Modelling And Design Of A Crankshaft Decoupler For Internal Combustion Engines

BY:

STEFANO MEDEI
B.S., Polytechnic of Turin, Italy, 2009
M.S., Polytechnic of Turin, Italy, 2011

THESIS

Submitted in partial fulfillment of the requirements for the degree of Master of Science in Mechanical Engineering in the Graduate College of the University of Illinois at Chicago, 2012

Chicago, Illinois

Defense Committee:
Farid Amrouche, Chief and Advisor
Michael Scott
Nicola Amati, Polytechnic of Turin
# TABLE OF CONTENTS

<table>
<thead>
<tr>
<th>CHAPTER</th>
<th>PAGE</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>INTRODUCTION</td>
</tr>
<tr>
<td>2</td>
<td>DESIGN AND WORKING PRINCIPLE OF THE CRANKSHAFT DECOUPLER</td>
</tr>
<tr>
<td>2.1</td>
<td>Design and working principle</td>
</tr>
<tr>
<td>2.2</td>
<td>Description of components</td>
</tr>
<tr>
<td>2.2.1</td>
<td>Spiral spring</td>
</tr>
<tr>
<td>2.2.2</td>
<td>Sliding pads</td>
</tr>
<tr>
<td>2.2.3</td>
<td>Helical springs</td>
</tr>
<tr>
<td>2.2.4</td>
<td>Pin head</td>
</tr>
<tr>
<td>2.2.5</td>
<td>Auxiliary spring housing</td>
</tr>
<tr>
<td>2.2.6</td>
<td>Journal bearing</td>
</tr>
<tr>
<td>2.2.7</td>
<td>Rubber ring</td>
</tr>
<tr>
<td>2.2.8</td>
<td>Inertia ring</td>
</tr>
<tr>
<td>2.2.9</td>
<td>Pulley</td>
</tr>
<tr>
<td>2.2.10</td>
<td>Hub</td>
</tr>
<tr>
<td>2.3</td>
<td>Internal clearances</td>
</tr>
<tr>
<td>2.4</td>
<td>Physical and geometrical data</td>
</tr>
<tr>
<td>3</td>
<td>STATIC CHARACTERISTIC</td>
</tr>
<tr>
<td>3.1</td>
<td>Basic module</td>
</tr>
<tr>
<td>3.1.1</td>
<td>Structure</td>
</tr>
<tr>
<td>3.1.2</td>
<td>Characteristic</td>
</tr>
<tr>
<td>3.2</td>
<td>Actual system</td>
</tr>
<tr>
<td>3.2.1</td>
<td>Structure</td>
</tr>
<tr>
<td>3.3</td>
<td>Experimental curve</td>
</tr>
<tr>
<td>3.4</td>
<td>Radial Load</td>
</tr>
<tr>
<td>3.4.1</td>
<td>Analytical solution</td>
</tr>
<tr>
<td>3.4.2</td>
<td>Numerical solution</td>
</tr>
<tr>
<td>4</td>
<td>KINEMATIC AND DYNAMIC ANALYSIS</td>
</tr>
<tr>
<td>4.1</td>
<td>Equivalent model</td>
</tr>
<tr>
<td>4.2</td>
<td>Configuration Space</td>
</tr>
<tr>
<td>4.3</td>
<td>State Space</td>
</tr>
<tr>
<td>4.4</td>
<td>Numerical model</td>
</tr>
<tr>
<td>5</td>
<td>EXPERIMENTAL VALIDATION</td>
</tr>
</tbody>
</table>
# TABLE OF CONTENTS (Continued)

