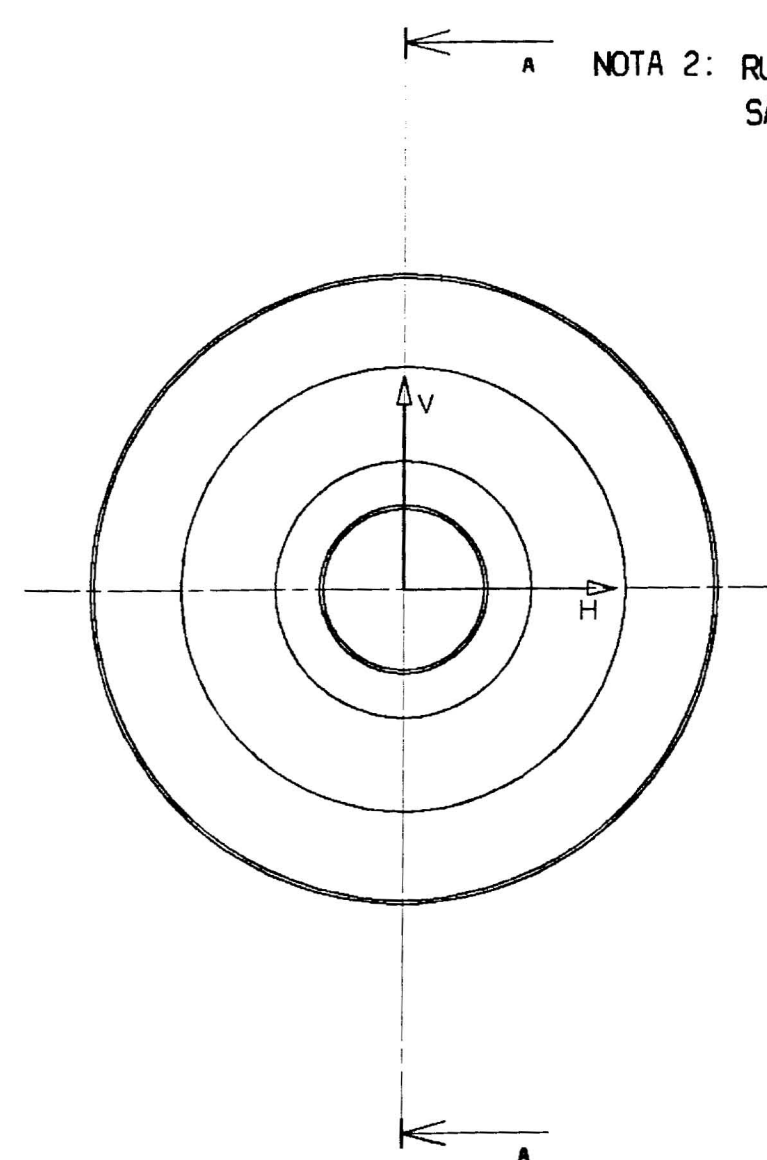
0 10 20 30 40 50 60 70 80 90 100mm

CONCEPTION ASSISTEE
PAR ORDINATEUR
CATIA
Version 4.24
CE PLAN NE DOIT SUBIR
AUCUNE MODIFICATION MANUELLE

NOTA 1: MATERIE: FONTE NODULARE

DURETE EN SURFACE: 226 HV

NOTA 2: RUGOSITE GENERALE R_{10} ✓
SAUF INDICATIONS CONTRAIRES



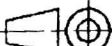
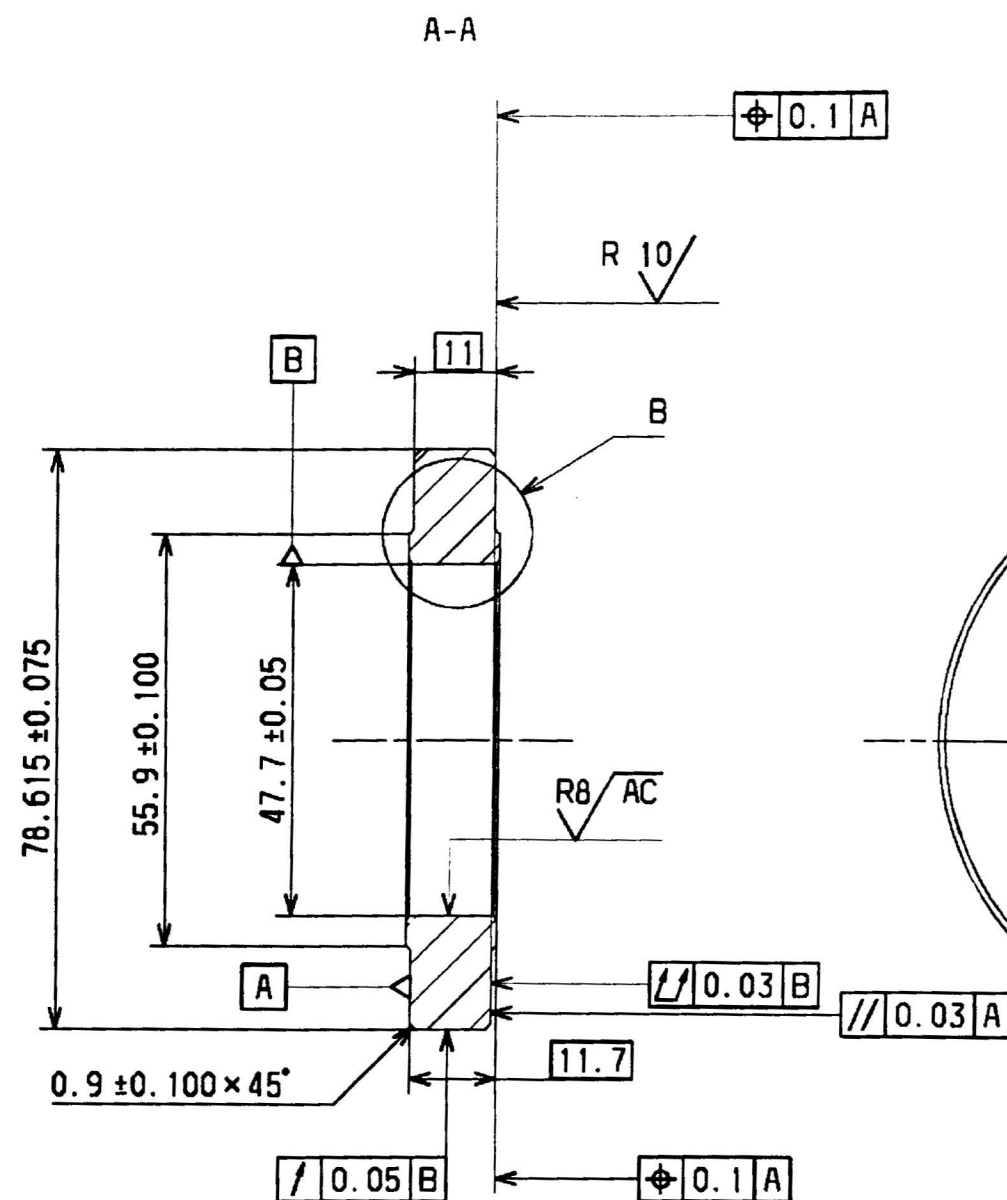
Objectif fonctionnel Etat de la planche Sheet state		PLAN DE SPECIFICATION		/Scale SPECIFICATION DRAW	
Echelle Scale		Dessiné le par Date of Issue by		PIGNON ARBRE EQUILIBRAGE + PINION-BAL SHAFT + Eng/ish Français	
+		12/03/2004 H. PINTO ---			
Symbole S/R S/R Symbol					
		Service Département		Numéro de plan / Drawing number	
Utilisateurs Users		DTC/BEE		1020105003	
				Modif planche Sheet revision level	

				Planche Sheet	
				1/1	
				Type planche Sheet type	
				+	
				Format	
				A3	
		Ce plan est la propriété de RENAULT - 92109 Boulogne Billancourt Cedex (France) et ne peut être reproduit ou communiqué sans son autorisation. <i>This drawing is the property of RENAULT - 92109 Boulogne Billancourt Cedex (France) and cannot be duplicated or circulated without its permission.</i>		 RENAULT	

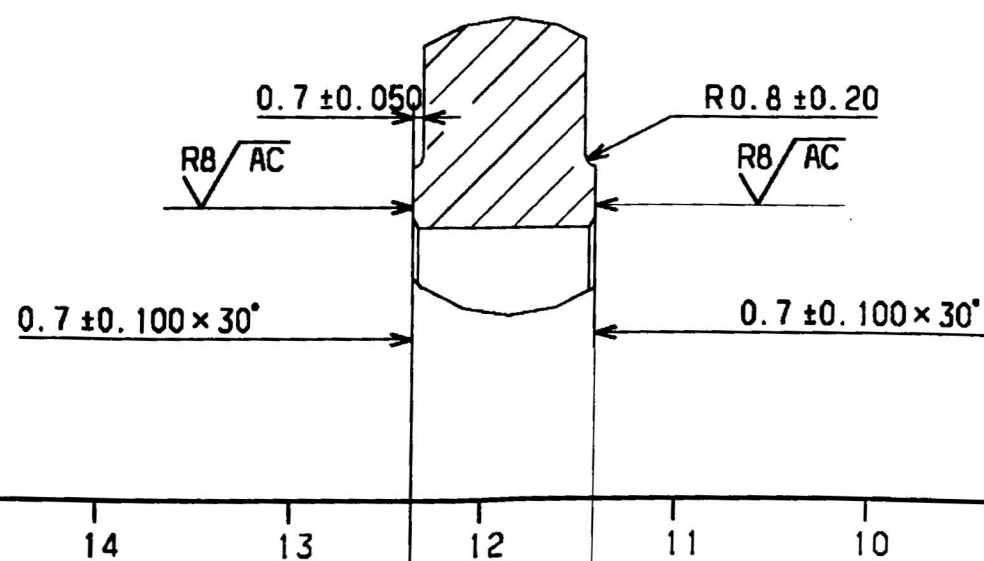


NOTA 1 MATIERE: FONTE NODULAIRE

DURETE EN SURFACE: 226 HV

NOTA 2: RUGOSITE GENERALE $R 10/\sqrt{\quad}$
SAUF INDICATIONS CONTRAIRES

DETAIL : B
ECHELLE : 2:1

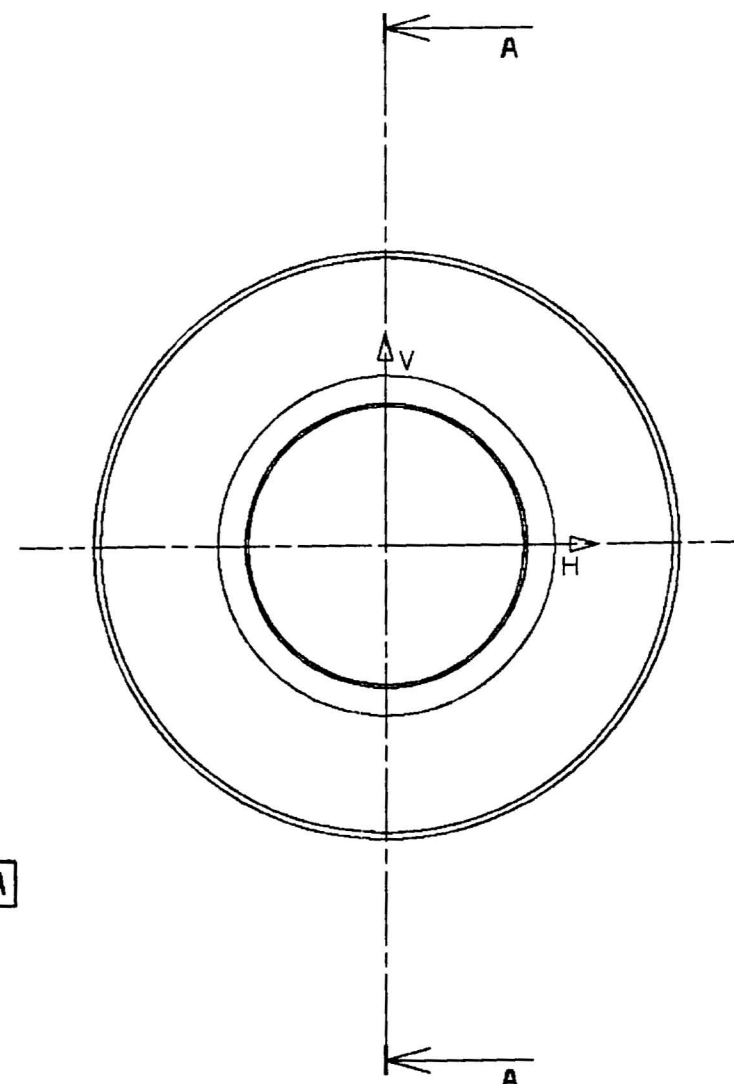
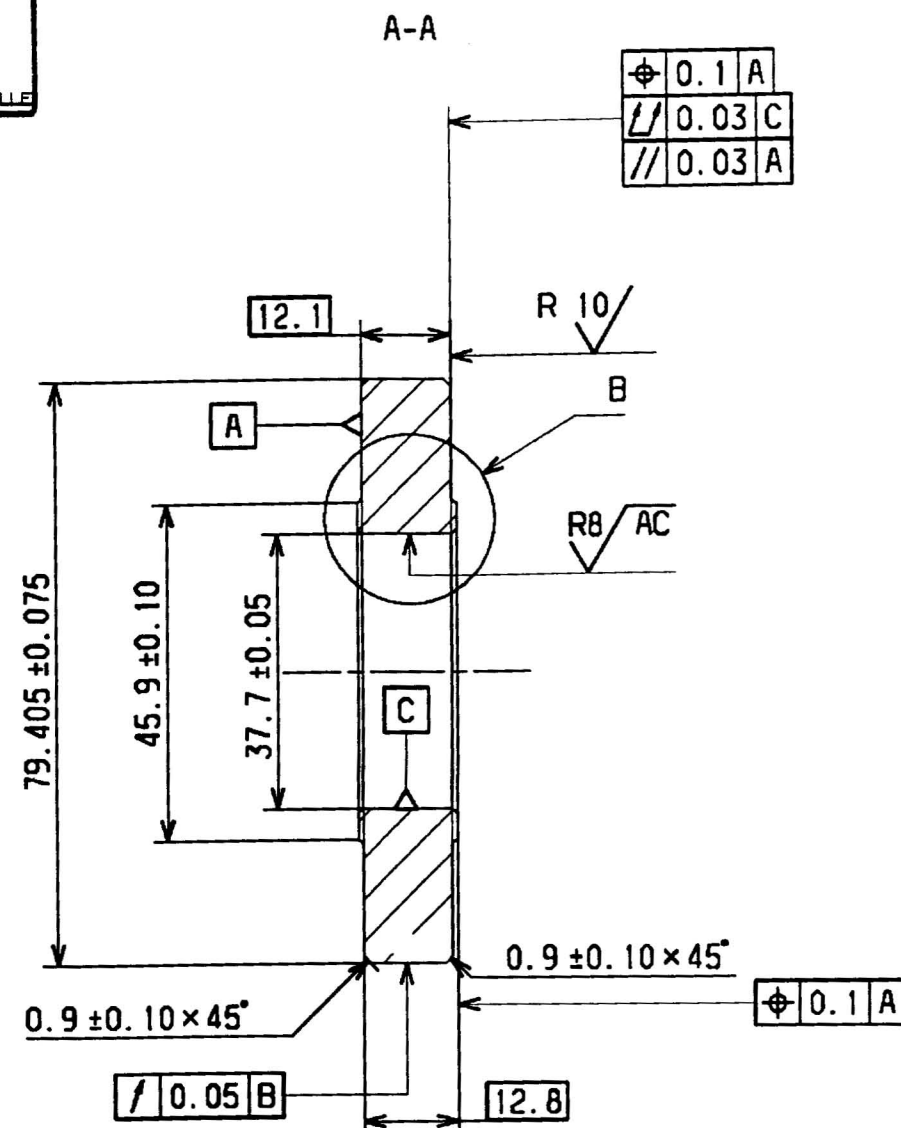


Objectif fonctionnel PLAN DE SPECIFICATION / Scope SPECIFICATION DRAW			
Etat de la planche Sheet state	Echelle Scale	Dessiné le par Date of Issue by	PIGNON ARBRE EQUILIBRAGE + PINION-BAL SHAFT
+	1:1	11/03/2004 H. PINTO	
Symbole S/R S/R Symbol		Service Department	Numero de plan / Drawing number 1020105004
		DTC/BEE	
Utilisateurs Users		Modif planche Sheet revision level	Planche Sheet
+		---	1/1
+		Type planche Sheet type	Format
+		+	A3
Ce plan est la propriété de RENAULT - 92109 Boulogne Billancourt Cedex (France) et ne peut être reproduit ou communiqué sans son autorisation. This drawing is the property of RENAULT - 92109 Boulogne Billancourt Cedex (France) and cannot be duplicated or circulated without its permission.			

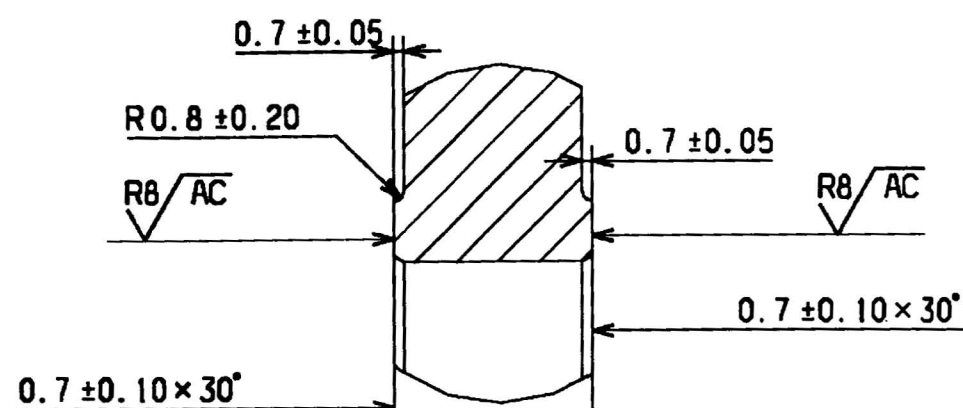


NOTA 1 MATIERE: FONTE NODULAIRE
DURETE EN SURFACE: 226 HV

NOTA 2: RUGOSITE GENERALE $R 10/\sqrt{\quad}$
SAUF INDICATIONS CONTRAIRES



DETAIL : B
ECHELLE : 2:1



Objectif fonctionnel PLAN DE SPECIFICATION / Scope SPECIFICATION DRAW			
Etat de la planche Sheet state	Echelle Scale	Dessiné le par Date of Issue by	PIGNON ARBRE EQUILIBRAGE + PINION-BAL SHAFT
+	1:1	11/03/2004 H. PINTO	
Symbole S/R S/R Symbol		Service Department	Numero de plan / Drawing number 1020105005
		Utilisateurs Users	
			Modif planche Sheet revision level
			+
			Type planche Sheet type
			+
			Format A3



Ce plan est la propriété de RENAULT - 92109 Boulogne Billancourt Cedex (France) et ne peut être reproduit ou communiqué sans son autorisation.
This drawing is the property of RENAULT - 92109 Boulogne Billancourt Cedex (France) and cannot be duplicated or circulated without its permission.

0 10 20 30 40 50 60 70 80 90 100mm



Appendix C

C.1. Dimensional control after turning.

Pinions M1D

HP1-201

Thickness				Outside diameter				Inside diameter			
Nominal dimension	Toler.	Measured	Error	Nominal dimension	Toler.	Measured	Error	Nominal dimension	Toler.	Measured	Error
11		11,074	0,074	78,615	0,075	78,656	0,041	47,700	0,050	47,643	-0,057
11		11,041	0,041		-0,075				-0,050		
11		10,997	-0,003								
Average		11,037	0,037								

HP2-202

Thickness				Outside diameter				Inside diameter			
Nominal dimension	Toler.	Measured	Error	Nominal dimension	Toler.	Measured	Error	Nominal dimension	Toler.	Measured	Error
11		11,099	0,099	78,656	0,075	78,625	-0,031	47,700	0,050	47,630	-0,070
11		10,949	-0,051		-0,075				-0,050		
11		11,053	0,053								
Average		11,034	0,034								

HP3-203

Thickness				Outside diameter				Inside diameter			
Nominal dimension	Toler.	Measured	Error	Nominal dimension	Toler.	Measured	Error	Nominal dimension	Toler.	Measured	Error
12,1		11,989	-0,111	79,405	0,075	79,426	0,021	37,700	0,050	37,638	-0,062



FINAL PROJECT

Experimental Comparison of the Mass Balance System
performance With Carbonitrided Steel Gears and
Austempered Ductile Iron Gears



12,1	12,153	0,053
12,1	11,975	-0,125
Average	12,039	-0,061

-0,075

-0,050

Pinions G9T

HP4-103				Inside diameter			
Thickness							
Nominal dimension	Toler.	Measured	Error	Nominal dimension	Toler.	Measured	Error
12		11,861	-0,139	21,7	0,02	21,623	-0,077
12		12,158	0,158		-0,02		
12		12,027	0,027				
Average		12,015	0,015				

HP5-103				Outside diameter			
Thickness							
Nominal dimension	Toler.	Measured	Error	Nominal dimension	Toler.	Measured	Error
12		11,695	-0,305	84,92	0	84,931	0,011
12		12,031	0,031		-0,15		
12		11,823	-0,177				
Average		11,850	-0,150				

HP6-101



Thickness				Outside diameter			Inside diameter		
Nominal dimension	Toler.	Measured	Error	Nominal dimension	Toler.	Measured	Nominal dimension	Toler.	Measured
10,97		11,809	0,839	63,66	0	63,687	21,3	0,02	21,244
10,97		12,113	1,143		-0,15			-0,02	
10,97		11,915	0,945						
Average									
									-0,056

Thickness				Outside diameter			Inside diameter		
Nominal dimension	Toler.	Measured	Error	Nominal dimension	Toler.	Measured	Nominal dimension	Toler.	Measured
10,97		12,178	1,208	63,66	0	63,683	21,3	0,02	21,247
10,97		12,065	1,095		-0,15			-0,02	
10,97		12,222	1,252						
Average									
									-0,053

Thickness				Outside diameter			Inside diameter		
Nominal dimension	Toler.	Measured	Error	Nominal dimension	Toler.	Measured	Nominal dimension	Toler.	Measured
12		12,09	0,09	63,66	0	63,731	21,3	0,02	21,254
12		12,118	0,118		-0,15			-0,02	
12		12,145	0,145						
Average									
									-0,046

Thickness				Outside diameter			Inside diameter		
-----------	--	--	--	------------------	--	--	-----------------	--	--



FINAL PROJECT

Experimental Comparison of the Mass Balance System
performance With Carbonitrided Steel Gears and
Austempered Ductile Iron Gears



Nominal dimension	Toler.	Measured	Error	Nominal dimension	Toler.	Measured	Error	Nominal dimension	Toler.	Measured	Error
12		11,905	-0,095	63,66	0	63,74	0,08	21,3	0,02	21,231	-0,069
12		12,104	0,104		-0,15				-0,02		
12		11,95	-0,05								
Average		11,986	-0,014								

HP6-101

Total thickness				Neckband height				Tack height			
Nominal dimension	Toler.	Measured	Error	Nominal dimension	Toler.	Measured	Error	Nominal dimension	Toler.	Measured	Error
29,46	0,1	29,502	0,042	13,49	0,03	13,436	-0,054	4	0,03	3,992	-0,008
	-0,1	29,413	-0,047		-0,03	13,326	-0,164		-0,03	3,929	-0,071
		29,382	-0,078			13,311	-0,179			3,856	-0,144
Average		29,432	-0,028	Average		13,358	-0,132	Average		3,926	-0,074

HP7-101

Total thickness				Neckband height				Tack height			
Nominal dimension	Toler.	Measured	Error	Nominal dimension	Toler.	Measured	Error	Nominal dimension	Toler.	Measured	Error
29,46	0,1	29,487	0,027	13,49	0,03	13,416	-0,074	4	0,03	4,014	0,014
	-0,1	29,418	-0,042		-0,03	13,346	-0,144		-0,03	3,917	-0,083
		29,414	-0,046			13,336	-0,154			3,887	-0,113
Average		29,440	-0,020	Average		13,366	-0,124	Average		3,939	-0,061



Appendix D

**D.1. M-file to calculate dimension over ball.**

%This calculus it was made following the Henriot book. Pag. 682 until pag. 685.