<table>
<thead>
<tr>
<th>CHAPTER</th>
<th>PAGE</th>
</tr>
</thead>
<tbody>
<tr>
<td>5.1</td>
<td>Static test</td>
</tr>
<tr>
<td>5.2</td>
<td>Run up test with accessories removed</td>
</tr>
<tr>
<td>5.3</td>
<td>Run up test with accessories included</td>
</tr>
<tr>
<td>5.4</td>
<td>Start and Stop test</td>
</tr>
<tr>
<td>5.5</td>
<td>Root-Mean-Square Deviation</td>
</tr>
<tr>
<td>6</td>
<td>SENSITIVITY ANALYSIS</td>
</tr>
<tr>
<td>6.1</td>
<td>Output parameter of the analysis</td>
</tr>
<tr>
<td>6.2</td>
<td>Input parameter of the analysis</td>
</tr>
<tr>
<td>6.3</td>
<td>Influence of T1</td>
</tr>
<tr>
<td>6.4</td>
<td>Influence of T2</td>
</tr>
<tr>
<td>6.5</td>
<td>Influence of G4</td>
</tr>
<tr>
<td>6.6</td>
<td>Influence of $k_{ss2}$</td>
</tr>
<tr>
<td>6.7</td>
<td>Influence of the ramp</td>
</tr>
<tr>
<td>7</td>
<td>PROPOSED DESIGN SOLUTIONS</td>
</tr>
<tr>
<td>7.1</td>
<td>First solution</td>
</tr>
<tr>
<td>7.2</td>
<td>Second Solution</td>
</tr>
<tr>
<td>8</td>
<td>CONCLUSION</td>
</tr>
<tr>
<td>APPENDICES</td>
<td>149</td>
</tr>
<tr>
<td>Appendix A</td>
<td>150</td>
</tr>
<tr>
<td>Appendix B</td>
<td>155</td>
</tr>
<tr>
<td>Appendix C</td>
<td>159</td>
</tr>
<tr>
<td>Appendix D</td>
<td>168</td>
</tr>
<tr>
<td>CITED LITERATURE</td>
<td>174</td>
</tr>
<tr>
<td>VITA</td>
<td>176</td>
</tr>
</tbody>
</table>
# LIST OF TABLES

<table>
<thead>
<tr>
<th>TABLE</th>
<th>PAGE</th>
</tr>
</thead>
<tbody>
<tr>
<td>I INTERNAL CLEARANCES</td>
<td>25</td>
</tr>
<tr>
<td>II DATA COMPONENT</td>
<td>26</td>
</tr>
<tr>
<td>III FEATURES OF THE FIRST MODULE</td>
<td>34</td>
</tr>
<tr>
<td>IV FEATURES OF THE SECOND MODULE</td>
<td>35</td>
</tr>
<tr>
<td>V FEATURES OF THE LAST TWO MODULES</td>
<td>36</td>
</tr>
<tr>
<td>VI ROOT MEAN SQUARE ERRORS</td>
<td>88</td>
</tr>
<tr>
<td>VII FILTERING RATIO OF THE FIRST SOLUTION, ACCESSORIES EXCLUDED</td>
<td>137</td>
</tr>
<tr>
<td>VIII FILTERING RATIO OF THE FIRST SOLUTION, ACCESSORIES INCLUDED</td>
<td>137</td>
</tr>
<tr>
<td>IX FILTERING RATIO OF THE SECOND SOLUTION, ACCESSORIES EXCLUDED</td>
<td>147</td>
</tr>
<tr>
<td>X FILTERING RATIO OF THE SECOND SOLUTION, ACCESSORIES INCLUDED</td>
<td>147</td>
</tr>
</tbody>
</table>
## LIST OF FIGURES

<table>
<thead>
<tr>
<th>FIGURE</th>
<th>DESCRIPTION</th>
<th>PAGE</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Front end accessory drive</td>
<td>1</td>
</tr>
<tr>
<td>2</td>
<td>Litens solution</td>
<td>6</td>
</tr>
<tr>
<td>3</td>
<td>Frontal view and section of the DDOS</td>
<td>9</td>
</tr>
<tr>
<td>4</td>
<td>Exploded view of the device</td>
<td>11</td>
</tr>
<tr>
<td>5</td>
<td>Exploded view of the TVD</td>
<td>12</td>
</tr>
<tr>
<td>6</td>
<td>Exploded view of the decoupler</td>
<td>13</td>
</tr>
<tr>
<td>7</td>
<td>Zoom on the secondary spring group</td>
<td>15</td>
</tr>
<tr>
<td>8</td>
<td>Forces acting on the pulley</td>
<td>18</td>
</tr>
<tr>
<td>9</td>
<td>Radial Load</td>
<td>18</td>
</tr>
<tr>
<td>10</td>
<td>Zoom on the inner surface of the pulley</td>
<td>20</td>
</tr>
<tr>
<td>11</td>
<td>Internal clearances</td>
<td>24</td>
</tr>
<tr>
<td>12</td>
<td>Basic module</td>
<td>28</td>
</tr>
<tr>
<td>13</td>
<td>Trend of the force at the beginning of motion</td>
<td>29</td>
</tr>
<tr>
<td>14</td>
<td>Force transmitted while the system is slipping</td>
<td>29</td>
</tr>
<tr>
<td>15</td>
<td>Maximum force transmitted</td>
<td>30</td>
</tr>
<tr>
<td>16</td>
<td>Basic module max compressed</td>
<td>30</td>
</tr>
<tr>
<td>17</td>
<td>Force transmitted during spring releasing and reloading</td>
<td>31</td>
</tr>
<tr>
<td>18</td>
<td>Force transmitted during moving backward</td>
<td>31</td>
</tr>
<tr>
<td>19</td>
<td>Basic module max tight</td>
<td>32</td>
</tr>
<tr>
<td>FIGURE</td>
<td>Description</td>
<td>PAGE</td>
</tr>
<tr>
<td>--------</td>
<td>------------------------------------------------------------------------------</td>
<td>------</td>
</tr>
<tr>
<td>20</td>
<td>Static characteristic of the basic module</td>
<td>33</td>
</tr>
<tr>
<td>21</td>
<td>Structure of the DDOS</td>
<td>34</td>
</tr>
<tr>
<td>22</td>
<td>Characteristic of the DDOS</td>
<td>36</td>
</tr>
<tr>
<td>23</td>
<td>Experimental characteristic of the DDOS</td>
<td>37</td>
</tr>
<tr>
<td>24</td>
<td>Actual static characteristic</td>
<td>42</td>
</tr>
<tr>
<td>25</td>
<td>Equivalent model of the DDOS</td>
<td>44</td>
</tr>
<tr>
<td>26</td>
<td>Graph of friction</td>
<td>45</td>
</tr>
<tr>
<td>27</td>
<td>Graph of G1</td>
<td>45</td>
</tr>
<tr>
<td>28</td>
<td>Graph of G2</td>
<td>46</td>
</tr>
<tr>
<td>29</td>
<td>Graph of G4</td>
<td>46</td>
</tr>
<tr>
<td>30</td>
<td>Equivalent linear model of the DDOS</td>
<td>47</td>
</tr>
<tr>
<td>31</td>
<td>Simulink model of the DDOS</td>
<td>57</td>
</tr>
<tr>
<td>32</td>
<td>State-Space block</td>
<td>58</td>
</tr>
<tr>
<td>33</td>
<td>Rotary speed block</td>
<td>58</td>
</tr>
<tr>
<td>34</td>
<td>Experimental angular speed filter block</td>
<td>59</td>
</tr>
<tr>
<td>35</td>
<td>Block of the external contribute due to the accessories</td>
<td>59</td>
</tr>
<tr>
<td>36</td>
<td>Non linearities block</td>
<td>61</td>
</tr>
<tr>
<td>37</td>
<td>Block of the two friction elements</td>
<td>62</td>
</tr>
<tr>
<td>38</td>
<td>Block of friction T1</td>
<td>63</td>
</tr>
<tr>
<td>39</td>
<td>Ramp friction block</td>
<td>64</td>
</tr>
<tr>
<td>40</td>
<td>Block of friction T2</td>
<td>65</td>
</tr>
<tr>
<td>FIGURE</td>
<td>FIGURE DESCRIPTION</td>
<td>PAGE</td>
</tr>
<tr>
<td>--------</td>
<td>-------------------</td>
<td>------</td>
</tr>
<tr>
<td>41</td>
<td>Bushing friction block</td>
<td>65</td>
</tr>
<tr>
<td>42</td>
<td>Internal clearances block</td>
<td>66</td>
</tr>
<tr>
<td>43</td>
<td>Simulink model for the static test simulation of the DDOS</td>
<td>68</td>
</tr>
<tr>
<td>44</td>
<td>Crankshaft rotary speed for static test simulation</td>
<td>69</td>
</tr>
<tr>
<td>45</td>
<td>External torque block diagram for static test simulation</td>
<td>70</td>
</tr>
<tr>
<td>46</td>
<td>Numerical static characteristic of the DDOS</td>
<td>71</td>
</tr>
<tr>
<td>47</td>
<td>Experimental static characteristic of the DDOS</td>
<td>72</td>
</tr>
<tr>
<td>48</td>
<td>Time history of the crankshaft rotary speed with accessories removed</td>
<td>73</td>
</tr>
<tr>
<td>49</td>
<td>Comparison between numerical and experimental result of a run up speed with accessories removed</td>
<td>74</td>
</tr>
<tr>
<td>50</td>
<td>Numerical and experimental response of the pulley moving at constant mean speed with accessories removed</td>
<td>75</td>
</tr>
<tr>
<td>51</td>
<td>Numerical and experimental response of the pulley on the speed ramp with accessories removed</td>
<td>76</td>
</tr>
<tr>
<td>52</td>
<td>Comparison between experimental and filtered pulley spin speed during the run up with no external load</td>
<td>76</td>
</tr>
<tr>
<td>53</td>
<td>Comparison between numerical and filtered result of a run up speed with no accessories</td>
<td>77</td>
</tr>
<tr>
<td>54</td>
<td>Numerical and filtered response of the pulley moving at constant mean speed with no accessories</td>
<td>78</td>
</tr>
<tr>
<td>55</td>
<td>Numerical and filtered response of the pulley on the ramp with accessories removed</td>
<td>78</td>
</tr>
<tr>
<td>56</td>
<td>Time history of the crankshaft spin speed with accessories included</td>
<td>79</td>
</tr>
<tr>
<td>57</td>
<td>Comparison between numerical and experimental result of a run up speed with the external load</td>
<td>80</td>
</tr>
</tbody>
</table>
LIST OF FIGURES (Continued)