%%%%%%%%%Data to introduce%%%%%%%%

%Ball diameter (mm)

a=2.6;

%Normal pressure angle (degrees)

alpha0=20;

%Normal module

m0=1.4;

%Normal helix angle

beta0=27.5;

%Teeth number

z=49;

%Profile theoretical displacement (x*m0)

x=-0.986;

%Over thickness that we want to (mm)

overth=0.08;

%%%%%%%%%Program%%%%%%%%%

fprintf('Over thikness: %6.5f\n', overth)

%Apparent pressure angle

alpat=solve('tan(alpat0*pi/180)=tan(alpha0*pi/180)/cos(beta0*pi/180)', 'alpat0');

alpat0=eval(alpat);

%Apparent module

mt0=m0/cos(beta0*pi/180);

%Reference pitch radius

r=(z/2)*mt0;



```

%Base helix angle
betab=solve('tan(betab*pi/180)=tan(beta0*pi/180)*cos(alphat0*pi/180)', 'betab');
betab=eval (betab);

%Fictitious ball diameter
%af=a/cos(betab);

%Apparent thickness
st=(pi*mt0/2)+2*x*tan(alphat0*pi/180)+overth*2;

%Thickness between two teeth
et=pi*mt0-st;

%Apparent angle pressure involute
invlaphat0=tan(alphat0*pi/180)-alphat0*pi/180;

%Theta involute
invtheta=invlaphat0+(a/(2*r*cos(betab*pi/180)*cos(alphat0*pi/180)))-et/(2*r);

% Newton-Raphson method (to calculate theta)
invtheta1=invtheta; theta0=1;
nmax=1000; ep=1E-10;
for i=1:nmax
    [f, df]=w(theta0, invtheta1);
    theta=theta0-f/df;
    if theta==0 them
        er=abs(theta-theta0);
    else
        er=abs((theta-theta0)/theta);
    end
    if er < ep
        break
    end
    theta0=theta;
end

theta=theta*180/pi;
fprintf('theta: %6.5f \n', theta)
fprintf('Iterations number: %d \n', i)

%Dimension to the ball center
l=r*cos(alphat0*pi/180)/cos(theta*pi/180);

%Dimension under ball
l2=l-a/2;

```



%Others dimensions

$l3 = 2 \cdot l + a;$

$l4 = 2 \cdot l \cdot \cos(\pi / (2 \cdot z)) + a;$

$l5 = 2 \cdot l \cdot (\cos(\pi / (2 \cdot z)))^2 + a;$

%Comment the values that you do not want and uncomment the values that you want

`fprintf('Dimension under ball: %6.5f\n', l2)`

`% fprintf('Dimension over 2 balls (z pair): %6.5f\n', l3)`

`fprintf('Dimension over 2 balls (z odd): %6.5f\n', l4)`

`fprintf('Dimension over 3 balls (z odd): %6.5f\n', l5)`

The following M-file code defines the expressions to apply in the Newton-Raphson method.

%Involute function to apply in the Newton-Raphson method

`function [f, df]=w(theta, invtheta1);`

`f=tan(theta)-theta-invtheta1;`

`df=(tan(teta))^2;`

D.2. Approximate expression to calculate dimension under ball

Over thickness = overth

pressure angle = α

Dimension under ball $_{initial} = DUB_i$

Dimension under ball $_{final} = DUB_f$

$$DUB_f = DUB_i + \frac{\text{overth}}{\sin\left(\frac{\alpha \cdot \pi}{180}\right)}$$



Appendix E



E.1. Dimensional control

Measures obtained after each operation in the post control of the line production and in the KLINGELNBERG machine.

Gears G9T

	Hobbing
	After heat treatment
	After hard turning
	After Grinding

Table 1 - Dimensions measured in the production line.

	HP4 (Treated gear)				HP5 ⁷			
RUB [mm]	39,691	39,648	39,645	39,584	39,814	39,811	39,811	39,583
Fr [μ m]	3	112,000	36	25	9	76	19	17
TR [mm]	36,613	36,629	-	-	36,766	36,763	-	-
Fr [μ m]	0,012	0,112	-	-	0,014	0,072	-	-
RR [mm]	42,02	42,018	-	-	42,17	42,159	-	-
Fr [μ m]	0,022	0,190	-	-	0,017	0,101	-	-

Table 2 - HP4 – ADI gear (G9T z49 left): profile control.

Profile								
Position	Left				Right			
	38	26	14	1	1	14	26	38
fH α	1,5	5,6	4	0,6	-8,8	-7,2	-6,7	-9,2
fH α m	2,8		w 4,5		-8		w 2,5	
fH α	39,6	15,3	-8,6	-9,4	13,5	-11,8	12,9	34,6
fH α m	9,2		w49		12,3		w46,4	
fH α	21,6	10,9	18,8	10,3	25,2	18,8	10,3	20,2
fH α m	15,4		w11,3		18,6		w14,9	
fH α	15,9	1,1	9,1	22	1,9	4,6	3,7	1,7
fH α m	12				3			
F α	99,6	93	96,3	99,5	108	105	106	110
F α	75,6	90	130	121	113	132	85,4	67,6
F α	42,2	36,6	40	35,6	45,8	34,7	38	60,2
F α	36,2	29	20,8	43,4	6,2	10,3	5,7	4,8
ff α	100	95	97,6	99,6	103	101	101	104
ff α	89	98,1	127	119	119	125	90,7	79,2
ff α	16,8	23,3	19,6	26,9	18,7	22	26,8	40,6

⁷ Only the metal (steel) sleeve was treated.

ff α	21,5	19,4	9	22,2	2,7	4	3,6	3,1
-------------	------	------	---	------	-----	---	-----	-----

Table 3 - HP4 (G9T z49 left): helix control.

Position	Helix				Right			
	Left				Right			
	38	26	14	1	1	14	26	38
fH β	10,3	9,5	11,8	7,1	-9,5	-8,5	-6,3	-7,4
fH β m	9,7		w 4,7		-7,9		w 3,2	
fH β	-2,6	-7,1	1,1	4,5	-1,6	8,1	1,8	-13,1
fH β m	-1		w 11,6		-1,2		w 21,2	
fH β	-4,7	-5,5	-10	0,8	-2,2	-0,9	3,3	-11
fH β m	-4,9		w 10,8		-2,7		w 14,3	
fH β	-1,5	-10,1	-1,3	8,4	-5,1	-8,8	-14,3	-10,5
fH β m	-1,1				-9,7			
F β	18	15,8	20,8	14,6	20	17,4	22,6	28,3
F β	17,8	27,2	18,6	17,1	23,1	25,4	23,3	18,1
F β	32	39,1	20,2	29,2	11,4	21,1	28,5	22,6
F β	14,3	22	7,3	18,2	8,5	10,2	14,4	11,7
ff β	11,3	9,9	13,7	10,1	21,4	19	24	30,4
ff β	16,5	29,5	18,4	18,2	24,2	23,4	23,6	15,2
ff β	39,5	44,7	28,5	30,8	11,4	21,4	26,3	14,9
ff β	13,5	18,2	7,4	16,5	1,4	1,1	1,7	2,1

Table 4 - HP4 (G9T z49 left): others dimensions controlled.

Fr	14,4	139	78,2	35,6	[μ m]
Tip d.	84,044	84,089	84,082	84,076	[mm]
Root d.	73,221	73,256	73,256	73,246	[mm]

	Left	Right	Left	Right	Left	Right
Fp [μ m]	10,2	10,7	112	130	29,2	19,8

Table 5 - HP5 (G9T z49 left): profile control.

Position	Profile				Right			
	Left				Right			
	38	26	14	1	1	14	26	38
fH α	11,5	16,4	19,2	12	7,5	11,8	7,8	5,2
fH α m	14,8		w 7,7		8,1		w 6,6	
fH α	4,2	3,5	31	30,2	18,6	28,2	-2,9	-9,6
fH α m	17,2		w 27,5		8,6		w 37,8	
fH α	11,7	11,3	14,5	10,7	4,6	5,4	1,8	2,2
fH α m	12,1		w 3,8		3,5		w 3,6	
fH α	0,8	4,4	2,2	-1,3	0,2	1,5	5,7	4
fH α m	1,5				2,9			

F α	27,9	30,7	34,9	28,1	29,7	36,4	34	30,2
F α	31,1	33,3	45,5	43,4	39,1	48,7	40	44,8
F α	25,2	24,4	29,5	24,7	25,3	26,2	23,1	24,2
F α	4,7	5,8	5,2	6,5	4,7	4,5	7	5,6
ff α	29	30,5	30,1	29,7	32,1	33,3	32,3	34,3
ff α	33,7	36	27,8	25,7	30,2	34,3	37,7	37,2
ff α	14,5	14,9	16,6	15,4	18,9	18,2	17,5	18,8
ff α	2,2	2,8	2,6	2,7	2,9	2,3	3,8	3,6

Table 6 - HP5 (G9T z49 left): helix control.

Helix								
	Left				Right			
Position	38	26	14	1b	1b	14	26	38
fH β	10	9,6	6,6	10	-6,5	-7,4	9,8	-8,2
fH β m	9,1		w 3,4		-8		w 3,3	
fH β	15,8	18,3	2,3	6,4	-12,2	-14,6	-6,7	-7,8
fH β m	10,7		w 16		-10,3		w 7,9	
fH β	7,6	9,4	7	7,9	-11,1	-8,5	-9,3	-14,3
fH β m	8		w 2,4		-10,8		w 5,8	
fH β	2,1	5,1	7,8	3,9	-11,7	-7,6	-7	-12,9
fH β m	4,7				-9,8			
F β	17,3	17,9	9,2	17,7	11,5	14,7	15,8	15,9
F β	21,9	28,4	11	21	15,1	19,3	15,3	14,7
F β	14,6	11,3	7,7	10,1	12,3	10,5	10,9	14,5
F β	5,2	6,1	7,9	6,6	10,3	8,7	8,2	11,3
ff β	12,1	13,3	6,4	12,5	12,6	15	15,3	17,4
ff β	13,2	18,4	10,9	18,8	11,4	16,7	16,4	15,9
ff β	11,6	5,3	5,5	5,2	3,2	4,2	5,9	5,2
ff β	1,8	1,6	2,1	1,8	2,2	2,1	1,6	1,7

Table 7 - HP5 (G9T z49 left): others dimensions controlled.

Fr	13,2	92,7	15,2	17,4	[μ m]
Tip d.	84,351	84,356	84,362	84,349	[mm]
Root d.	73,521	73,527	73,532	73,527	[mm]

	Left	Right	Left	Right	Left	Right	Left	Right
Fp [μ m]	14,4	13,4	89,4	94,4	12,5	10,9	15,3	18,4



FINAL PROJECT

Experimental Comparison of the Mass Balance System
performance With Carbonitrided Steel Gears and
Austempered Ductile Iron Gears



Table 8 – HP6 and HP7 (G9T z41 right): dimensions measured in the production line.

	HP6				HP7			
RUB	29,592	29,579	29,578	29,36	29,536	29,524	29,514	29,362
Fr	18	105,000	47	14	13	74	67	10
TR	31,518				31,444			
RR	27,876				27,811			

Table 9 - HP6 (G9T z41 right): profile control.

	Profile							
	Left				Right			
Position	32	22	12	1	1	12	22	32
fH α	1	2,3	-3	-0,2	-1,6	-1,2	-2,9	-0,7
fH α m	0		w 5,3		-1,6		w 2,2	
fH α	2,6	2,2	3,5	10,7	4,6	6,8	-1,2	1,6
fH α m	4,8				3			
fH α	-1,6	-0,8	-0,8	-1,2	14,9	-0,5	1,5	1,7
fH α m	-1,1				4,4			
F α	5,8	4,9	5,1	4,3	3,9	3	3,9	6
F α	19,7	7	9,8	15,1	15,4	15,3	14,5	9,9
F α	5,1	5,6	4,6	5,1	28	4,3	4,2	6,3
ff α	5,6	3,2	5,8	4,5	4,3	2,7	4,4	6,7
ff α	18	6,6	7,3	10,5	11,9	10,9	14,7	9,5
ff α	2,1	2,8	2,4	2,3	10,8	2,8	3,2	2,1

Table 10 - HP6 (G9T z41 right): helix control.

	Helix							
	Left				Right			
Position	32	22	12	1	1	12	22	32
fH β	4,3	2,3	5,8	7,6	-4,6	-4,6	-7,9	-7,6
fH β m	5		w 5,3		-6,2		w 3,3	
fH β	-22,4	-8,8	-1,5	16,1	9,7	-9,2	-19,1	-21,1
fH β m	-4,2				-9,9			
fH β	-2,3	2,8	4,9	1,5	-0,4	3	3,5	-1,4
fH β m	1,7				1,2			
F β	4,5	2,9	12,4	6,7	11,6	9,9	9,6	14,1
F β	38,5	11,2	4,9	18,6	18,3	9,8	24,3	24,6
F β	5,9	6,3	6,4	5	3,7	6,1	6,7	4,7
ff β	2	2,3	8,1	2,9	13,5	11	8,7	14,9
ff β	32,3	7,4	4,4	11,9	13	5,4	13,5	14,3
ff β	1	2,2	0,9	1,1	1,9	0,5	1	0,8



Table 11 - HP6 (G9T z41 right): control of the others dimensions.

Fr	20	102	7,8	[μm]
Tip d.	63,05	63,065		[mm]
Root d.	55,73	55,753		[mm]

	Left	Right	Left	Right	Left	Right
Fp [μm]	22,6	22,4	27,9	66,9	6	11,4

Table 12 - HP7 (G9T z41 right): profile control.

Profile								
	Left				Right			
Position	32	22	12	1	1	12	22	32
fH α	-0,8	-5,5	-4,7	-1,3	-6	-5	-2,3	0,3
fH α m	-3,1		w 4,7		-3,3		w 6,3	
fH α	9	-4,7	2,2	1,8	-1,5	5,5	6,4	8
fH α m	2,1				4,6			
fH α	-1,1	-0,9	-1,5	-2	1,9	0,3	1,1	-0,3
fH α m	-1,4				0,8			
F α	7,4	12,1	9,1	6,2	11,5	10,7	8,6	8,1
F α	13	18,2	8,1	5,9	10,4	13,8	14,3	13,6
F α	4,4	4	4,3	4,8	3,8	4	11,4	5
ff α	7,1	9	5,2	5,6	8,3	7	7,6	8,2
ff α	12,3	16,4	7,2	6,3	9,6	9,5	16,4	9,2
ff α	2,2	1,8	1,8	1,7	2,6	2,4	7,1	2,5

Table 13 - HP7 (G9T z41 right): helix control.

Helix								
	Left				Right			
Position	32	22	12	1	1	12	22	32
fH β	4,9	1,3	2,3	7,3	1,1	0,7	-4,5	-2,5
fH β m	4		w 6		-1,3		w 5,6	
fH β	-0,8	-0,8	-6,6	8,4	2,9	13,4	-17,9	-22,4
fH β m	-1,8				-6			
fH β	11,3	-0,2	-13,5	-2,2	7,6	-8,3	-1,4	14,4
fH β m	-1,2				3,1			
F β	5,5	5,1	4,7	6,5	20,6	7,6	9,4	36
F β	7,9	14,7	18,6	10,9	6,9	22,3	21,5	21,4
F β	9,8	4	11,2	4,2	8,9	9,9	5,6	12,4
ff β	4	5	4,2	4,4	20,1	8	9,6	37
ff β	7,6	11,1	16,3	7,7	6	13,6	13,1	11,6



FINAL PROJECT

Experimental Comparison of the Mass Balance System
performance With Carbonitrided Steel Gears and
Austempered Ductile Iron Gears



Universidade do Porto
Faculdade de Engenharia
FEUP



CACIA



RENAULT



ff β	0,8	0,7	0,8	1	1,2	0,9	0,6	0,7
------------	-----	-----	-----	---	-----	-----	-----	-----

Table 14 – HP7 (G9T z41 right): control of the others dimensions.

Fr	11,4	114	4,8	[μ m]
Tip d.	62,894	62,924		[mm]
Root d.	55,6	55,617		[mm]

	Left	Right	Left	Right	Left	Right
Fp [μ m]	12,9	11	64,7	40,6	13,8	17,2

Table 15 - HP8 and HP9 (G9T z41 left): control in the production line.

	HP8				HP9			
RUB	29,604	29,603	29,598	29,366	29,604	29,592	29,599	29,363
Variation		0,00%	-0,02%	-0,78%		-0,041%	0,024%	-0,797%
Fr	22	147,000	48	10	15	146	70	10
TR	31,513				31,516			
RR	27,86				27,861			

Table 16 - HP8 (G9T z41 left): profile control.

	Profile							
	Left				Right			
Position	32	22	12	1	1	12	22	32
fH α	-0,5	-4,4	-3,7	-0,9	-1,1	-4,4	-2,3	-0,2
fH α m	-2,4		w 3,9		-2		w 4,2	
fH α	9,6	22,2	3,9	-11,2	-3,9	1,9	11,6	24,6
fH α m	6,1				8,6			
fH α	-1,7	-3,1	-1,4	-0,2	-2,4	-2,8	-2,6	-0,4
fH α m	-1,6				-2,1			
F α	5,1	5,3	4,7	6,2	3,9	6,9	4,1	4,4
F α	15,1	24,7	10,9	15	7,6	11,3	18,9	27,1
F α	5,8	6,1	4,9	4,9	4	5,6	5,8	12,7
ff α	5,2	5	3,6	6,4	3,5	4	4,2	4,4
ff α	6,9	8,2	10	9,6	5,1	11,8	9,5	9,7
ff α	2,8	2,4	2,6	3,1	2,1	2,3	1,5	10,1

Table 17 - HP8 (G9T z41 left): helix control.

	Helix							
	Left				Right			
Position	32	22	12	1	1	12	22	32



FINAL PROJECT

Experimental Comparison of the Mass Balance System
performance With Carbonitrided Steel Gears and
Austempered Ductile Iron Gears



fHβ	-11,2	-5,6	-3,4	-10,5	-11,2	-3,7	-3,5	-8
fHβm	-7,7		w 7,8		-6,6		w 7,7	
fHβ	58,8	10,7	-94,2	-19,7	-1,8	-88,4	-21,7	57,9
fHβm	-11,1				-13,5			
fHβ	-11	-1,7	1	-6,6	-4,2	5,7	5,3	-3,7
fHβm	-4,6				0,8			
Fβ	20,2	18,6	19,2	20,1	23,8	22,9	17,4	26,6
Fβ	49,3	14,4	78,2	18,5	5,9	85,1	26,6	48
Fβ	11	6,7	5,6	7,8	4,3	7,7	5	7,7
ffβ	14,5	14,6	17,4	15,7	18,2	20,1	16	20
ffβ	10,3	10,2	14,4	5	5,3	26,6	13,1	8,9
ffβ	1,3	1,7	2,5	1,8	1,7	3	2	7,1

Table 18 - HP8 (G9T z41 left): control of the others dimensions.

Fr	19,7	137	13,1	[μm]
Tip diam.	63,053	63,076		[mm]
Root diam.	55,736	55,749		[mm]

	Left	Right	Left	Right	Left	Right
Fp [μm]	24,7	25,2	161	96,9	14	13,5

Table 19 - HP9 (G9T z41 left): profile control.

	Profile							
	Left				Right			
Position	32	22	12	1	1	12	22	32
fHα	-1,6	-1,5	-2	-5,1	-0,3	-1,9	0,4	-1,5
fHαm	-2,6		w 3,6		-0,8		w 2,3	
fHα	8	-4,9	2,2	10	12,1	0,7	7,6	0
fHαm	3,8				5,1			
fHα	-2,8	-1,1	-1,9	-3,7	-4,4	-2,8	-2,7	-1,4
fHαm	-2,4				-2,8			
Fα	3,8	3,6	3,6	7,7	2,6	3,8	4,7	3,8
Fα	11,4	9,5	6,5	15,3	18,3	7,2	17,6	12,4
	6,1	4,9	4,8	6,4	6,4	6,8	5,9	7,1
ffα	4,2	4	3,3	7,6	2,6	4,4	4,5	3,6
ffα	8,4	8,1	5,3	8	8,9	7	13,5	12,4
ffα	3,1	2,5	2,4	2,8	2,5	2,9	2,7	4,3

Table 20 - HP9 (G9T z41 left): helix control.

Helix	
Left	Right



FINAL PROJECT

Experimental Comparison of the Mass Balance System
performance With Carbonitrided Steel Gears and
Austempered Ductile Iron Gears



Universidade do Porto
Faculdade de Engenharia
FEUP



CACIA



RENAULT



Position	32	22	12	1	1	12	22	32
fH β	-11,4	-13,1	-6,3	-3,4	-3,4	0,7	-8,4	-12,7
fH β m	-8,6		w 9,7		-6		w 13,4	
fH β	-52,3	6,6	20,5	-37,7	-52,9	-7,8	21,6	27,7
fH β m	-15,7				-16,7			
fH β	-3,7	-5,5	-3,1	-3,2	0,5	1,5	0,3	-0,3
fH β m	-3,9				0,5			
F β	19,7	18	16,2	18,5	21,4	24	28,8	24,5
F β	54,8	11,7	27,4	32	43,3	27,2	19,5	25,7
	7,1	8,4	7,1	6,5	2,9	3,7	1,7	2,4
ff β	21,2	14,9	12,6	16,5	21,8	24,3	21,8	15,8
ff β	16,8	9,6	13,8	7,7	6	26,8	9	7
ff β	1,5	2,3	1,8	2,3	1,4	2,4	1,5	2

Table 21 - HP9 (G9T z41 left): control of the others dimensions.

Fr	20,8	145	11,2	[μ m]
Tip diam.	63,05	63,069		[mm]
Root diam.	55,738	55,755		[mm]

	Left	Right	Left	Right	Left	Right
Fp [μ m]	23,5	20,1	149	99,6	12,2	14,7

Table 24 - HP3 (M1D z49 left): helix control.

Gears M1D

	After Hobbing
	After Heat Treatment
	After Grinding and hard turning

Table 22 - HP3 (M1D z49 left): control in the production line.

	RUB[mm]	Fr	Tip Radius	Fr	Root Radius	Fr
Hobbing dimension*	37,36	-	-	-	-	-
After Hobbing	37,362	21	39,376	31	34,667	25
After Heat Treatment	37,324	72	39,375	49	34,673	30
After Grinding	37,108	6	39,375	49	34,673	30

*Value calculated with the MatLab



FINAL PROJECT

Experimental Comparison of the Mass Balance System
performance With Carbonitrided Steel Gears and
Austempered Ductile Iron Gears



Table 23 - HP3 (M1D z49 left): profile control.

Profile								
	Left				Right			
Position	38	26	14	1	1	14	26	38
fH α	-8,4	-6,6	-8,4	-8,3	8,5	14,5	12,4	11,7
fH α m	-7,9				11,8			
fH α	-0,2	0,5	2,3	2,2	37,1	31,3	21	28,6
fH α m	1,2				29,5			
fH α	-1,5	-0,2	0,7	-1	-2,1	-0,6	-1,5	-2,8
fH α m	-0,5				-1,8			
F α	10	12,2	12,4	12,4	23,6	24,3	26,3	23,6
F α	19,5	24,7	24,2	18,3	52,5	48,5	36,5	45,1
F α	3,1	3	3,9	6,4	3,8	2,4	4	4,8
ff α	11,8	13,4	14,8	9,4	19,5	15,4	20	18
ff α	19,6	24,4	23,1	17,1	20,4	28,6	25,2	24,3
ff α	1,8	1,8	2,3	4	2,1	1,6	2	1,7

Table 24 - HP3 (M1D z49 left): helix control.

Helix								
	Left				Right			
Position	38	26	14	1	1	14	26	38
fH β	-1,6	-5,5	-6,8	-0,3	-1,5	-1	-1,7	-0,8
fH β m	-3,6				-1,3			
fH β	-8,6	-8,5	-19,8	-19,2	22,7	-10,9	-10,6	-11,9
fH β m	-14				-2,7			
fH β	1,8	1,6	1,6	2	-2,3	-2,4	-1,7	-1
fH β m	1,8				-1,9			
F β	23,4	39,8	28,4	16	8,9	7,8	14,9	8,1
F β	13,2	17,7	18,5	21,7	21,1	15	16,6	15,3
F β	5,8	5,8	5	5,6	5,1	5,1	4,8	5,1
ff β	24	41,6	30,3	16,6	9,5	8,5	14,5	8,5
ff β	11,8	12,8	11,1	13,8	19,5	11,3	10,5	11,7
ff β	1,9	2,5	1,3	2	1,5	1,6	1,8	1,7

Table 25 - HP3 (M1D z49 left): control of the others dimensions.



Fr	13,3	88,1	7,2	[mm]
----	------	------	-----	------

	Left	Right	Left	Right	Left	Right
Fp [μm]	14,3	10,9	40,5	52,4	7,3	6,1

Table 26 - HP2 (M1D z47 left): control in the production line.

	RUB[mm]	Fr	Tip Radius	Fr	Root Radius	Fr
Hobbing dimension*	37,124	-	-	-	-	-
After Hobbing	37,13	18	38,962	36	34,154	25
After Heat Treatment	37,107	106	38,978	56	34,185	43
After Grinding	36,887	8	38,978	56	34,185	43

*Value calculated with the MatLab

Table 27 - HP2 (M1D z47 left): profile control.

	Profile							
	Left				Right			
Position	36	24	13	1	1	13	24	36
fH α	-10,8	-8	-8,5	-11,9	13	11,4	12,8	14
fH α m	-9,8				12,8			
fH α	-0,4	5	5,6	-11,9	10,3	26,8	25,6	27
fH α m	-0,4				22,4			
fH α	-6,3	-5,7	-6,5	-5,8	-0,6	-0,2	-3,5	-2,5
fH α m	-6,1				-1,7			
F α	14,4	11,8	13,6	17,2	23,1	24,9	24,3	26,6
F α	23,6	20,7	23,9	23	35,9	42,7	45,6	44,9
F α	6,9	12,6	7,3	6,7	7,2	6,8	9,7	9,2
ff α	12,2	13	14,8	14,9	14,8	17,3	15,9	15,6
ff α	23,9	19,8	20,4	21,7	30,9	24,5	28,9	22,3
ff α	1,8	10	1,1	1,7	1,1	1,1	1,2	1,2

Table 28 - HP2 (M1D z47 left): helix control.

	Helix							
	Left				Right			
Position	36	24	13	1	1	13	24	36
fH β	1,7	-1,1	0,7	0,4	5,6	4	9,4	9,6
fH β m	0,4				7,2			
fH β	-10,5	-15,8	-3	-2,4	9,4	4,3	1	-4,5
fH β m	-7,9				2,6			
fH β	-6,5	-5,6	-4,5	-7,2	0,7	3,2	4,7	2,7
fH β m	-6				2,8			





FINAL PROJECT

Experimental Comparison of the Mass Balance System
performance With Carbonitrided Steel Gears and
Austempered Ductile Iron Gears



F β	5,9	9,9	8,1	4,4	9,3	9,6	9,8	12,4
F β	17,6	20,8	9	15,9	15,9	10	12,5	16,5
F β	10,8	10,5	8,8	11,8	7	5,8	5	5,6
ff β	5,4	9,8	8,3	4,5	9	11,1	7,8	9,8
ff β	12,8	14	8,6	16,3	11,9	11,3	12,8	15,1
ff β	1,6	1,6	1,8	1,8	3,9	2,2	1,7	2,1

Table 29 - HP2 (M1D z47 left): control of the others dimensions.

Fr	19,4	121	18,6	[mm]		
	Left	Right	Left	Right	Left	Right
Fp [μm]	10,2	14,3	50,7	68,5	15,8	15,2

Table 30 - HP1 (M1D z47 right): control in the production line.

	RUB[mm]	Fr	Tip Radius	Fr	Root Radius	Fr
Hobbing dimension*	37,124	-	-	-	-	-
After Hobbing	37,126	14	39,001	21	34,206	36
After Heat Treatment	37,108	75	39,009	36	34,227	31
After Grinding	36,874	12	39,009	36	34,227	31

*Value calculated with the MatLab

Table 31 - HP1 (M1D z47 right): profile control.

Profile								
	Left				Right			
Position	36	24	13	1	1	13	24	36
fH α	-13,9	-9,6	-3,7	-10,2	-2,6	3,4	-2,9	-5
fH α m	-9,4				-1,8			
fH α	0	9,7	13,2	-14,3	2,7	7,8	13,5	1,7
fH α m	2,2				6,4			
fH α	-5,3	-7,9	-4,5	-4	-6,1	-7,3	-8,5	-6,1
fH α m	-5,4				-7			
F α	14,8	11,6	10,6	15,5	19,3	18,6	21,4	20,7
F α	18,7	24,8	27,3	25,7	30,8	30,8	30,8	23,3
F α	5,4	8,8	5,7	5,6	7,1	7,7	8,8	7,2
ff α	14,1	13,9	11,1	16,9	20,3	16,8	22,3	21,8
ff α	18,7	18,1	14,8	21,8	29,1	26,6	34,9	22,7
ff α	2,9	1,6	1,8	2,8	2,2	3	3,4	4,3

Table 32 - HP1 (M1D z47 right): helix control.



Helix								
	Left				Right			
Position	36	24	13	1	1	13	24	36
fH β	-13,8	-10,7	-7,7	-12,6	-12,4	-11	-15,6	-20,4
fH β m	-11,2				-14,9			
fH β	-14,7	-24,7	-24,1	-7,1	-13,4	-15,2	-31,5	-30,4
fH β m	-17,7				-22,6			
fH β	-0,4	-3,5	-2,1	-1,3	-5,2	-6,1	-8,9	-5,1
fH β m	-1,8				-6,3			
F β	12,8	10,3	8,6	12,6	14,6	10,9	12,8	15
F β	16	25,8	24,7	15,8	15,7	14,8	28	24,5
F β	2,3	3,4	4,7	4,2	5,6	5,3	6,4	5,5
ff β	7,6	9,5	5,9	8	16,1	8,4	10,3	10,5
ff β	9	8,2	13	13,8	14,9	9	16,9	9,9
ff β	1,8	1,5	1,5	2,6	1,6	1,6	3,3	2,1

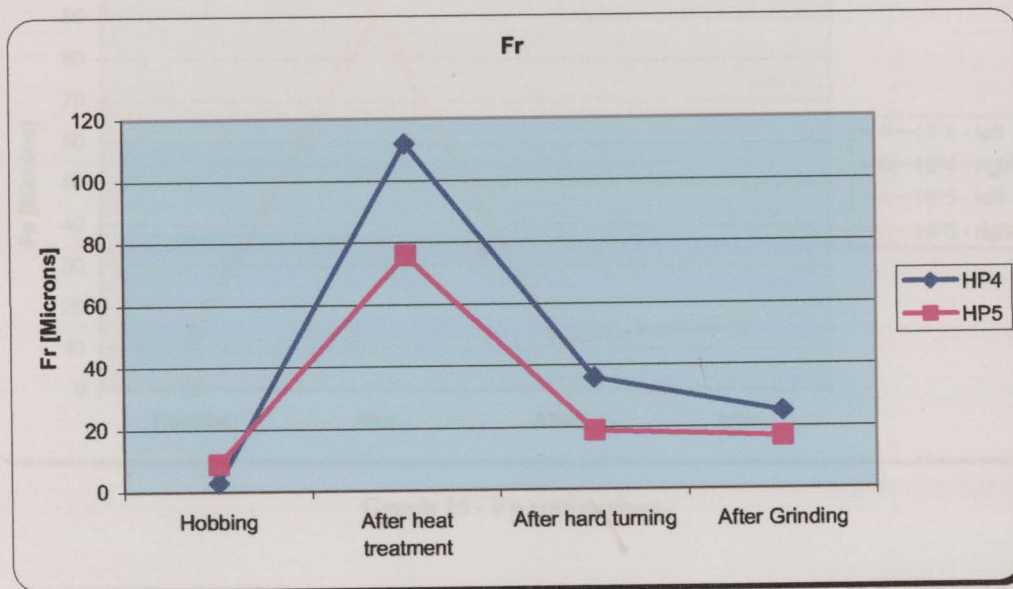
Table 33 - HP1 (M1D z47 right): control of the others dimensions.

Fr	22,9	94,8	12,9	[mm]
----	------	------	------	------

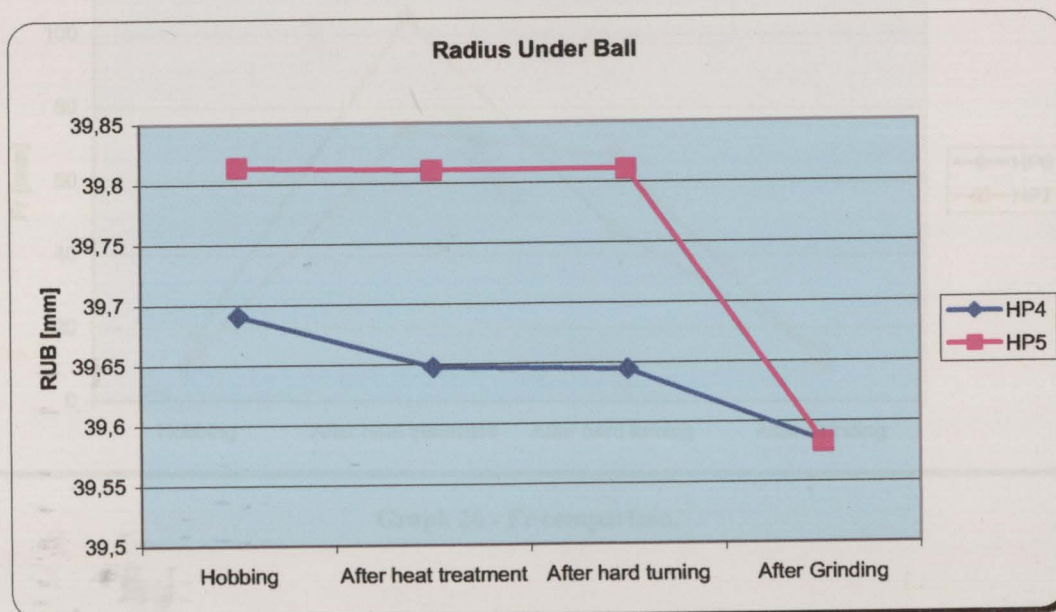
	Left	Right	Left	Right	Left	Right
Fp[μ m]	10,5	14,9	68	42,9	7,1	10,5



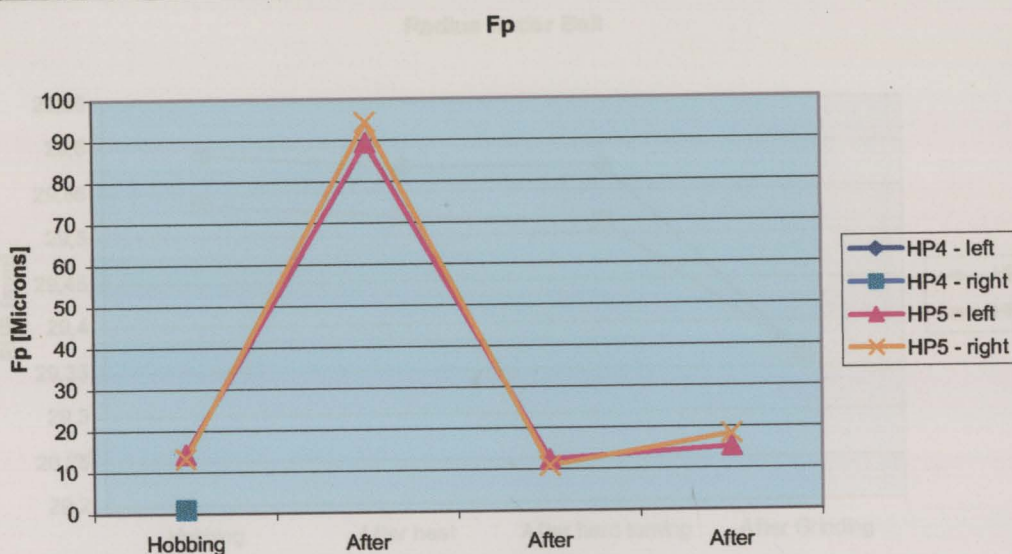
E.2. Dimensional control (graphics) of the G9T.gears



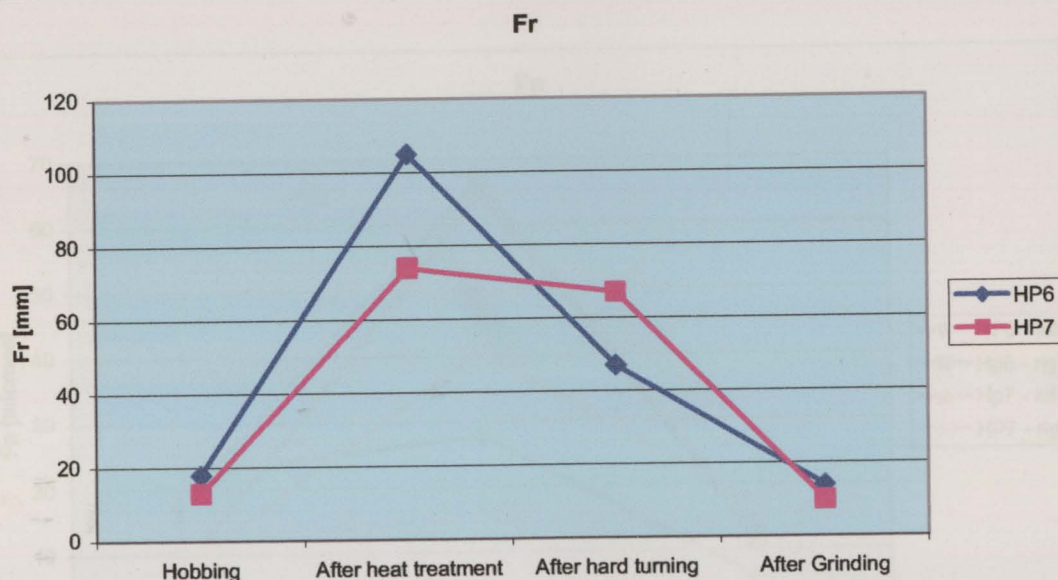
Graph 23 - Fr comparison.



Graph 24 - Radius under ball comparison.



Graph 25 - Fp comparison.



Graph 26 - Fr comparison.



FINAL PROJECT

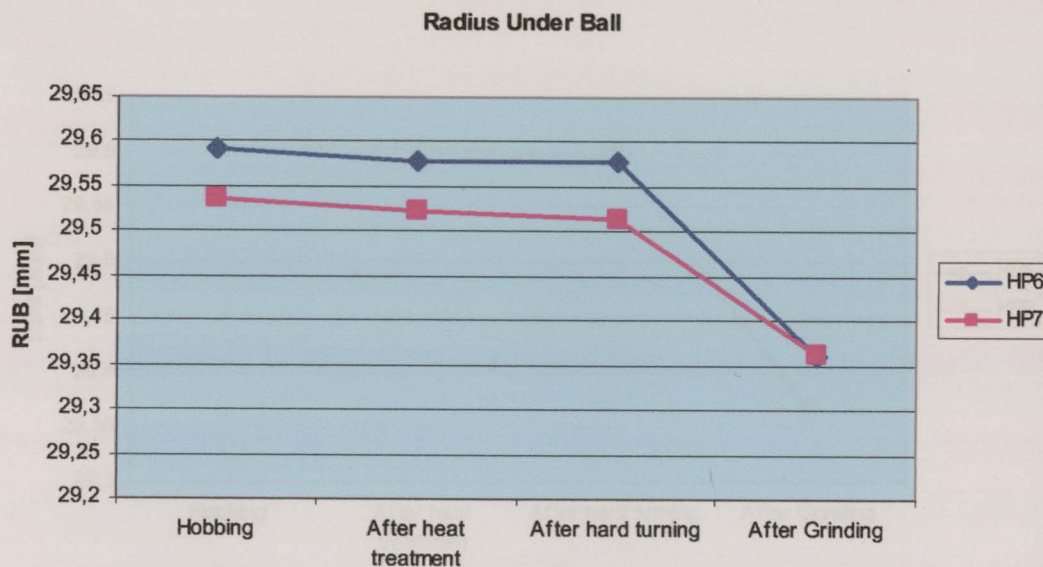
Experimental Comparison of the Mass Balance System
performance With Carbonitrided Steel Gears and
Austempered Ductile Iron Gears



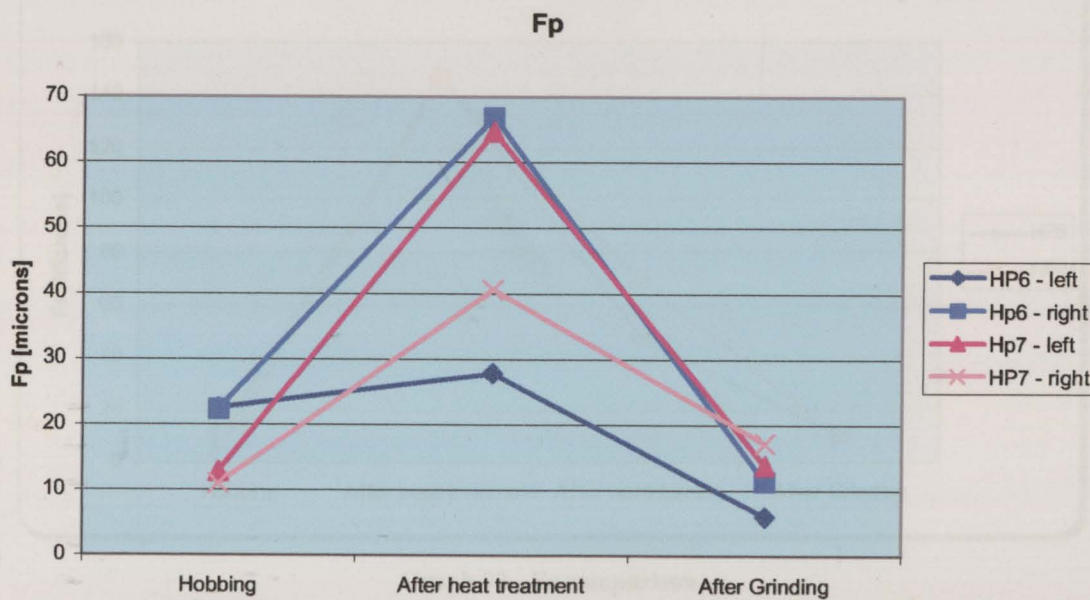
Universidade do Porto
Faculdade de Engenharia
FEUP



RENAULT



Graph 27 - Radius under ball comparison.



Graph 28 - Fp comparison.



FINAL PROJECT

Experimental Comparison of the Mass Balance System
performance With Carbonitrided Steel Gears and
Austempered Ductile Iron Gears



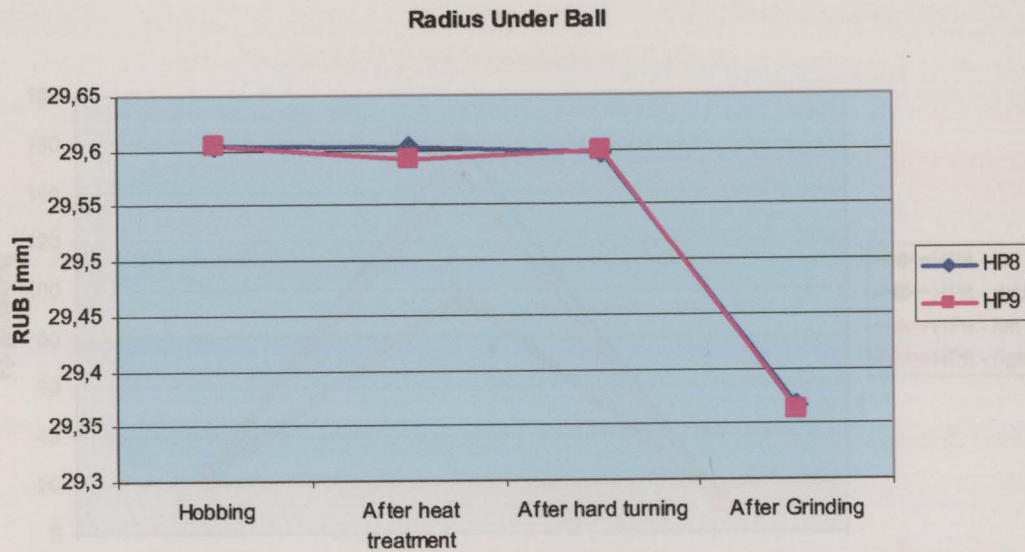
Universidade do Porto
Faculdade de Engenharia
FEUP



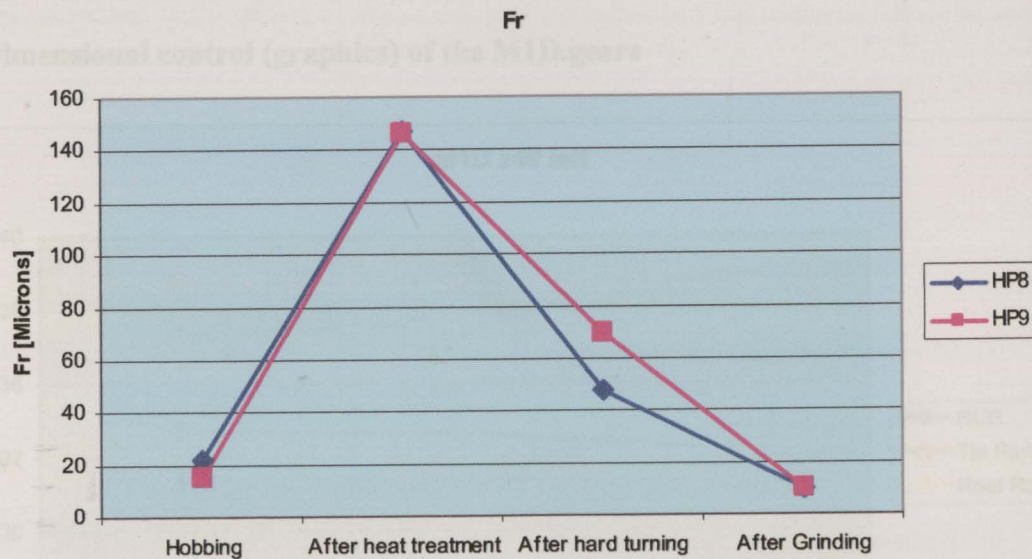
GACIA



RENAULT



Graph 29 - Radius under ball comparison.



Graph 30 - Fr comparison.



FINAL PROJECT

Experimental Comparison of the Mass Balance System
performance With Carbonitrided Steel Gears and
Austempered Ductile Iron Gears



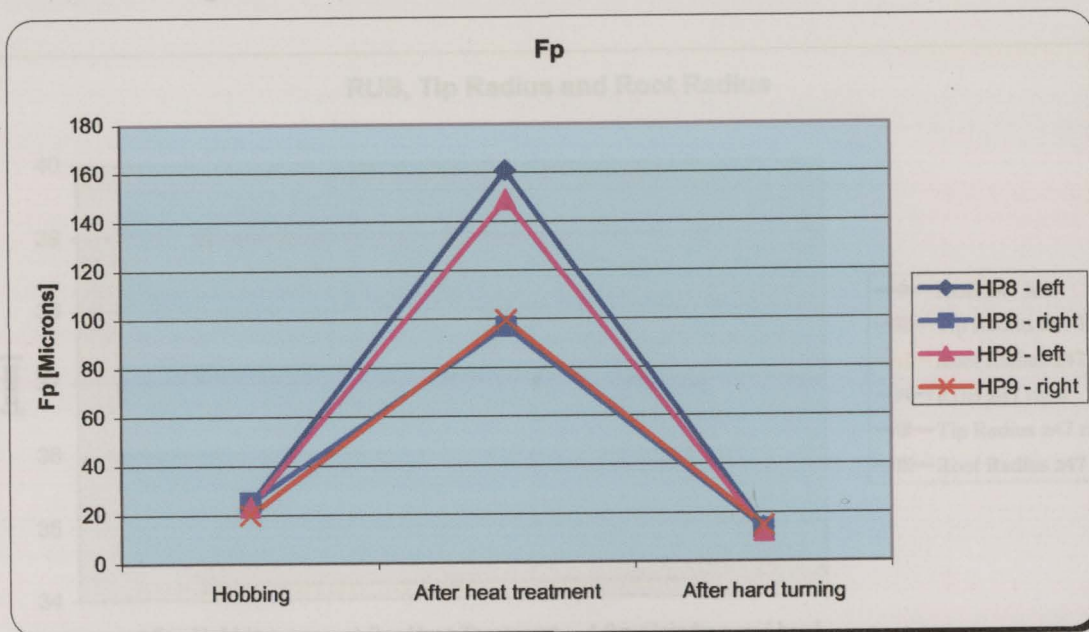
Universidade do Porto
Faculdade de Engenharia
FEUP



GACIA

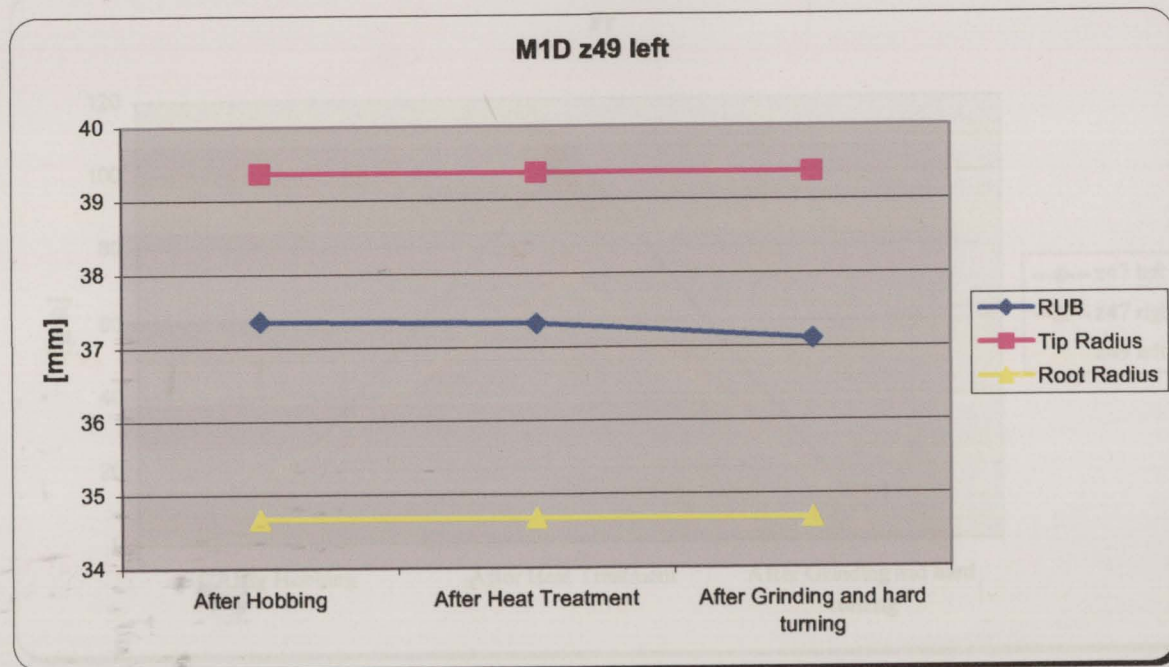


RENAULT



Graph 31 - Fp comparison.

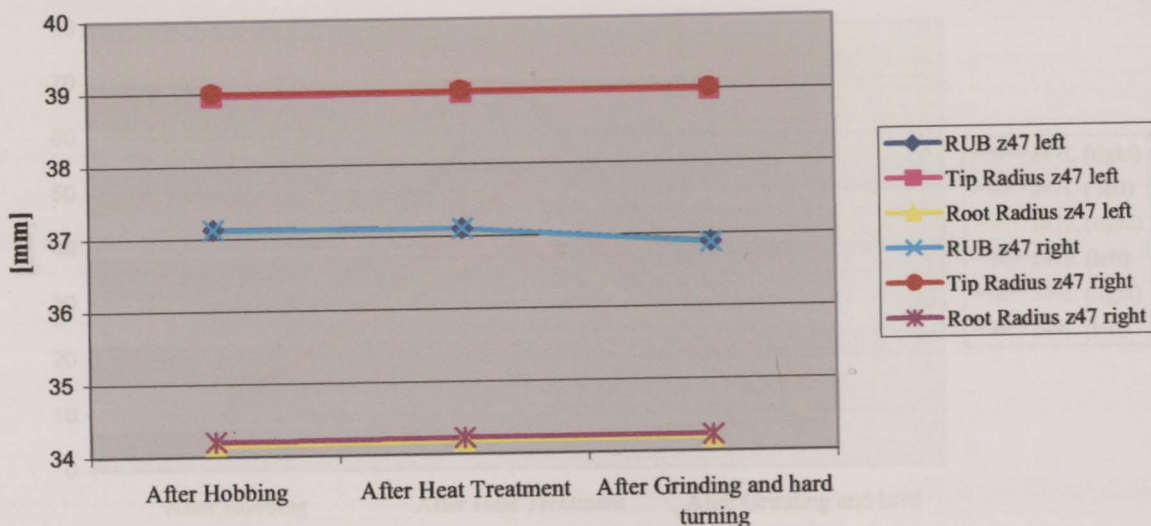
E.2. Dimensional control (graphics) of the M1D.gears



Graph 32 - Comparison of the radius under ball, tip radius and root radius of the M1D.gears.

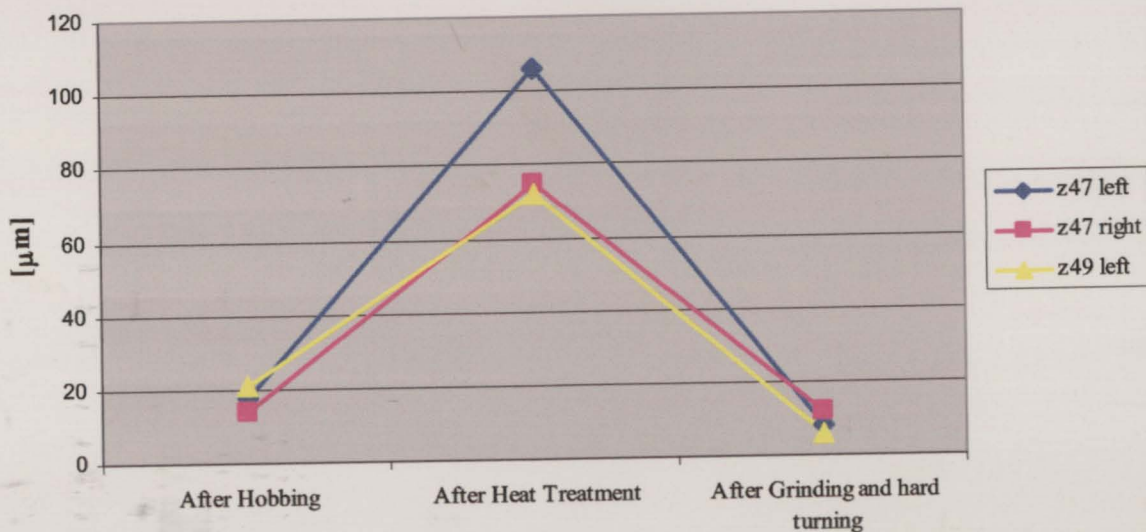


RUB, Tip Radius and Root Radius

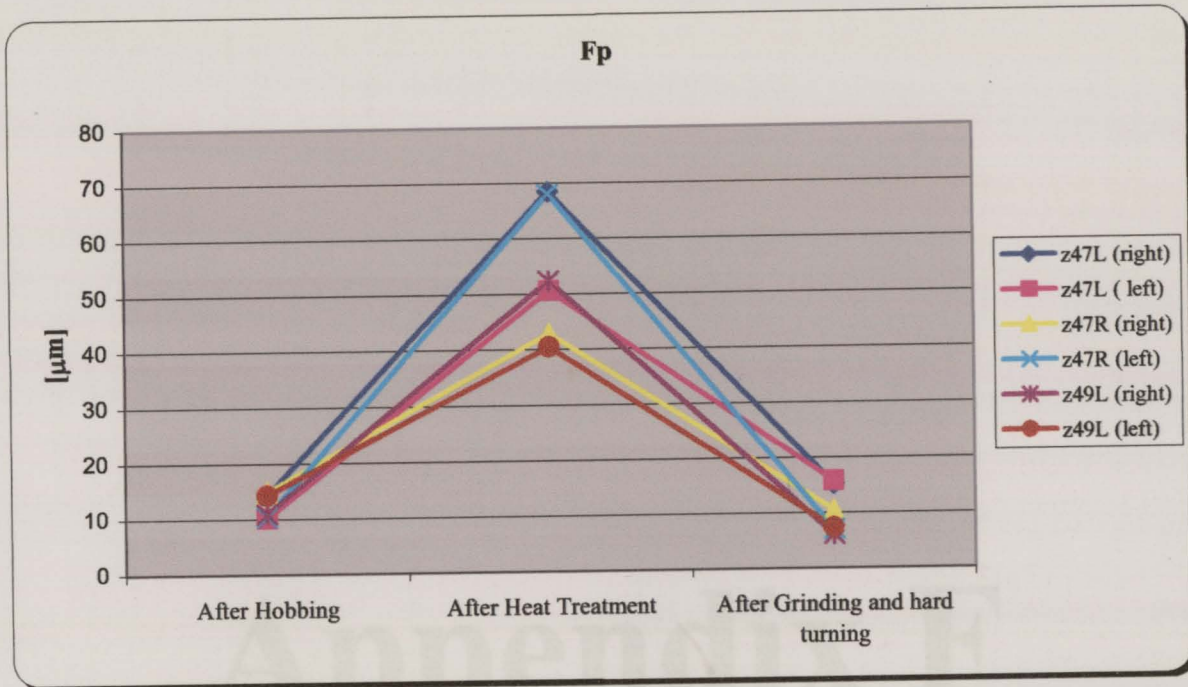


Graph 33 - Comparison of the radius under ball, tip radius and root radius of the M1D gears.

Fr



Graph 34 - Fr comparison of the M1D gears.



Graph 35 - Fp comparison of the M1D gears.



FINAL PROJECT

Experimental Comparison of the Mass Balance System
performance With Carbonitrided Steel Gears and
Austempered Ductile Iron Gears



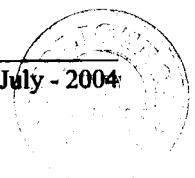
Universidade do Porto
Faculdade de Engenharia
FEUP



RENAULT



Appendix F



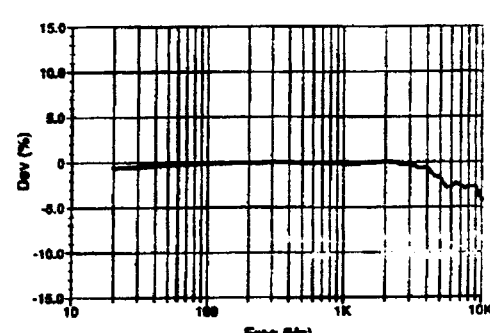


KISTLER

Kistler Instrument Corporation

 75 John Glenn Drive
 Amherst, NY 14228-2171
 Phone 716-691-5188
 Fax 716-691-5226
KALIBRIERSCHEIN
BESCHLEUNIGUNGSSENSORCALIBRATION CERTIFICATE
ACCELEROMETER

Typ	Type	8710A50M5
Serien Nr	Serial No	C147089
Bereich	Range	g ± 50
Max. Bereich	Max. Range	g ± 300
Betriebstemperaturbereich	Temperature Range, operating	°C -54 to 165
Ausgangsimpedanz	Output Impedance	Ω ≤ 100
Zeitkonstante (τ)	Time Constant (τ)	s 1,1
Ausgangsruhespannung	Bias Voltage	V DC 11,8
Resonanzfrequenz montiert (in)	Mounted Resonant Frequency (in)	kHz 23,8
Seitenempfindlichkeit	Transverse Sensitivity	% 0,4
Anzugsmoment	Mounting Torque	Nm $2,0 \pm 0,2$

Messungen	Measurements		Frequenzgang / Frequency Response
Frequenz / Frequency	Empfindlichkeit / Sensitivity	Abweichung / Deviation	
Hz	mV/g	%	
20	97,7	-0,6	
50	97,9	-0,4	
100	98,1	-0,2	
159	98,3	0,0	
200	98,2	-0,1	
500	98,1	-0,2	
1000	98,0	-0,3	
2000	98,3	0,0	
5000	96,5	-1,8	
7000	95,7	-2,6	
10000	94,0	-4,3	

Empfindlichkeit bei 159 Hz, 10,0 g rms	Sensitivity at 159 Hz 10,0 g rms	mV/g 98,3	$g = g_0 = 9,80665 \text{ m/s}^2$
Umgebungstemperatur	Environmental Temperature	°C 22 ± 4	
Relative Feuchte	Relative Humidity	% 30 ± 25	
Kalibriert von	Calibrated by	Mark Thomas	
Datum	Date	30.4.2001	

Wir bestätigen, dass das oben identifizierte Gerät nach den vorgeschriebenen Verfahren geprüft wurde. Alle Messmittel sind auf nationale Normale rückführbar, die Bezugsnormale auf das National Institute of Standards and Technology. Das Kalibrierlabor entspricht MIL-STD-45662A.

We confirm that the device identified above was tested by the prescribed procedures. All measuring devices are traceable to national standards, the reference standards through the National Institute of Standards and Technology per test report 822/261353. The Kistler calibration laboratory is in conformance with MIL-STD-45662A.

Referenz Geräte	Reference Equipment	Typ / Type	Seiten-Nr / Serial No
Beschleunigungs-Sensor (Gebrauchsnormale)	Accelerometer (Working Standard)	809K112	C121186
Ladungsverstärker (Gebrauchsnormale)	Charge Amplifier (Working Standard)	5020	C34168
Beschleunigungs-Sensor (Bezugsnormale)	Accelerometer (Primary Standard)	8002K	C17447
Ladungsverstärker (Bezugsnormale)	Charge Amplifier (Primary Standard)	5020	C4870

040-0035-001 - Seite 1 von 1

ISO 9001 CERTIFIED QUALITY SYSTEM



KISTLER

Kistler Instrument Corporation

 75 John Glenn Drive
 Amherst, NY 14228-2171
 Phone 716-691-5100
 Fax 716-691-5226

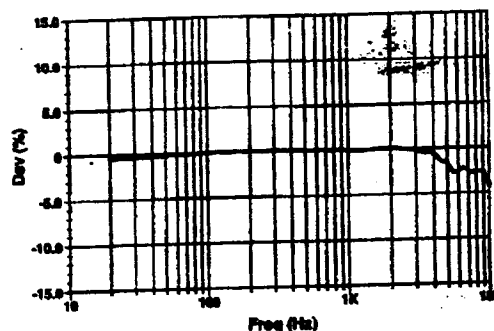
ACCELEROMETER CALIBRATION CERTIFICATE

Type	8710A50M5		
Serial Number	C147089		
Time Constant	sec.	1.1	Mounted Resonant Frequency kHz 23.8
Transverse Sensitivity	%	0.4	Mounting Torque lbf-in 18 ± 2.0
Bias Voltage	V _{cc}	11.8	Nm 2.0 ± 0.2
Reference Specifications			
Range	g	± 50	Temp. Range, operating °C -54 to 165
Max Range	g	± 300	Output Impedance Ω ≤ 100

Measurements

Frequency Hz	Sensitivity mV/g	Deviation % (ref = 100 Hz)
20	97.7	-0.4
50	97.9	-0.2
100	98.1	0.0
200	98.2	0.1
500	98.1	0.0
1000	98.0	-0.1
2000	98.3	0.2
5000	96.5	-1.6
7000	95.7	-2.5
10000	94.0	-4.1

Frequency Response



Sensitivity at 100 Hz, 10.0 g rms	mV/g	98.1	$g = 9.807 \text{ m/s}^2$	159.2 Hz = 1000 revs/sec
Sensitivity at 159 Hz, 10.0 g rms	mV/g	98.3		
Environmental Temperature	°C	22 ± 4	Condition	New
Relative Humidity	%	30 ± 25	NIST Test Report Number	822/281353
Calibration Date		4/30/2001	Calibrated By:	<i>W. J. Kistner</i>

Original Certificate signed in blue ink by Kistler Calibration Technician

This accelerometer was calibrated per Kistler Test Procedure 978-5304-701 using a back-to-back comparison technique against a Kistler Working Standard. Kistler Working Standards are periodically calibrated against a Kistler Reference Standard System which in turn is periodically recertified, traceable to the National Institute of Standards and Technology (NIST). The Kistler calibration laboratory is in conformance with ANSI-STD-45462A and ANSI/NCSL Z540-1-1994. Estimated uncertainty of this calibration is 2.8 % with respect to reference standard.

This certificate shall not be reproduced, except in full, without written approval of Kistler Instrument Corporation.

Reference Equipment	Type	Serial Number
Accelerometer (Working Standard)	809K112	C121186
Charge Amplifier (Working Standard)	5020	C34168
Accelerometer (Primary Standard)	8002K	C17447
Charge Amplifier (Primary Standard)	5020	C4870

040-0035-001 - Page 1 of 1

ISO 9001 CERTIFIED QUALITY SYSTEM





FINAL PROJECT

Experimental Comparison of the Mass Balance System
performance With Carbonitrided Steel Gears and
Austempered Ductile Iron Gears



KISTLER

Piezo-Instrumentation

K-SHEAR®-Piezotron®-Beschleunigungssensor Accéléromètre Piezotron® K-SHEAR® K-SHEAR® Piezotron® Accelerometer

Nr. 8.8710 Ed. 3.97 P. 1... 2
Type
8710A...

Universell einsetzbarer, hermetisch dichter Beschleunigungssensor mit eingebautem Impedanzwandler (Piezotron®), der trotz hoher Empfindlichkeit sehr robust ist.

Der Sensor ist masseisoliert und eignet sich speziell für Anwendungen, bei denen die Gefahr von Erdschleifenproblemen besteht.

Sonderausführung für Betrieb bis 165 °C erhältlich.

Accéléromètre d'usage universel, hermétique-ment étanche avec convertisseur d'impédance incorporé (Piezotron®). Très robuste malgré sa sensibilité très élevée.

L'accéléromètre est isolé par rapport à la masse et est tout spécialement qualifié pour les applications avec des problèmes de circuit de retour par la terre.

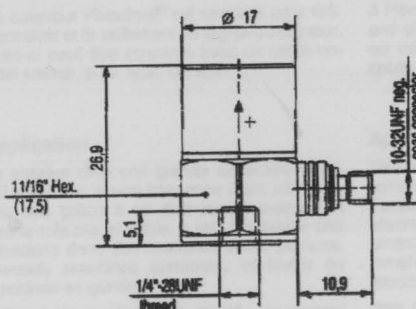
Un modèle spécial est disponible pour les températures de service atteignant 165 °C.

Universal, hermetically sealed accelerometer with built-in impedance converter (Piezotron combining high sensitivity with ruggedness.

This accelerometer is ground isolated and especially suited for applications where potential ground loop problems exist.

Special version available for operation up to 165 °C.

- Hohe Auflösung bei minimalem Rauschen
Résolution élevée à bruit minimale
High resolution at low noise
- Quarz-Sensorelemente
Éléments sensibles à quartz
Quartz sensor elements
- Masseisoliert und hermetisch dicht
Isolé par rapport à la masse et hermétiquement étanche
Ground isolated and hermetically sealed
- Minimale Seitenempfindlichkeit
Sensibilité transversale minimale
Minimal transverse sensitivity
- CE-konform
Conforme au CE
Conforming to CE



1:1

Technische Daten

Données techniques

Technical Data*

Type	Type	Type		8710A50M1	8710A50M5
Bereich	Gamme	Range	g	±50	±50
Empfindlichkeit	Sensibilité	Sensitivity	mV/g	100 (±5 %)	±100 (±10 %)
Frequenzbereich ± 5 % Grenze ±10 % Grenze	Gamme de fréquence ± 5 % limite ±10 % limite	Frequency range ± 5 % limit ±10 % limit	Hz	0.6 ... 7 k 0.4 ... 10 k	1.0 ... 7 k 0.6 ... 10 k
Betriebstemperaturbereich (bei 4 mA Speisung)	Gamme de température d'utilisation (à l'alimentation 4 mA)	Operating temperature range (at power supply 4 mA)	°C	-55 ... 120	-55 ... 165
Masse	Masse	Mass	g	43	29
Überlast Shock (1 ms Puls)	Surcharge Choc (impulsion 1 ms)	Overload Shock (1 ms impulse)	g		<2000
Ansprechschwelle	Seuil de réponse	Threshold	g _{rms}		~0.002
Resonanzfrequenz (montiert)	Fréquence de résonance (montée)	Resonant frequency (mounted)	kHz	~28	~24
Seitenempfindlichkeit	Sensibilité transversale	Transverse sensitivity	%		~1.5
Basisebnungs-empfindlichkeit bei 250 µε	Sensibilité aux contraintes de montage à 250 µε	Base strain sensitivity at 250 µε	g / µε		~0.004
Temperaturkoeffizient der Empfindlichkeit	Coefficient de température de la sensibilité	Temperature coefficient of sensitivity	% / °C		~0.03
Lagerungstemperaturbereich	Gamme de température de stockage	Storage temperature range	°C	-75 ... 150	-75 ... 180
Zeitkonstante	Constante de temps	Time constant	s	~1.5	~1.0
Speisung Konstantstrom Spannung (Kuppler)	Alimentation Courant constant Tension (coupleur)	Supply Constant current Voltage (coupler)	mA V		2 ... 18 20 ... 30

1 m = 1000 mm, 1 s = 1000 ms, 1 g = 0.1019... g, 1 inch = 25.4 mm, 1 Nm = 0.73756... lbf; 1 g = 0.03527... oz

* In all Kistler documents, the decimal sign is a comma on the line (ISO 31-0:1992).

Kistler AG, P.O. Box 131, CH-8406 Winterthur, Switzerland. Tel. +41 (0)52 28 21 11. Fax +41 (0)52 28 21 11. USA: Phone (714) 951-1000.



8.8/ TU

3.9/

<

Type	Type	Type	8710A50M1	8710A50M5
Ausgang	Sortie	Output		
Spannung (Vollbereich)	Tension (pleine échelle)	Voltage (full scale)	V	±5
Ruhe Spannung	Tension de repos	Bias voltage	V DC	±11
Impedanz	Impédance	Impedance	Ω	100
			MΩ	±10
Masseisolation	Isolation par rapport à la masse	Ground isolation		
Gehäusewerkstoff	Matériau du boîtier	Housing material	rostfreier Stahl / acier inox / stainless steel	Titan / titane / titanium
Anzugsdrehmoment	Couple de serrage	Mounting torque	Nm	2 ±0,7

Beschreibung

Auf Scherung beanspruchte Quarzelemente sind im hermetisch dichten Gehäuse eingebaut. Mit dieser Bauart werden extrem kleine Werte für thermische Transienten und Basisdehnungsempfindlichkeit erreicht.

Die von den Quarzelementen abgegebenen elektrischen Ladungen werden durch eingebaute Piezotron®-Impedanzwandler (ICP kompatibel) in niederohmige Ausgangssignale umgewandelt. Preisgünstige, handelsübliche Kabel von fast beliebiger Länge zwischen Sensor und Kuppler genügen daher.

Für die Speisung und Signalaufbereitung wird ein Piezotron®-Kuppler verwendet. Der Sensor kann mit einem normalen Koaxialkabel, z.B. Typ 1761Cx, angeschlossen werden.

Anwendung

Der Sensor bietet hohe Empfindlichkeit (hohe Auflösung) und niedere untere Grenzfrequenz dank einem robusten und zuverlässigen Quarz-Sensorelement. Er eignet sich zur Schwingungsmessung an Werkzeugmaschinen, Pumpen, Pressen, Stanzen, Fahrzeugen oder ähnlichen Einrichtungen.

Zusammen mit Kabel Typ 1939A... kann der Sensor zudem in nasser, verschmutzter Umgebung eingesetzt werden.

Dank der masseisolierten Basis werden Probleme wegen elektrischen Erdschleifen vermieden.

Montage

Beim Einsatz von Beschleunigungssensoren ist eine korrekte Montage wesentlich. Folgende Montagearten sind möglich:

- **Direkte Montage** mit Befestigungsschraube Typ 8412 (1/4" - 28) oder Typ 8421 (M8).
- **Klebe montage** mit Adapter Typ 8438.
- **Klebe montage** mit Petro-Wax Typ 8432 für provisorische Befestigung.
- **Montage mit Magnet**
Für Messungen bis 2,5 kHz können die Sensoren zusammen mit Magnet Typ 8456 eingesetzt werden.

Lieferumfang

Typ 8410, 8412 und 8421

Zubehör

Abmessungen der oben beschriebenen Montagehilfen: siehe Datenblatt 9.011.

- Anschlusskabel Typ 1761Cx
- Anschlusskabel mit integriertem Stecker Typ 1939Ax
- Adapter für Triaxial-Montage Typ 8506

Description

Des éléments de quartz sensibles au cisaillement sont encastrés dans un boîtier hermétiquement étanche. Cette construction permet d'obtenir de très faibles sensibilités aux chocs thermiques et aux contraintes de montage.

Les charges électriques engendrées par les éléments en quartz sont transformées en tensions de sortie à basse impédance par les convertisseurs d'impédance Piezotron® incorporés. Des câbles usuels d'un prix avantageux suffisent pour relier le capteur au coupleur à une distance presque illimitée.

Un coupleur Piezotron® est employé pour l'alimentation et le traitement du signal du capteur. Celui-ci peut être connecté avec un câble coaxial normal, p.ex. type 1761Cx.

Application

Le capteur offre une grande sensibilité (résolution élevée) et une fréquence limite inférieure très basse grâce à un élément de mesure en quartz robuste et fiable. Il sert à mesurer des vibrations dans des machines outils, pompes, presses, machines estampes, véhicules ou machines en général.

Utilisé avec le câble type 1939A... le capteur fonctionne aussi dans un milieu humide et pollué.

Grâce à son isolement par rapport à la masse on évite les problèmes avec des circuits de terre.

Montage

Lorsque les accéléromètres sont utilisés, un montage adéquat est d'importance. Les méthodes de montage suivantes sont possibles:

- **Montage direct** avec vis de montage type 8412 (1/4" - 28) ou type 8421 (M8).
- **Montage adhésif** avec adaptateur type 8438.
- **Montage adhésif** avec cire Petro-Wax type 8432 pour fixation provisoire.
- **Montage avec aimant**
Pour des mesures jusqu'à 2,5 kHz les capteurs peuvent être montés avec l'aimant type 8456.

Etendue de la fourniture

Typ 8410, 8412 et 8421

Accessoires

Dimensions des accessoires de montage décrits ci-dessus: voir notice technique 9.011.

- Câble de connexion type 1761Cx
- Câble de connexion avec connecteur intégré type 1939Ax
- Adaptateur pour montage triaxial type 8506

Description

Shear quartz sensing elements are built in a hermetically sealed housing. This design results in extremely low sensitivity to base strain and thermal transients.

The electrical charges yielded by the quartz elements are transformed by the built-in Piezotron® impedance converters (ICP compatible) into low impedance output voltages. Therefore, regular low-cost cables of nearly any length can be used between sensor and coupler.

A Piezotron® coupler is used for power supply and signal processing of the sensor. The sensor can be connected with a common coaxial cable, e.g. Type 1761Cx.

Application

This sensor combines high sensitivity (high resolution) and low frequency measurements thanks to a rugged and reliable quartz sensing element. It is very useful for vibration measurements on machine tools, pumps, presses, punching machines, vehicles or similar equipments.

With cable Type 1939A... the sensor can be used in wet and contaminated environments.

This unit has a ground-isolated base to eliminate problems with ground loops.

Mounting

Correct mounting is important when accelerometers are used. The following mounting methods are possible:

- **Direct mounting** with mounting stud Type 8412 (1/4" - 28) or Type 8421 (M8).
- **Adhesive mounting** with adapter Type 8438.
- **Adhesive mounting** with Petro-Wax Type 8432 for temporary mounting.
- **Mounting with magnet**
For measurements up to 2.5 kHz the sensors can be used together with magnet Type 8456.

Scope of delivery

Typ 8410, 8412 and 8421

Accessories

Dimensions of the mounting accessories described above: see data sheet 9.011.

- Connecting cable Type 1761Cx
- Connecting cable with integrated connector Type 1939Ax
- Adapter for triaxial mounting Type 8506

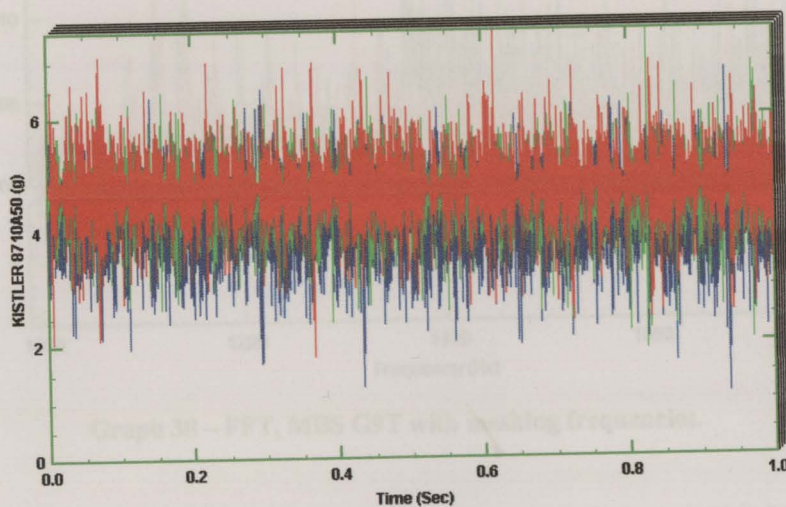


Appendix G

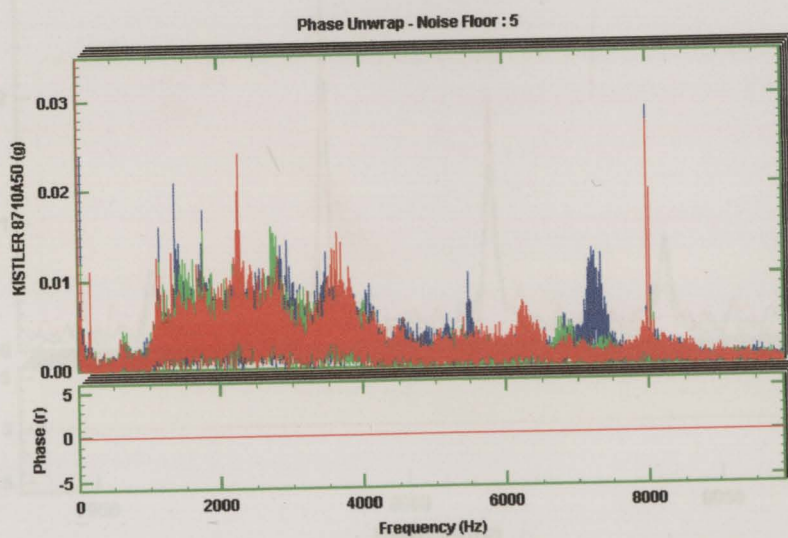


G9T

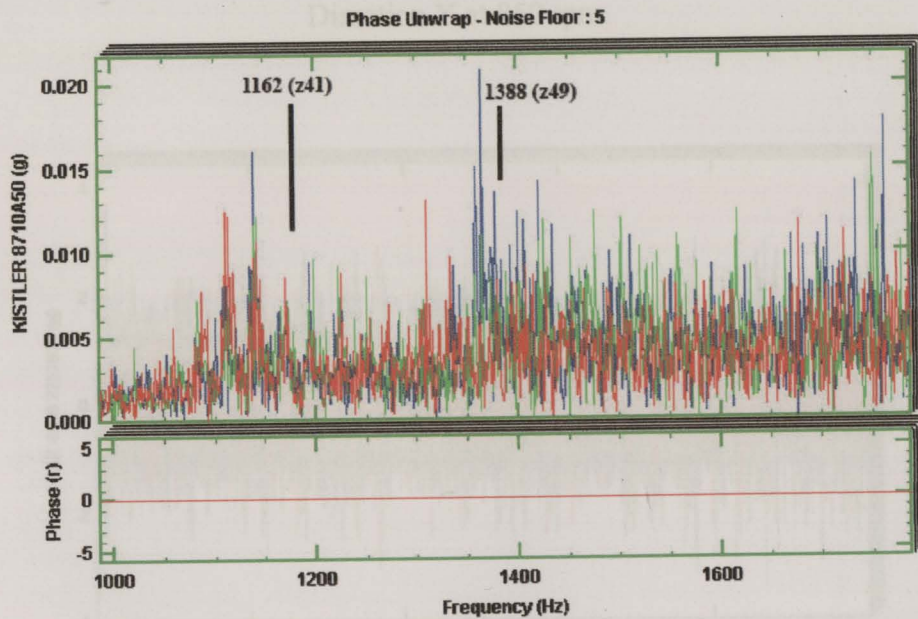
Direction X 850 rpm



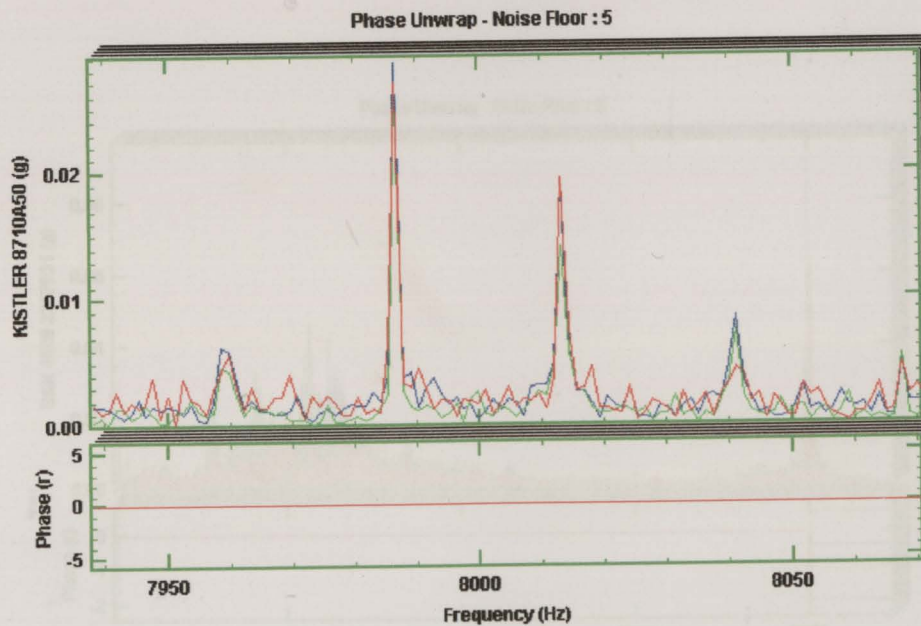
Graph 36 – MBS G9T, temporal response



Graph 37 – MBS G9T, FFT.



Graph 38 – FFT, MBS G9T with meshing frequencies.



Graph 39 – FFT MBS G9T detail.

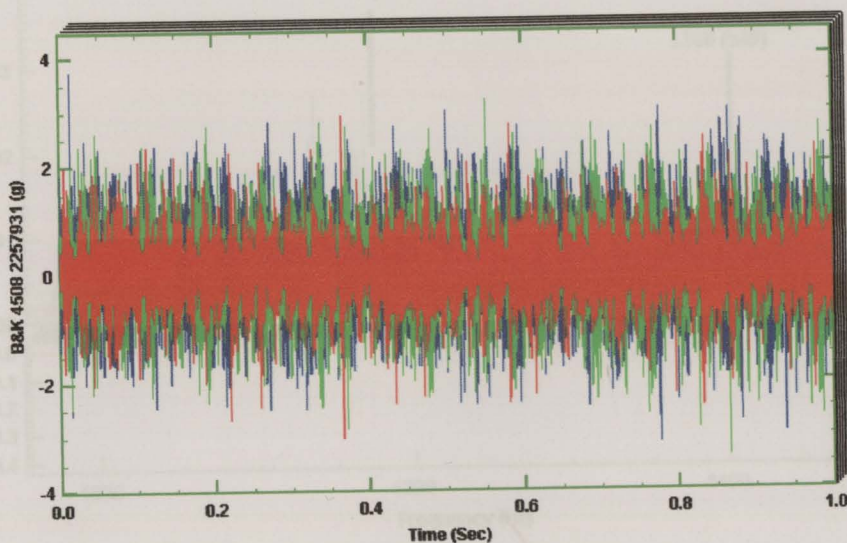


FINAL PROJECT

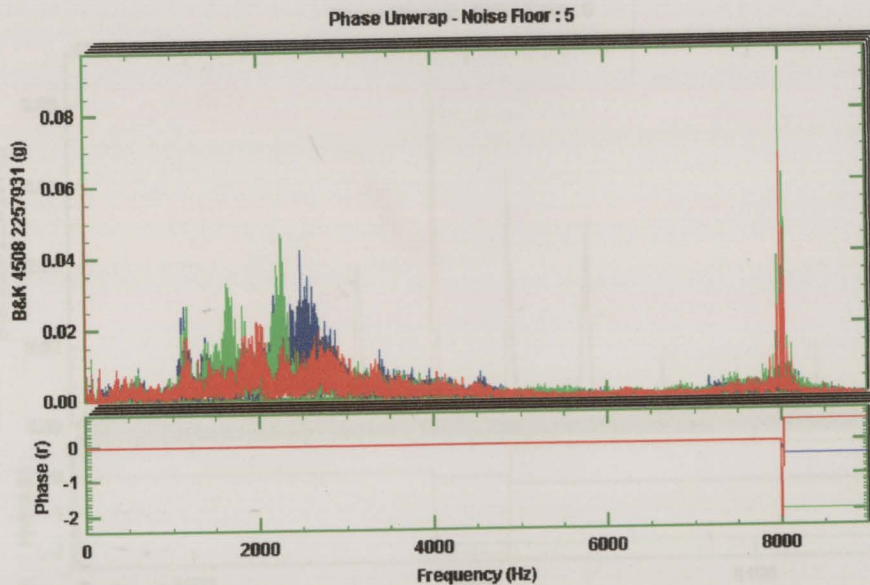
Experimental Comparison of the Mass Balance System
performance With Carbonitrided Steel Gears and
Austempered Ductile Iron Gears



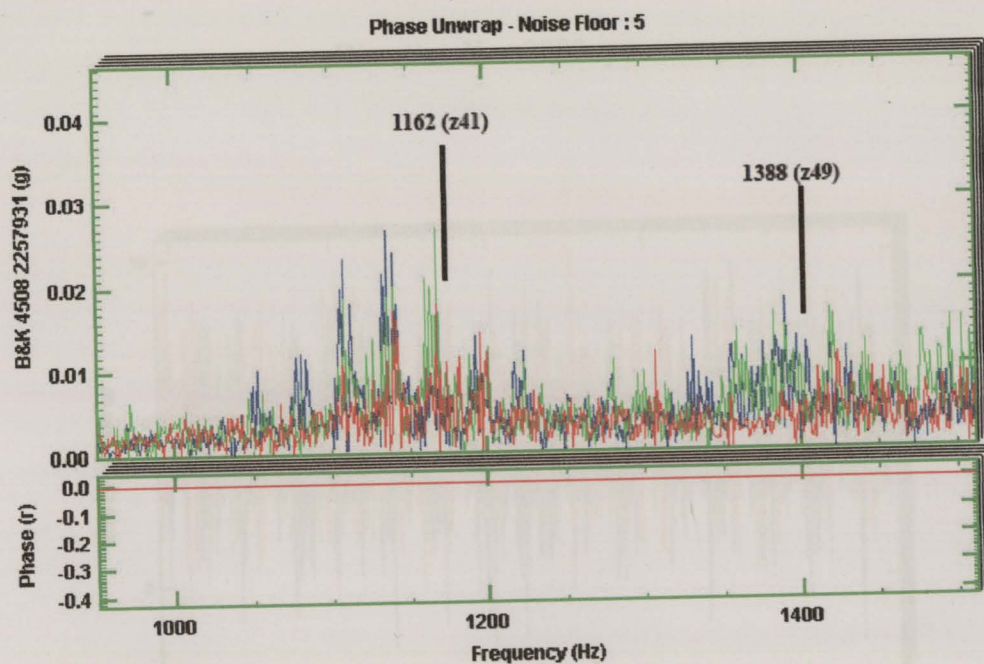
Direction Y at 850 rpm.



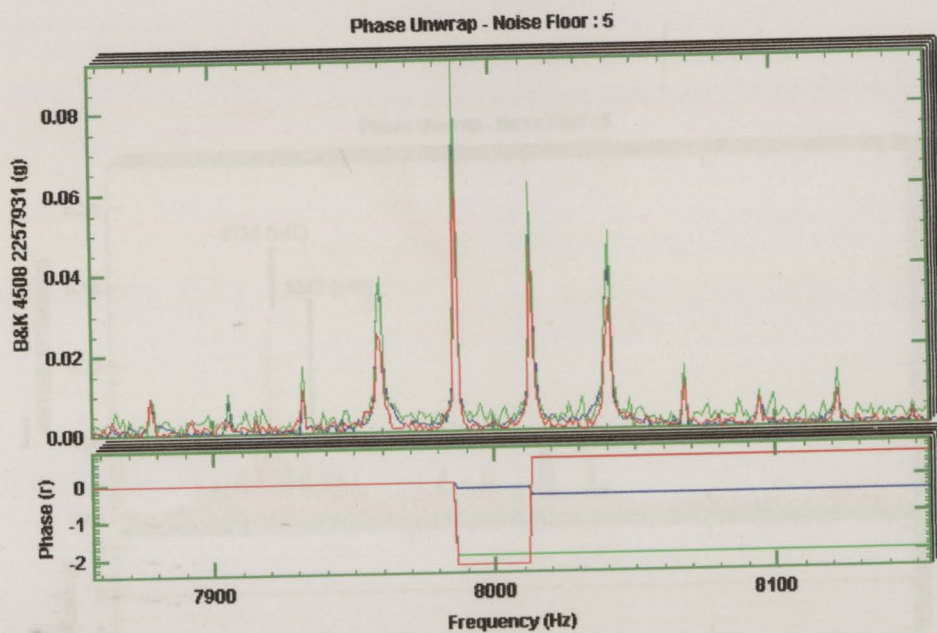
Graph 40 – MBS G9T, temporal response.



Graph 41



Graph 42 – MBS G9T, FFT with meshing frequencies.



Graph 43 – MBS G9T, FFT detail.



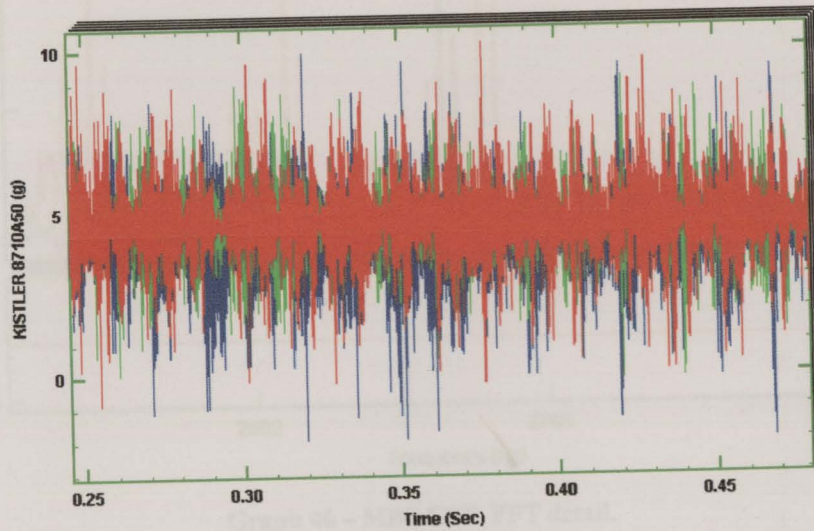


FINAL PROJECT

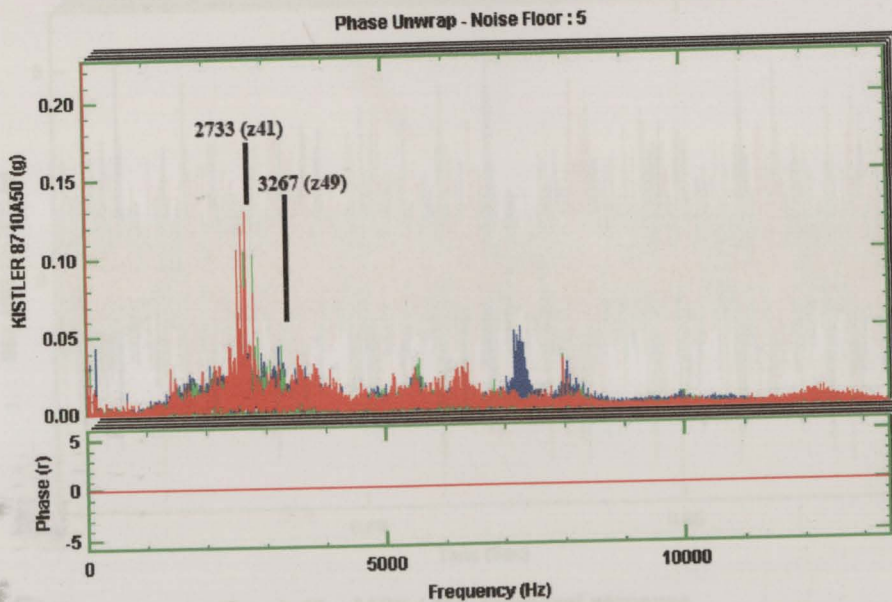
Experimental Comparison of the Mass Balance System
performance With Carbonitrided Steel Gears and
Austempered Ductile Iron Gears



Direction X at 2000 rpm.



Graph 44 – MBS G9T, temporal response.



Graph 45 – MBS G9T, FFT with meshing frequencies.



FINAL PROJECT

Experimental Comparison of the Mass Balance System
performance With Carbonitrided Steel Gears and
Austempered Ductile Iron Gears



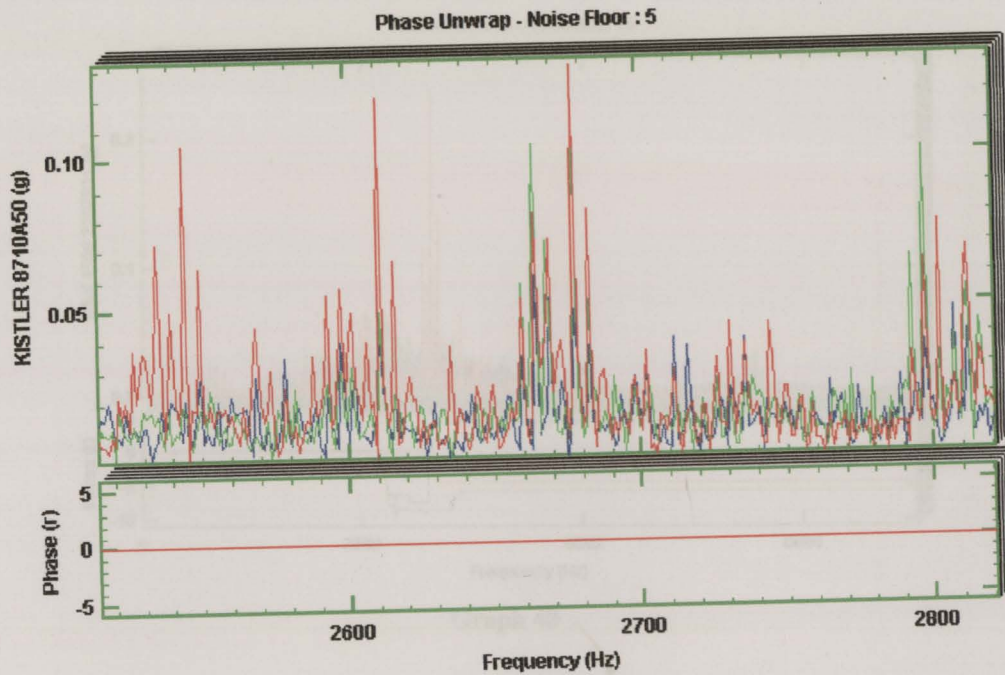
Universidade do Porto
Faculdade de Engenharia
FEUP



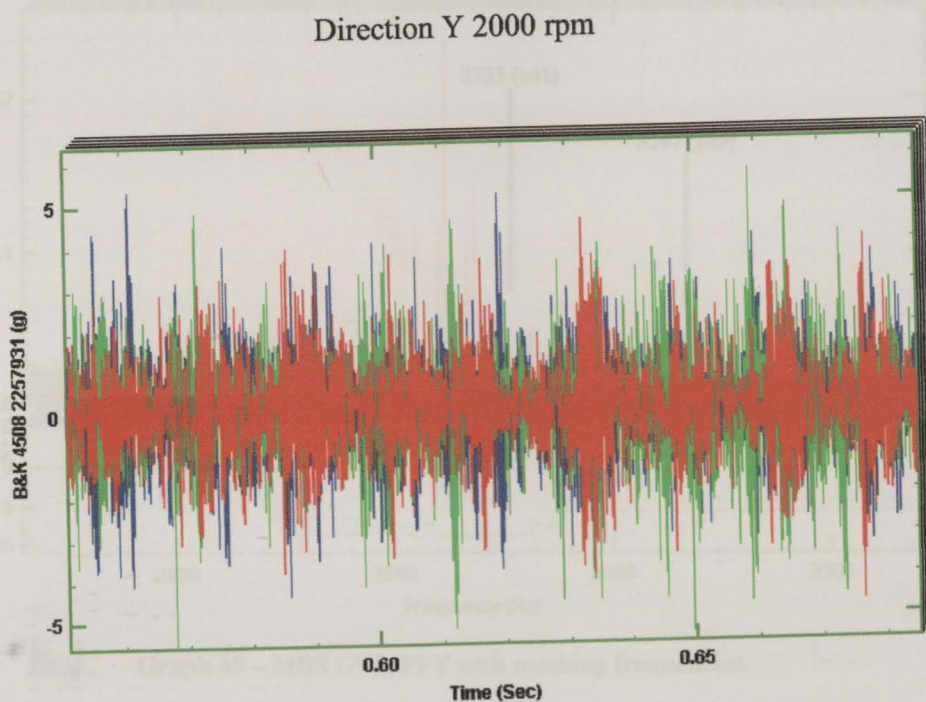
CAGIA



RENAULT



Graph 46 - MBS G9T, FFT detail.



Graph 47 - MBS G9T, temporal response.



FINAL PROJECT

Experimental Comparison of the Mass Balance System
performance With Carbonitrided Steel Gears and
Austempered Ductile Iron Gears



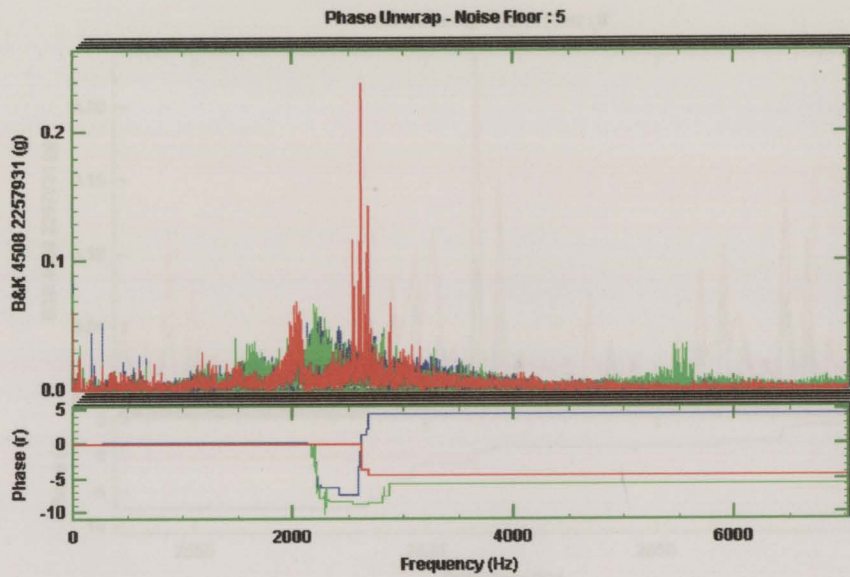
Universidade do Porto
Faculdade de Engenharia
FEUP



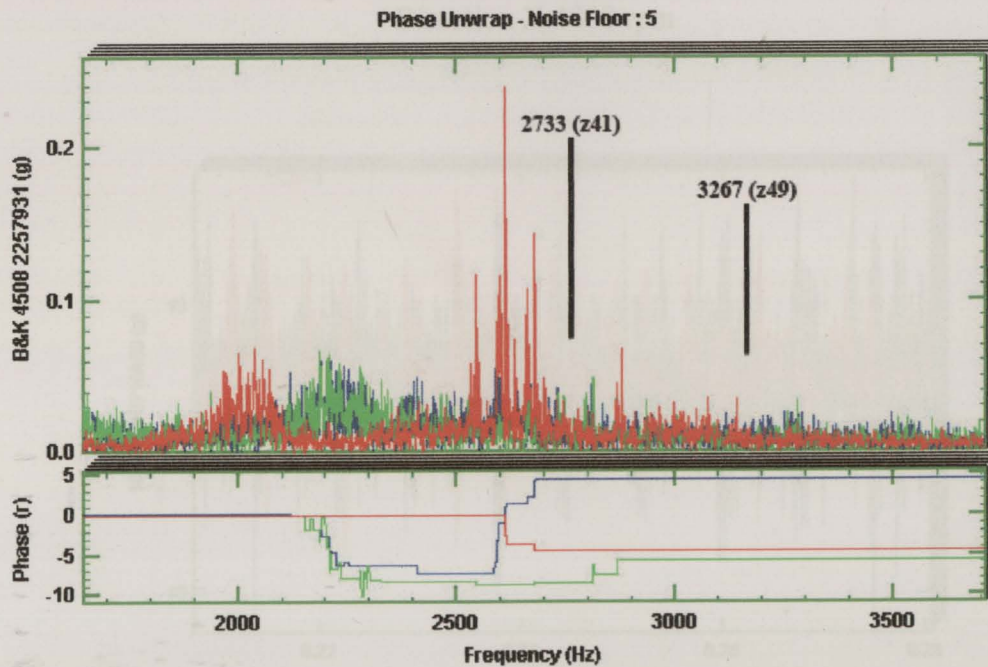
C.A.C.I.A.



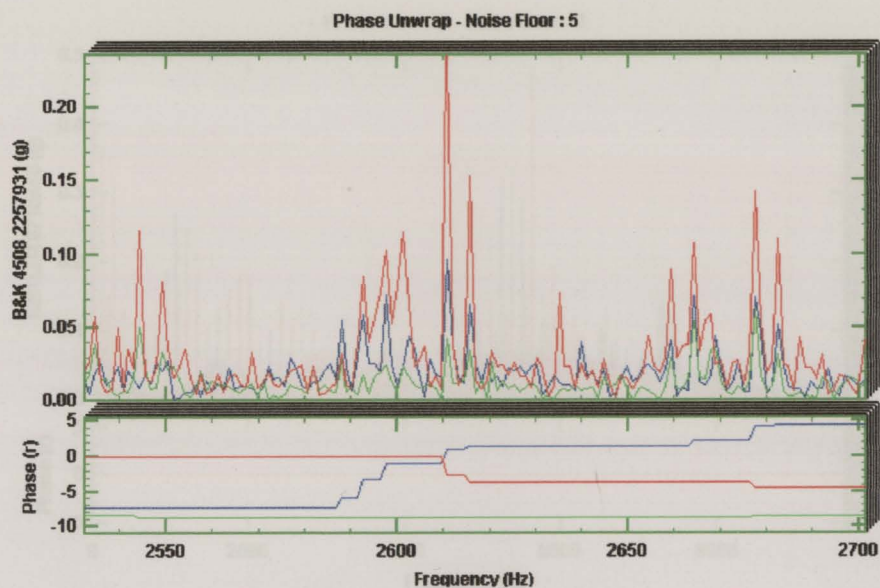
RENAULT



Graph 48

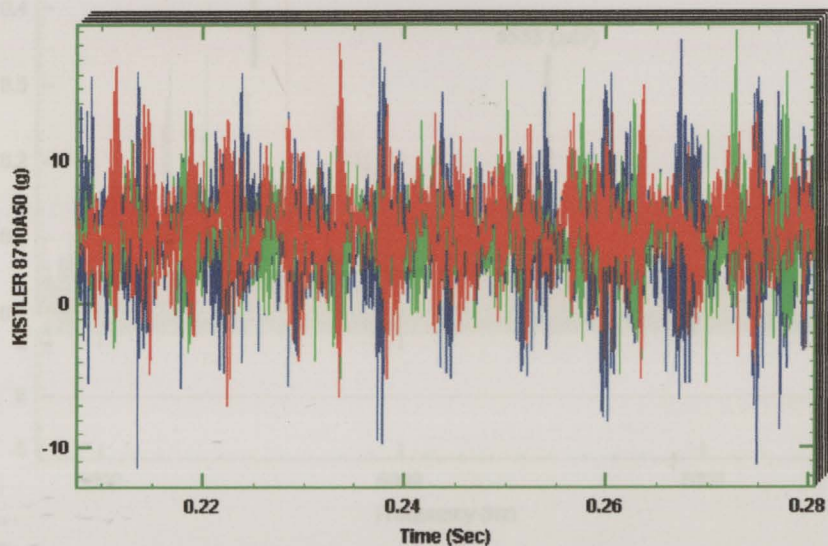


Graph 49 – MBS G9T, FFT with meshing frequencies.



Graph 50 – MBS G9T, FFT detail.

Direction X 4000 rpm

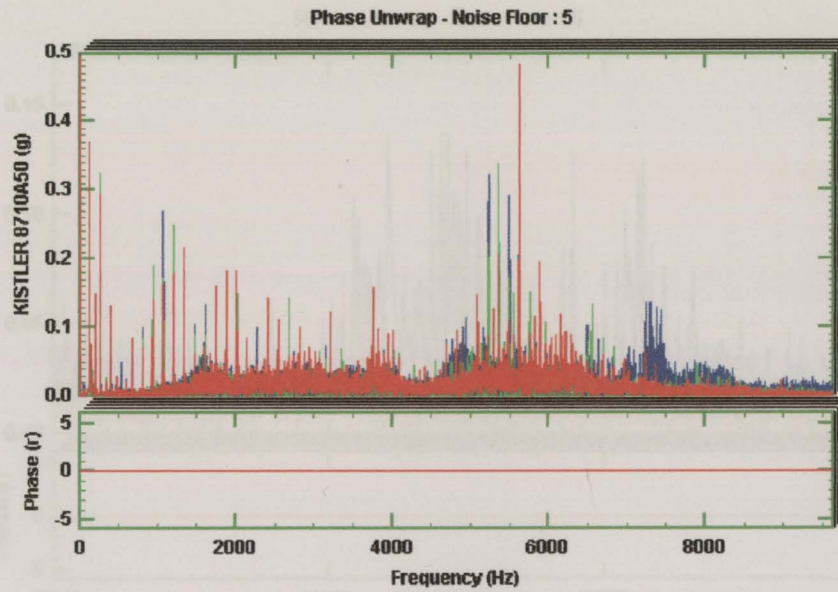


Graph 51 – MBS G9T, temporal response.

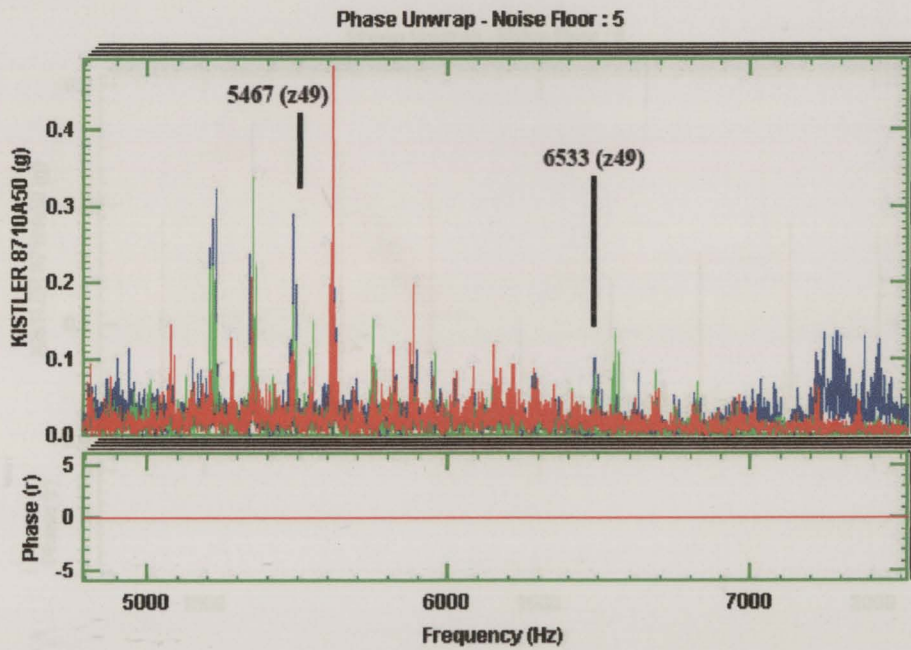


FINAL PROJECT

Experimental Comparison of the Mass Balance System
performance With Carbonitrided Steel Gears and
Austempered Ductile Iron Gears



Graph 52- MBS G9T, FFT.



Graph 53- MBS G9T, FFT with meshing frequencies.



FINAL PROJECT

Experimental Comparison of the Mass Balance System
performance With Carbonitrided Steel Gears and
Austempered Ductile Iron Gears



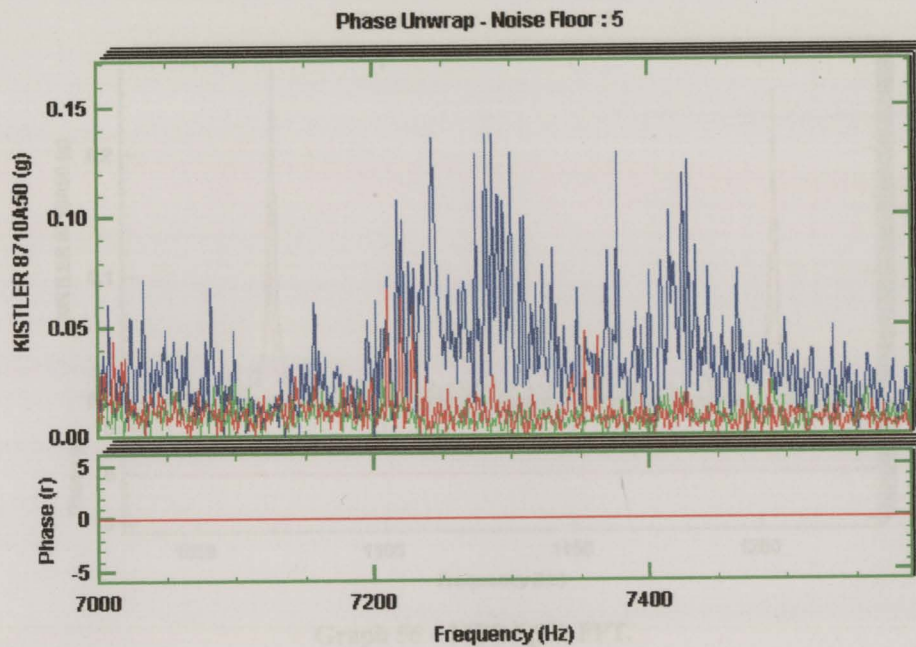
Universidade do Porto
Faculdade de Engenharia
FEUP



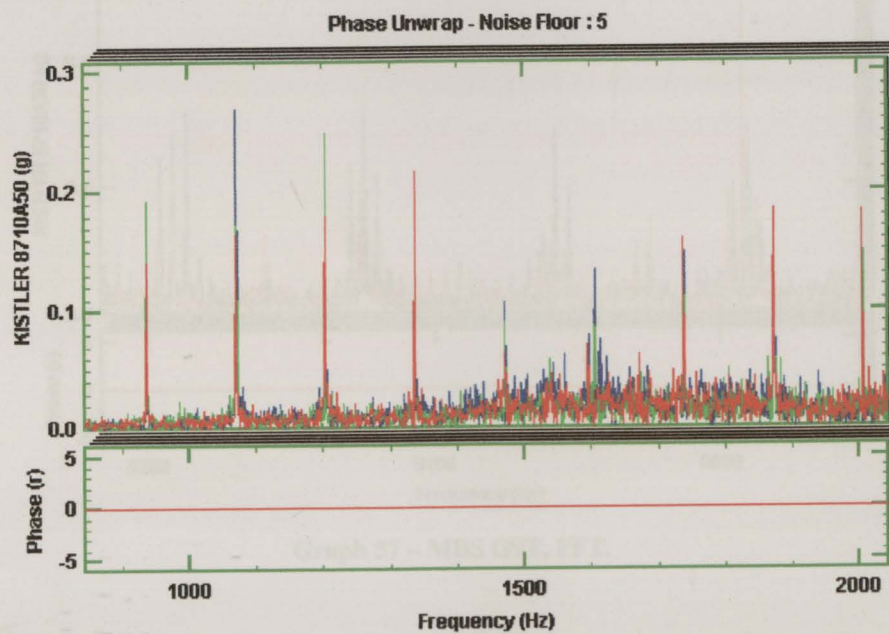
CACIA



RENAULT



Graph 54 – MBS G9T, FFT.

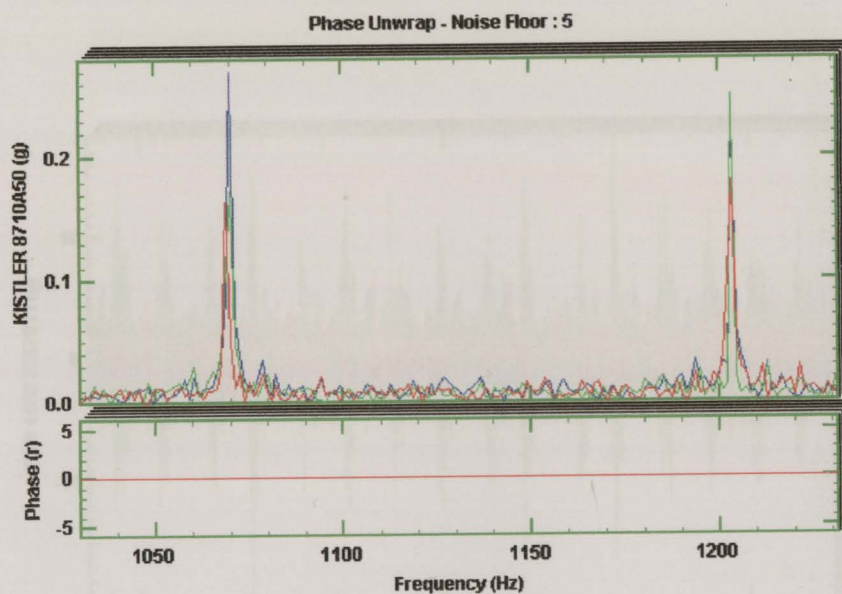


Graph 55 – MBS G9T, FFT.

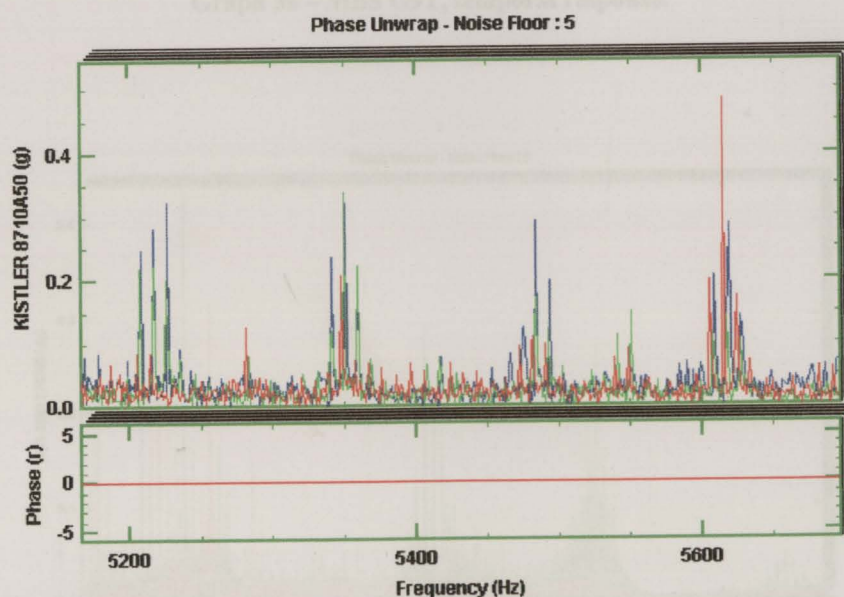


FINAL PROJECT

Experimental Comparison of the Mass Balance System
performance With Carbonitrided Steel Gears and
Austempered Ductile Iron Gears

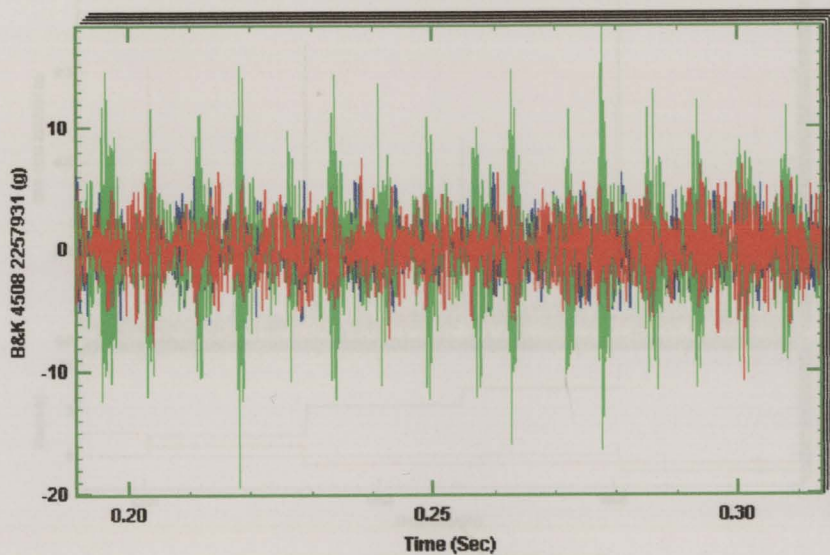


Graph 56 – MBS G9T, FFT.

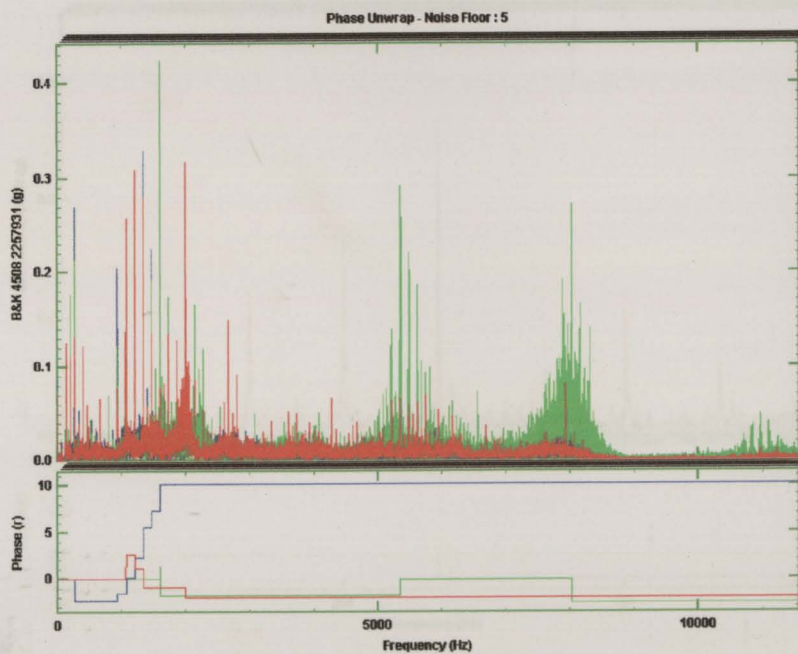


Graph 57 – MBS G9T, FFT.

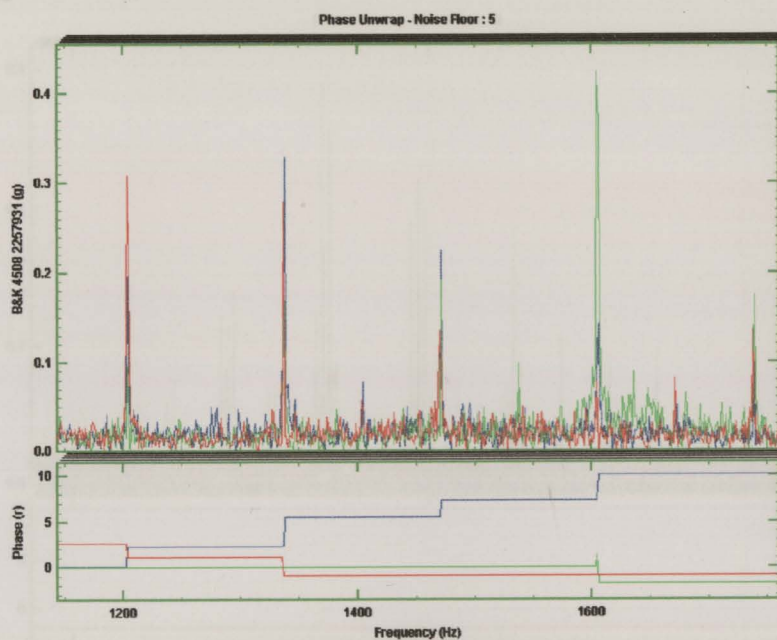
Direction Y 4000 rpm



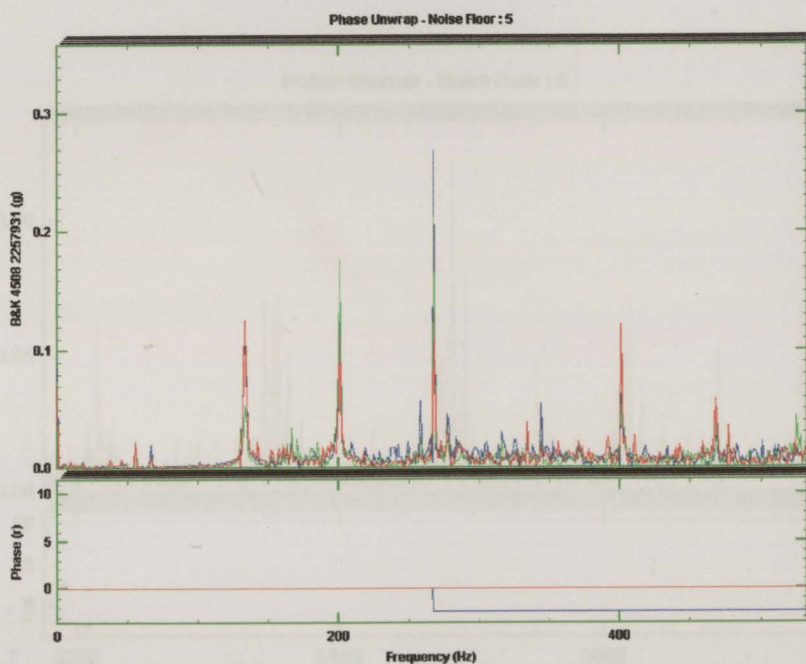
Graph 58 – MBS G9T, temporal response.



Graph 59 – MBS G9T, FFT



Graph 60 – MBS G9T, FFT detail.

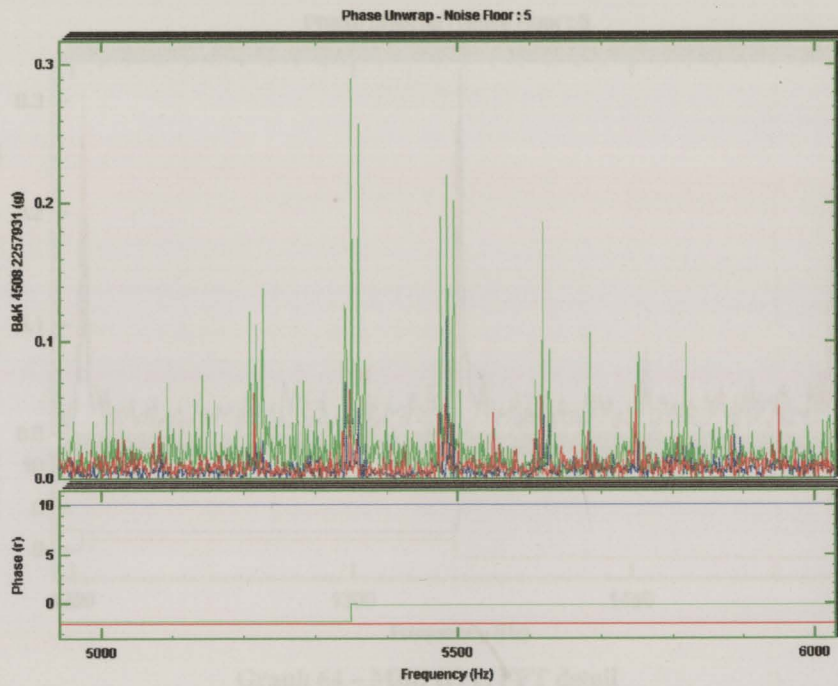


Graph 61 – MBS G9T, FFT detail

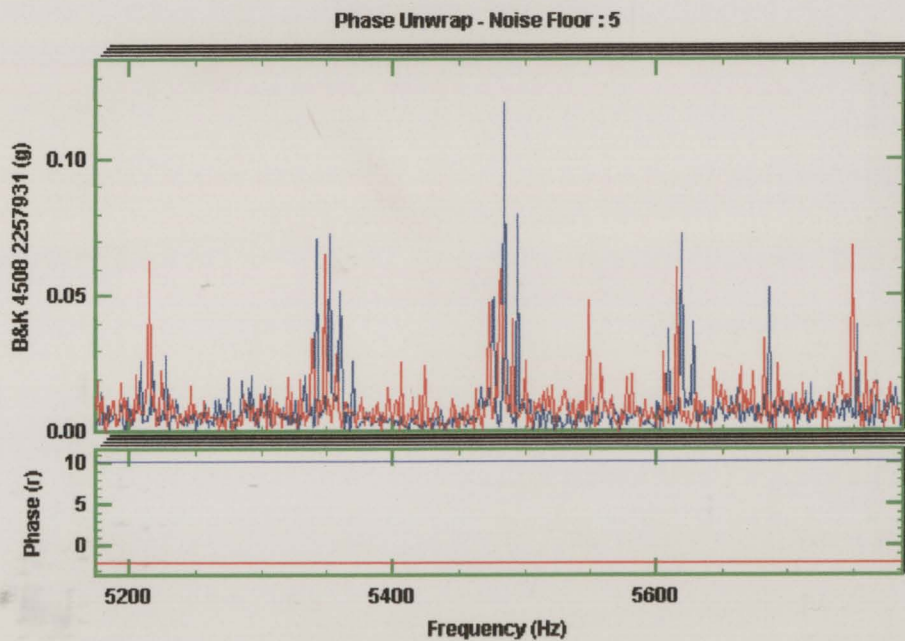


FINAL PROJECT

Experimental Comparison of the Mass Balance System
performance With Carbonitrided Steel Gears and
Austempered Ductile Iron Gears



Graph 62 – MBS G9T, FFT detail



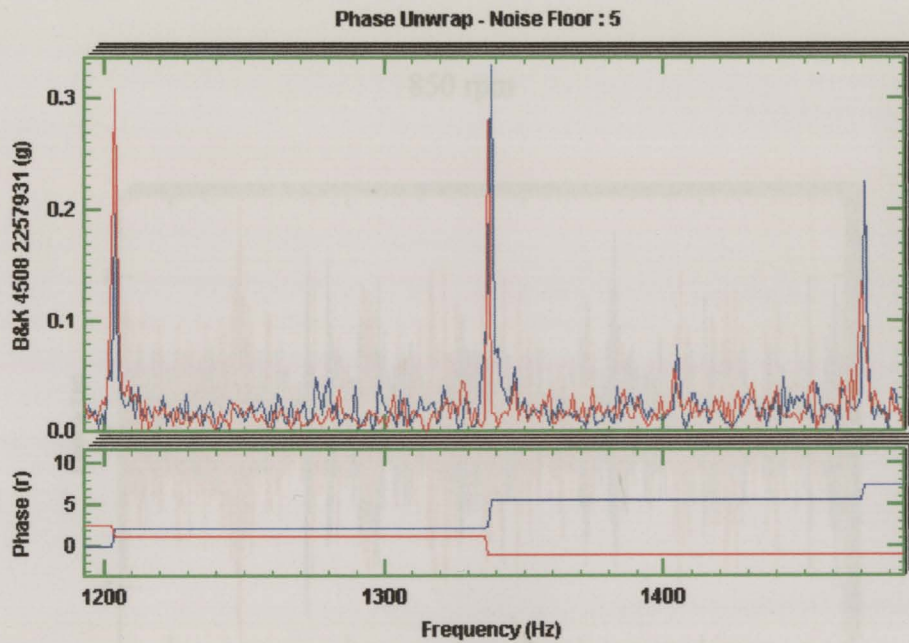
Graph 63 – MBS G9T, FFT detail





FINAL PROJECT

Experimental Comparison of the Mass Balance System
performance With Carbonitrided Steel Gears and
Austempered Ductile Iron Gears



Graph 64 – MBS G9T, FFT detail



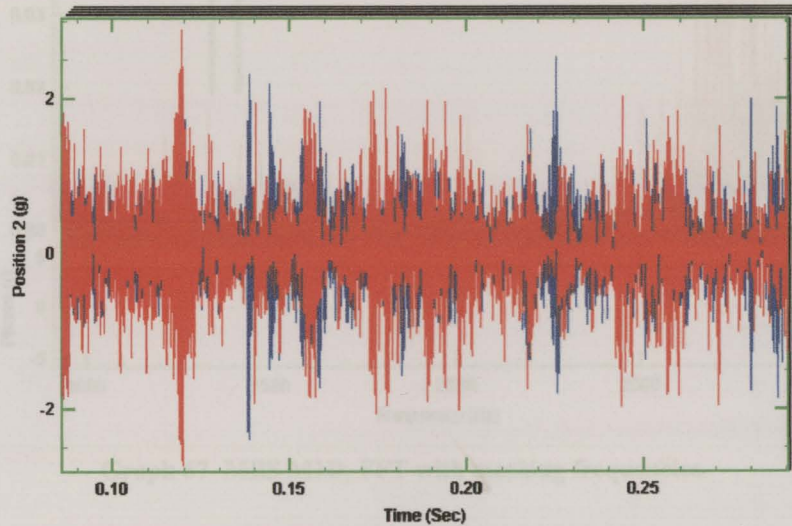
FINAL PROJECT

Experimental Comparison of the Mass Balance System
performance With Carbonitrided Steel Gears and
Austempered Ductile Iron Gears

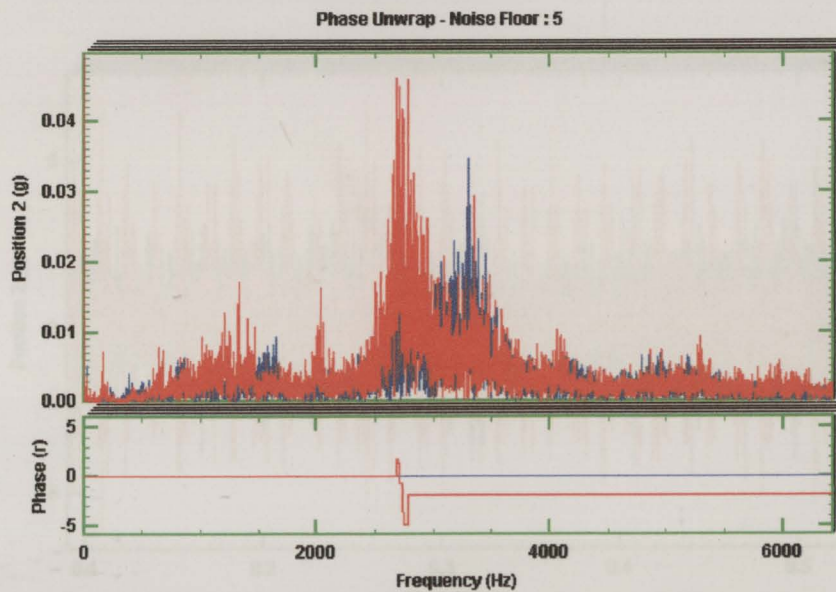


M1D

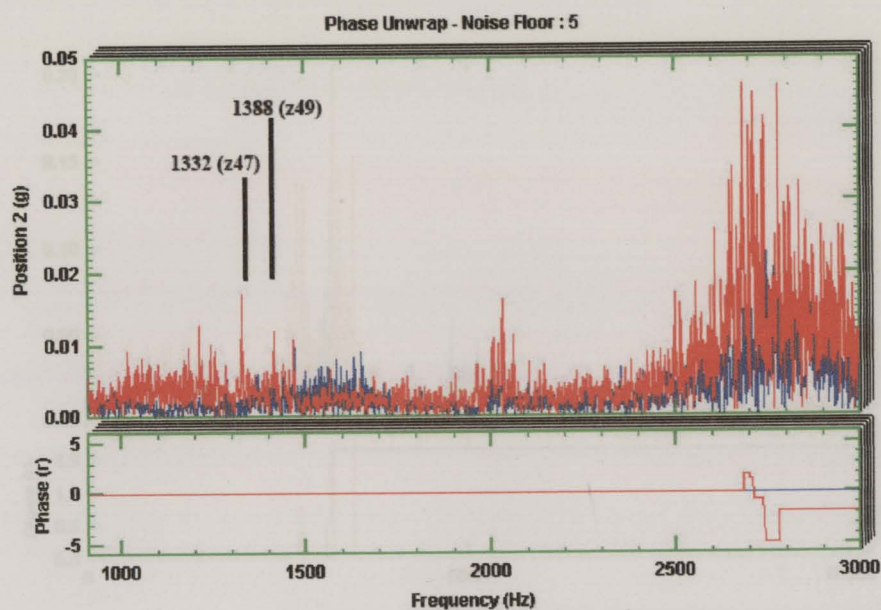
850 rpm



Graph 65 – MBS M1D, temporal response.

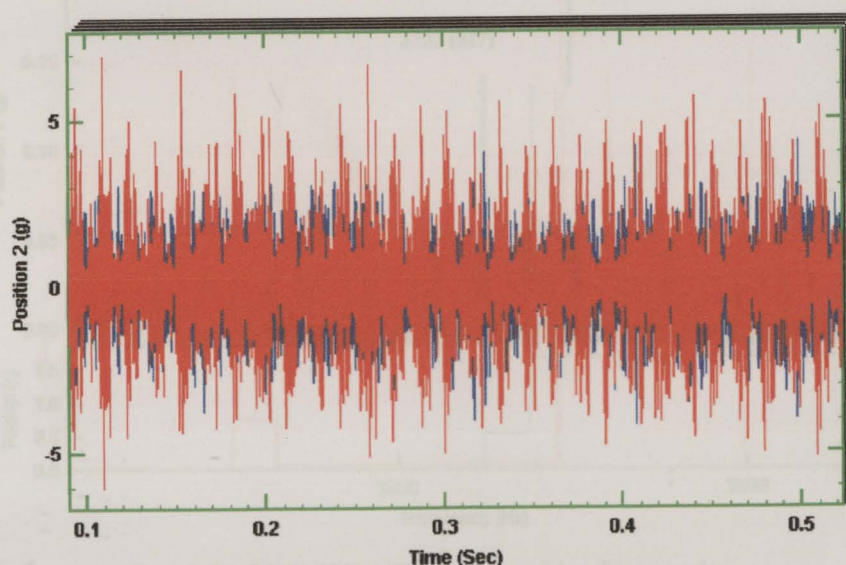


Graph 66- MBS M1D, FFT.



Graph 67- MBS M1D, FFT with meshing frequencies.

2000 rpm



Graph 68 - MBS M1D, temporal response.



FINAL PROJECT

Experimental Comparison of the Mass Balance System
performance With Carbonitrided Steel Gears and
Austempered Ductile Iron Gears



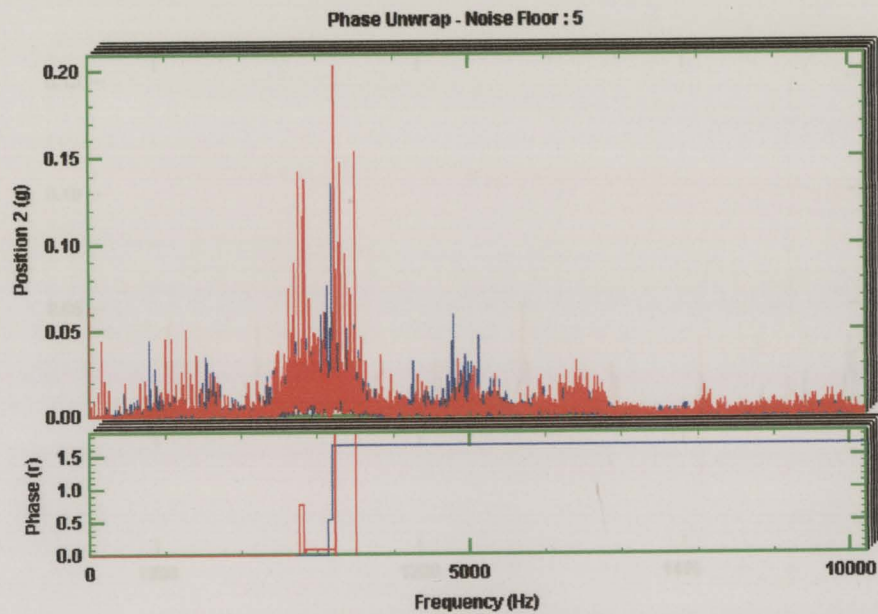
Universidade do Porto
Faculdade de Engenharia
FEUP



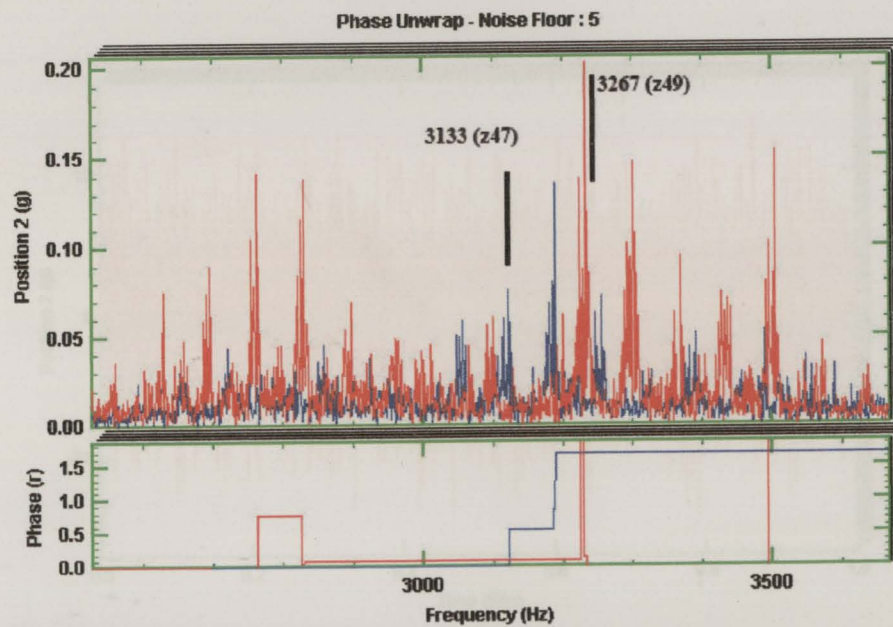
CACIA



RENAULT



Graph 69- MBS M1D, FFT.

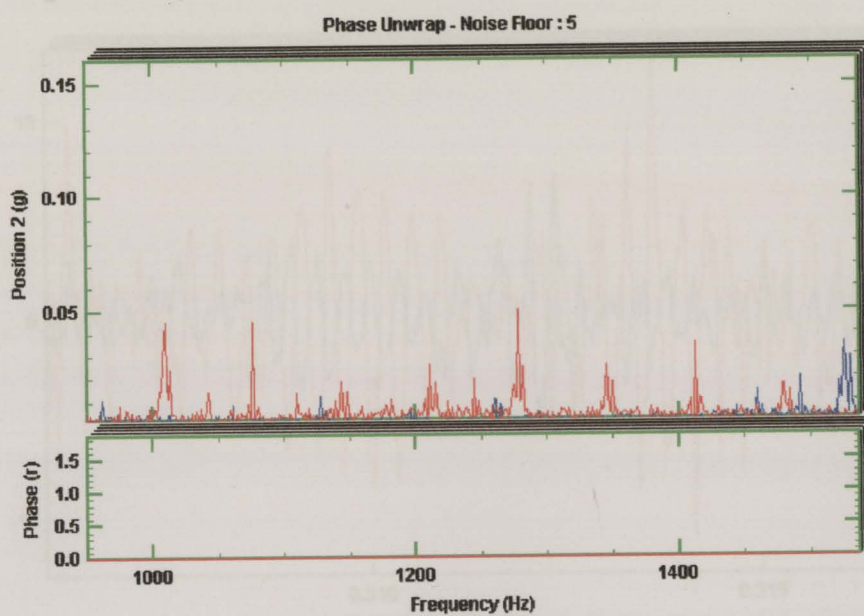


Graph 70- MBS M1D, FFT with meshing frequencies.



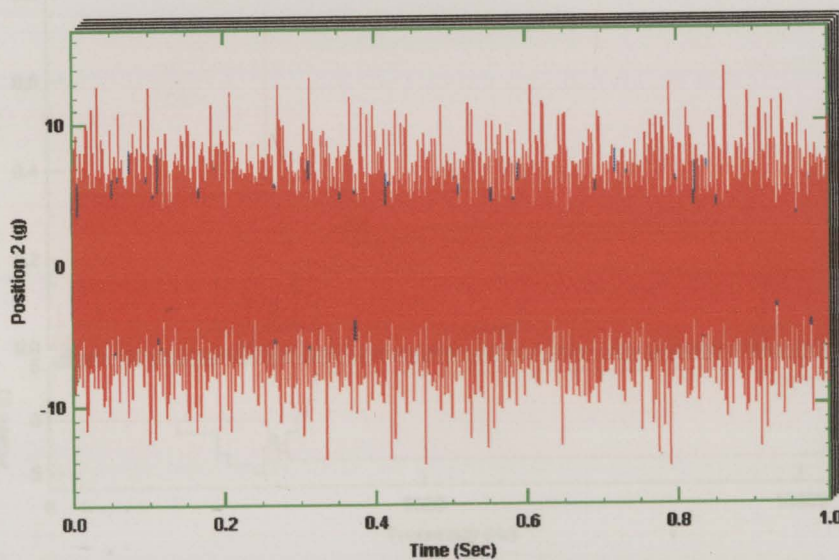
FINAL PROJECT

Experimental Comparison of the Mass Balance System
performance With Carbonitrided Steel Gears and
Austempered Ductile Iron Gears



Graph 71- MBS M1D, FFT detail.

4000 rpm

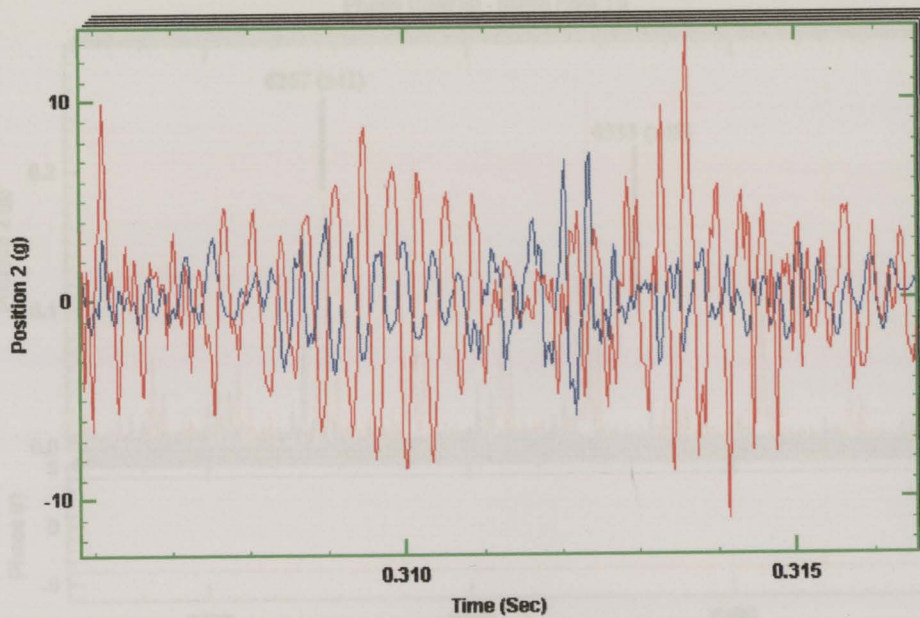


Graph 72 – MBS M1D, temporal response.

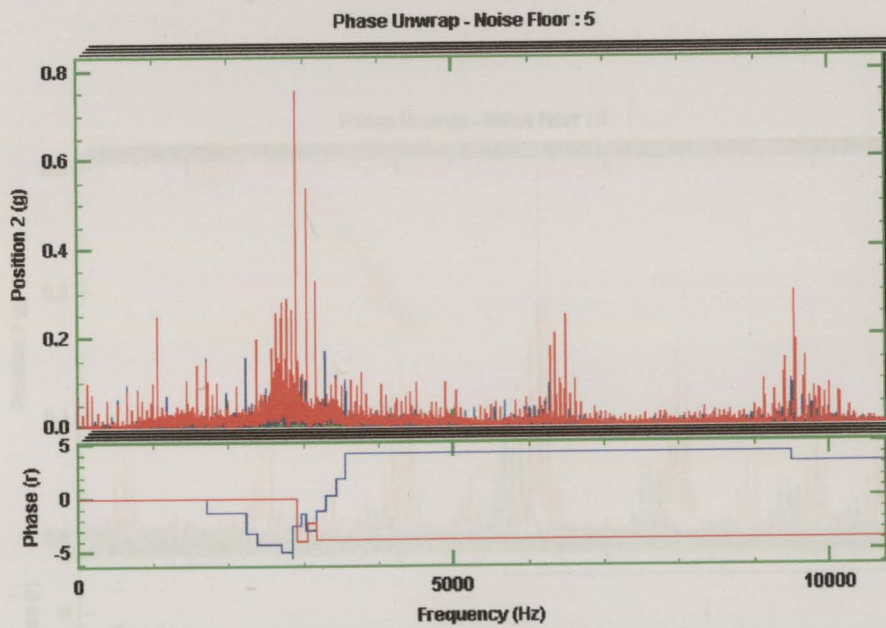


FINAL PROJECT

Experimental Comparison of the Mass Balance System
performance With Carbonitrided Steel Gears and
Austempered Ductile Iron Gears



Graph 73 - MBS M1D, temporal response detail.



Graph 74- MBS M1D, FFT.



FINAL PROJECT

Experimental Comparison of the Mass Balance System
performance With Carbonitrided Steel Gears and
Austempered Ductile Iron Gears



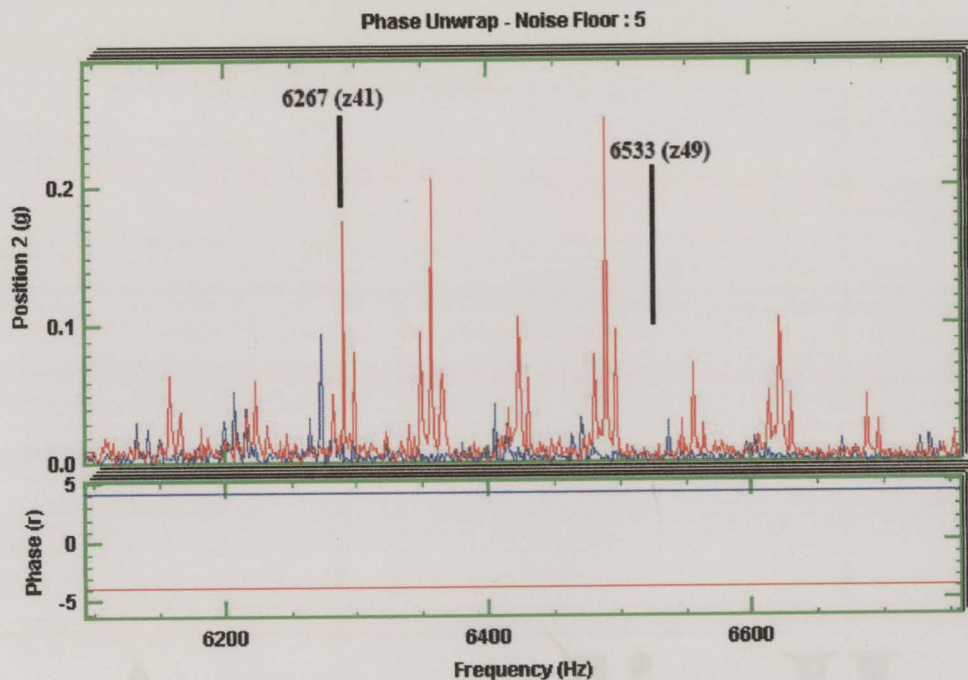
Universidade do Porto
Faculdade de Engenharia
FEUP



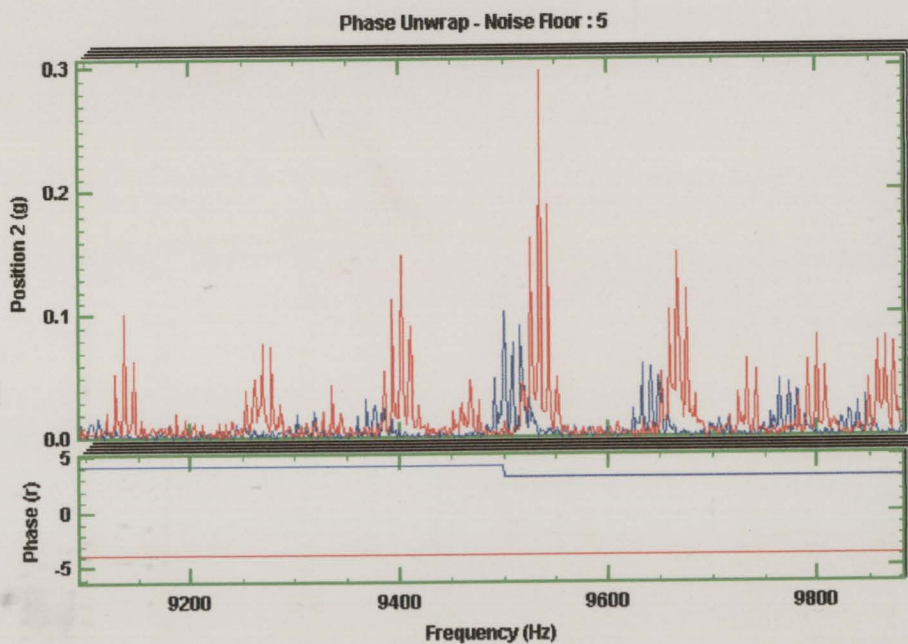
CACIA



RENAULT



Graph 75 - MBS M1D, FFT with meshing frequencies.



Graph 76 - MBS M1D, FFT detail.



FINAL PROJECT

Experimental Comparison of the Mass Balance System
performance With Carbonitrided Steel Gears and
Austempered Ductile Iron Gears



Universidade do Porto
Faculdade de Engenharia
FEUP



CAVIA



RENAULT



Appendix H



G.1. Inertias

These values were obtained with software CATIA.

Mass balance system G9T - Model 1.

Primary shaft

Gears	Shaft - part 1	Shaft - part 2	Mass	Shaft - part 3	Shaft - part 4	Mass	Shaft - part 5	Anilha
J 5,973E-04	J 1,014E-05	J 1,263E-05	J 1,881E-04	J 4,338E-06	J 1,697E-05	J 1,881E-04	J 6,624E-06	J 7,291E-06

Inertias to impute in AMESim

JA= 6,024E-04 JD= 1,310E-04
JB= 7,910E-05 JE= 7,951E-05
JC= 1,289E-04 JF= 1,060E-05

Secondary shaft

Gears	Shaft - part 1	Shaft - part 2	Mass	Shaft - part 3	Shaft - part 4	Mass	Shaft - part 5	Anilha
J 1,648E-04	J 1,014E-05	J 1,263E-05	J 1,881E-04	J 4,338E-06	J 1,697E-05	J 1,881E-04	J 6,624E-06	J 7,291E-06

Inertias to impute in AMESim

JA= 1,699E-04 JD= 1,310E-04
JB= 7,910E-05 JE= 7,951E-05
JC= 1,289E-04 JF= 1,060E-05



FINAL PROJECT

Experimental Comparison of the Mass Balance System
performance With Carbonitrided Steel Gears and
Austempered Ductile Iron Gears



Mass balance system G9T - Model 1 with ADI gears.

Primary shaft

Gears	Shaft - part 1	Shaft - part 2	Mass	Shaft - part 3	Shaft - part 4	Mass	Shaft - part 5	Anilha
J 5,376E-04	J 1,014E-05	J 1,263E-05	J 1,881E-04	J 4,338E-06	J 1,697E-05	J 1,881E-04	J 6,624E-06	J 7,291E-06

Inertias to impute in AMESim

JA= 5,427E-04 JD= 1,310E-04
JB= 7,910E-05 JE= 7,951E-05
JC= 1,289E-04 JF= 1,060E-05

Secondary shaft

Gears	Shaft - part 1	Shaft - part 2	Mass	Shaft - part 3	Shaft - part 4	Mass	Shaft - part 5	Anilha
J 1,483E-04	J 1,014E-05	J 1,263E-05	J 1,881E-04	J 4,338E-06	J 1,697E-05	J 1,881E-04	J 6,624E-06	J 7,291E-06

Inertias to impute in AMESim

JA= 1,534E-04 JD= 1,310E-04
JB= 7,910E-05 JE= 7,951E-05
JC= 1,289E-04 JF= 1,060E-05





FINAL PROJECT

Experimental Comparison of the Mass Balance System
performance With Carbonitrided Steel Gears and
Austempered Ductile Iron Gears



Mass balance system G9T - Model 2.

Primary shaft

Gears		Shaft - part 1		Shaft - part 2		Mass		Shaft - part 3		Mass		Anilha
J	5,973E-04	J	1,014E-05	J	1,263E-05	J	1,881E-04	J	1,697E-05	J	1,881E-04	J 7,29E-06

Inertias to impute in AMESim

JA=	6,024E-04	JC=	1,944E-04
JB=	1,224E-04	JD=	1,578E-05

Secondary shaft

Gears		Shaft - part 1		Shaft - part 2		Mass		Shaft - part 3		Mass		Anilha
J	1,648E-04	J	1,014E-05	J	1,263E-05	J	1,881E-04	J	1,697E-05	J	1,881E-04	J 7,29E-06

Inertias to impute in AMESim

JA=	1,699E-04	JC=	1,944E-04
JB=	1,224E-04	JD=	1,578E-05



FINAL PROJECT

Experimental Comparison of the Mass Balance System
performance With Carbonitrided Steel Gears and
Austempered Ductile Iron Gears



Mass balance system G9T - Model 2 with ADI gears.

Primary shaft

Gears	Shaft - part 1	Shaft - part 2	Mass	Shaft - part 3	Mass	Anilha
J 5,376E-04	J 1,014E-05	J 1,263E-05	J 1,881E-04	J 1,697E-05	J 1,881E-04	J 7,29E-06

Inertias to impute in AMESim

JA=	5,427E-04	JC=	1,944E-04
JB=	1,224E-04	JD=	1,578E-05

Secondary shaft

Gears	Shaft - part 1	Shaft - part 2	Mass	Shaft - part 3	Mass	Anilha
J 1,483E-04	J 1,014E-05	J 1,263E-05	J 1,881E-04	J 1,697E-05	J 1,881E-04	J 7,29E-06

Inertias to impute in AMESim

JA=	1,534E-04	JC=	1,944E-04
JB=	1,224E-04	JD=	1,578E-05



Mass balance system M1D.

Primary shaft

Part 1	Part 2	Part 3	Part 4	Part 5	Part 6
J 7,394E-04	J 8,992E-06	J 8,796E-06	J 1,668E-04	J 6,955E-06	J 1,664E-04

Inertias to impute in AMESim

JA= 3,697E-04	JE= 8,688E-05
JB= 3,742E-04	JF= 8,668E-05
JC= 8,894E-06	JG= 1,664E-04
JD= 8,780E-05	

Secondary shaft

Part 1	Part 2	Part 3	Part 4	Part 5	Part 6
J 3,868E-04	J 8,992E-06	J 8,796E-06	J 1,668E-04	J 6,955E-06	J 1,664E-04

Inertias to impute in AMESim

JA= 1,934E-04	JE= 8,688E-05
JB= 1,979E-04	JF= 8,668E-05
JC= 8,894E-06	JG= 1,664E-04
JD= 8,780E-05	



Mass balance system M1D with ADI gears.

Primary shaft

Part 1		Part 2		Part 3		Part 4		Part 5		Part 6	
J	6,758E-04	J	8,992E-06	J	8,796E-06	J	1,668E-04	J	6,955E-06	J	1,664E-04

Inertias to impute in AMESim

JA=	3,379E-04	JE=	8,688E-05
JB=	3,424E-04	JF=	8,668E-05
JC=	8,894E-06	JG=	1,664E-04
JD=	8,780E-05		

Secondary shaft

Part 1		Part 2		Part 3		Part 4		Part 5		Part 6	
J	3,585E-04	J	8,992E-06	J	8,796E-06	J	1,668E-04	J	6,955E-06	J	1,664E-04

Inertias to impute in AMESim

JA=	1,793E-04	JE=	8,688E-05
JB=	1,838E-04	JF=	8,668E-05
JC=	8,894E-06	JG=	1,664E-04
JD=	8,780E-05		



FINAL PROJECT

Experimental Comparison of the Mass Balance System
performance With Carbonitrided Steel Gears and
Austempered Ductile Iron Gears



G.2. Stiffness

Mass balance system G9T - Model 1.

Gxx $\nu = 0,3$
E= 2,10E+11 Pa
G= 8,08E+10

Primary shaft	Part 1		Part 2		Part 3		Part 4		Part 5	
	L1	53,39	L2	74	L3	24	L4	74	L5	39,51
	d	0	d	0	d	0	d	0	d	0
	D	22	D	22	D	22	D	22	D	22
	K1	34791,78	K2	25101,8	K3	77397,21	K4	25101,8	K5	47014,25 [N.m/rad]
	K1	1993422	K2	1438227	K3	4434534	K4	1438227	K5	2693718 [N.m/grau]

Secondary shaft	Part 1		Part 2		Part 3		Part 4		Part 5	
	L1	53,39	L2	74	L3	24	L4	74	L5	39,51
	d	0	d	0	d	0	d	0	d	0
	D	22	D	22	D	22	D	22	D	22
	K1	34791,78	K2	25101,8	K3	77397,21	K4	25101,8	K5	47014,25 [N.m/rad]
	K1	1993422	K2	1438227	K3	4434534	K4	1438227	K5	2693718 [N.m/grau]

Mass balance system G9T - Model 2



FINAL PROJECT

Experimental Comparison of the Mass Balance System
performance With Carbonitrided Steel Gears and
Austempered Ductile Iron Gears



Gxx

$\nu = 0,3$

$E = 2,10E+11 \text{ Pa}$

$G = 8,08E+10$

Primary shaft	Part 1		Part 2		Part 3	
	L1	53,39	L2	98	L3	113,51
	d	0	d	0	d	0
	D	22	D	22	D	22
	K1	34791,78	K2	18954,42	K3	16364,49 [N.m/rad]
	K1	1993422	K2	1086008	K3	937616,14 [N.m/grau]

Secondary shaft	Part 1		Part 2		Part 3	
	L1	53,39	L2	98	L3	113,51
	d	0	d	0	d	0
	D	22	D	22	D	22
	K1	34791,78	K2	18954,42	K3	16364,49 [N.m/rad]
	K1	1993422	K2	1086008	K3	937616,14 [N.m/grau]

Mass balance system M1D



$\nu = 0,3$
 $E = 2,10E+11 \text{ Pa}$
 $G = 8,08E+10$

Primary Shaft						Parte 1		Parte 2		Parte 3		Parte 4	
D	38,00	D	48,00	D	56,20	Ka	1181006,24	L2	26,00	L3	18,85	L4	25,87
d	0,00	d	0,00	d	0,00	Ka	3017425,69	d	8,50	d	8,50	d	8,50
L	14,00	L	13,95	L	3,00	Ka	26367554,28	D	25,94	D	28,00	D	44,60
Ka	1181006,24	Kb	3017425,69	Kc	26367554,28	(K1) ⁻¹	0,00	K2	136548,09	K3	256367,25	K4	1211199,03
						K1	822321,55	K2	7823629,28	K3	14688761,50	K4	69396592,50
						K1	47115554,48						

Parte 5					Parte 6							
D	44,00	D	32,94	Ka	4946522,89	D	44,60	D	44,60	Ka	7761634,60	
d	8,50	d	8,50	Kb	334082,11	d	8,50	d	12,00	Kb	1278969,19	
L	6,00	L	27,82			L	4,04	L	24,40			
Ka	4946522,89	Kb	334082,11	(K5) ⁻¹	0,00	Ka	7761634,60	Kb	1278969,19	(K6) ⁻¹	0,00	
				K5	312946,11					K6	1098034,13	[N.m/rad]
				K5	17930491,26					K6	62912721,57	[N.m/grau]



FINAL PROJECT

Experimental Comparison of the Mass Balance System
performance With Carbonitrided Steel Gears and
Austempered Ductile Iron Gears



$v = 0,3$
 $E = 2,10E+11 \text{ Pa}$
 $G = 8,08E+10$

Secondary shaft													
Parte 1						Parte 2		Parte 3		Parte 4			
D	38,00	D	48,00	D	56,20	Ka	1181006,24	L2	26,00	L3	18,85	L4	25,87
d	0,00	d	0,00	d	0,00	Ka	3017425,69	d	8,50	d	8,50	d	8,50
L	14,00	L	13,95	L	3,00	Ka	26367554,28	D	25,94	D	28,00	D	44,60
Ka	1181006,24	Kb	3017425,69	Kc	26367554,28	(kl) ⁻¹	0,00	K2	136548,09	K3	256367,25	K4	1211199,03
						K1	822321,55	K2	7823629,28	K3	14688761,50	K4	69396592,50
						K1	47115554,48						

Parte 5					Parte 6						
D	44,00	D	32,94	Ka	4946522,89	D	44,60	D	44,6	Ka	7761634,60
d	8,50	d	8,50	Kb	334082,11	d	8,50	d	12	Kb	1278969,19
L	6,00	L	27,82			L	4,04	L	24,403		
Ka	4946522,89	Kb	334082,11	(K5) ⁻¹	0,00	Ka	7761634,60	Kb	1278969,19	(K6) ⁻¹	0,00
				K5	312946,11					K6	1098034,13 [N.m/rad]
				K5	17930491,26					K6	62912721,57 [N.m/grau]



G.3. Mesh stiffness

Example: G9T 41-41 (steel) gears.

Mesh stiffness

ISO 6336-1: 1996(E)

Method B

Data introduction

$$e_{\alpha} = 1,61$$

$$c_{\gamma} = c' * (0,75 * e_{\alpha} + 0,25)$$

$$c_{\gamma} = 5,15171 \quad [N/(mm * mm)]$$

$$c_{\gamma} = 61820525 \quad [N/m]$$

Single Stiffness (steel/steel)

$$c'_{st/st} = c'_{th} * C_M * C_R * C_B * \cos(\beta)$$

$$c'_{st/st} = 10,61923$$

c'_{th}	C_M	C_R	C_B	β
13	0,8	1	1,087	20,05

Theoretical single stiffness

From figure 1

$$c'_{th} = 13$$

Correction factors

From figure 2

$$C_M = 0,8$$

$$C_R = 1$$

Materials combinations

$$c'_1 = c'_{st/st} * (E/E_{st})$$

$$c'_1 = 10,61923$$

$$(E_{st}/E_{st}) = 1 \quad \text{For steel / steel}$$

$$(E/E_{st}) = 0,74 \quad \text{For steel / gray cast iron}$$

$$(E/E_{st}) = 0,59 \quad \text{For grey cast iron / grey cast iron}$$



FINAL PROJECT

Experimental Comparison of the Mass Balance System
performance With Carbonitrided Steel Gears and
Austempered Ductile Iron Gears



Basic rack factor

h_{fp}	1,36	mm
m_n	1,38	
α_{pn}	18	°
C_B	1,087	

Specific loading

c'	3,534621
F_t	65,46323 N
K_A	2,25
b	12 mm

P	2,09446	W
T	2	N*m
n	10000	rpm
v	31,99445	m/s
d	61,103	mm

β	Nominal helix angle
ε_α	Transverse contact ratio
h_{fp}	Dedendum of basic rack of cylindrical gears
m_n	The rack normal module
α_{pn}	Normal pressure angle of the basic rack for cylindrical gears
K_A	Constant application factor (2,25 - in this case, for others situations see the table in the norm)
b	facewidth
T	Torque
n	Rotation speed
d	Pitch diameter

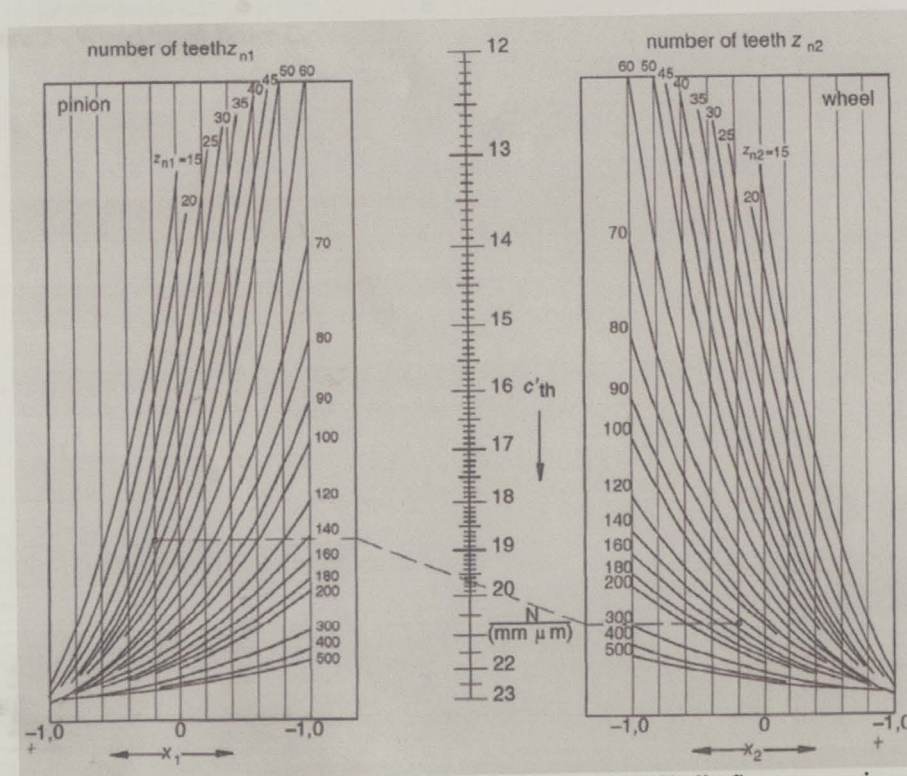
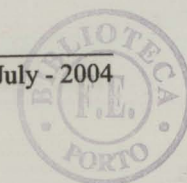


Figure 1 - Theoretical single stiffness of spur and virtual spur (of helical) gears, conjugate to basic rack profile, ISO 53: $\alpha_p=20^\circ$, $h_{ap}=m_n$, $h_{fp}=1,25 m_n$, $\rho_{fp}=0,25 m_n$; steel/steel; method B2 (see 9.3.1.1).





FINAL PROJECT

Experimental Comparison of the Mass Balance System
performance With Carbonitrided Steel Gears and
Austempered Ductile Iron Gears



Universidade do Porto
Faculdade de Engenharia
FEUP



CAGIA



RENAULT

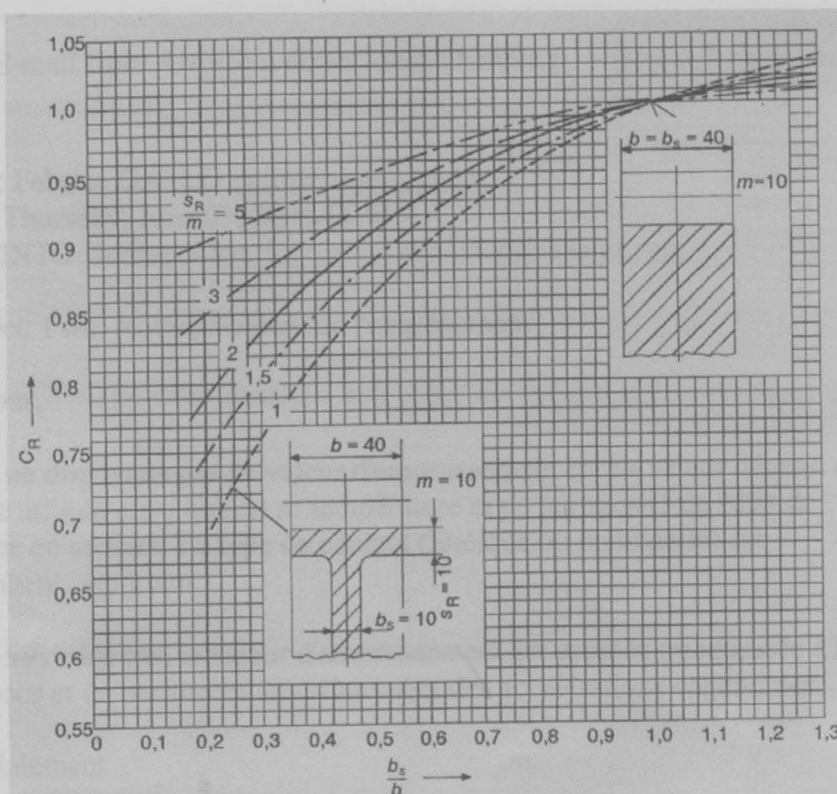


Figure 2 - Wheel blank factor C_R

Lyon technical office
3, rue Albert Thomas
69416 LYON Cedex 03 - FRANCE
tel : 33 (0)4 37 69 72 31
fax : 33 (0)47 8 34 39 61

Société IMAGINE SA
5, rue Dejean 42300 ROANNE - FRANCE
tel : 33 (0)477236030 fax : 33 (0)477236031

Advanced tools & Expertise in Systems Engineering
AMESim Simulation Software - Consulting - Training
Visit our Web Site : www.amesim.com

Original Message

From: "Pinto Henrique" <henrique.pinto@renault.com>
To: <galle@renault.com>
Cc: "Van Vu-do" <van.vu-do@renault.com>, "Pinto Rui"
<rui.pinto@renault.com>

Sent: Thursday, June 03, 2004 2:39 PM
Subject: Amortissement de l'engrènement



G.4. E-mail from AMESim about Mesh Damping.

From: Fabrice GALLO gallo@amesim.com
Sent: Thursday, June 03, 2004 13:46
To: PINTO Henrique henrique.pinto-renexter@renault.com
Cc:
Subject: Fwd. Amortissement de l'engrenement

Bonjour,

Nous ne disposons pas de valeur d'amortissement car sa valeur dépend de l'huile utilisée ainsi que de sa température et de la rugosité de l'état de surface en contact. Le type de contact (linéique ou ponctuelle) est également important.

Dans 99% des cas, la valeur d'amortissement est estimée ou mesurée à l'aide de chocs et de mesure en amortissement des accélérations engendrées.

Cordialement

Fabrice GALLO

Lyon technical office
3, cours Albert Thomas
69416 LYON Cedex 03 - FRANCE
tel : 33 (0)4 37 69 72 31
fax : 33 (0)47 8 54 39 61

Société IMAGINE SA
5, rue Brison 42300 ROANNE - FRANCE
tel : 33 (0)477236030 fax: 33 (0)477236031

| Advanced tools & Expertise in Systems Engineering
| AMESim Simulation Software - Consulting - Training
| Visit our Web Site : www.amesim.com

----- Original Message -----

From: "PINTO Henrique" <henrique.pinto-renexter@renault.com>
To: <gallo@amesim.com>
Cc: "Que Vu-do" <que.vu-do@renault.com>; "Pedro Reis" <pedro.reis@renault.com>
Sent: Thursday, June 03, 2004 2:39 PM
Subject: Amortissement de l'engrènement

Cher M. Gallo,

Je réalise actuellement un projet de fin d'études chez Renault - C.A.C.I.A. au Portugal, pour lequel je fais une étude sur l'utilisation de la fonte dans les engrenages. Pour cette étude, j'ai besoin de faire une simulation sur AMESim. Mon problème est que actuellement je ne sais pas comment calculer l'amortissement de l'engrènement. J'aimerais donc savoir si vous avez des valeurs d'amortissement d'engrènement pour l'acier et pour la fonte, ou si vous disposer d'une méthode pour obtenir ces valeurs ; car je ne trouve aucun article à ce sujet dans la bibliographie.

Je vous remercie d'avance pour de votre réponse, positive ou négative.

Cordialement,

Henrique Pinto



FACULDADE DE ENGENHARIA
UNIVERSIDADE DO PORTO

BIBLIOTECA



0000087311