<table>
<thead>
<tr>
<th>FIGURE</th>
<th>PAGE</th>
</tr>
</thead>
<tbody>
<tr>
<td>58</td>
<td>Numerical and experimental response of the pulley moving at constant mean speed with accessories included</td>
</tr>
<tr>
<td>59</td>
<td>Numerical and experimental response of the pulley on the speed ramp with accessories turned on</td>
</tr>
<tr>
<td>60</td>
<td>Comparison between experimental and filtered pulley spin speed during the run up with the external load</td>
</tr>
<tr>
<td>61</td>
<td>Comparison between numerical and filtered result of a run up speed with accessories included</td>
</tr>
<tr>
<td>62</td>
<td>Numerical and filtered response of the pulley moving at constant mean speed with accessories included</td>
</tr>
<tr>
<td>63</td>
<td>Numerical and filtered response of the pulley on the ramp with the external load acting on the pulley</td>
</tr>
<tr>
<td>64</td>
<td>Time history of the crankshaft spin speed during the Start and Stop test</td>
</tr>
<tr>
<td>65</td>
<td>Numerical and experimental pulley angular speed during the Start and Stop test</td>
</tr>
<tr>
<td>66</td>
<td>Numerical and experimental pulley angular speed during the initial phase</td>
</tr>
<tr>
<td>67</td>
<td>Numerical and experimental pulley rotary speed during the final phase</td>
</tr>
<tr>
<td>68</td>
<td>Experimental and filtered pulley angular speed</td>
</tr>
<tr>
<td>69</td>
<td>Comparison between filtered and experimental response of the pulley during the initial phase</td>
</tr>
<tr>
<td>70</td>
<td>Comparison between filtered and experimental response of the pulley during the final phase</td>
</tr>
<tr>
<td>71</td>
<td>Filtered and numerical pulley speed during the Start and Stop test</td>
</tr>
<tr>
<td>72</td>
<td>Zoom of the filtered and numerical response during the initial phase</td>
</tr>
<tr>
<td>73</td>
<td>Zoom of the filtered and numerical response during the final phase</td>
</tr>
<tr>
<td>74</td>
<td>Simulation response for initial condition equal to 10</td>
</tr>
<tr>
<td>FIGURE</td>
<td>DESCRIPTION</td>
</tr>
<tr>
<td>--------</td>
<td>-------------------------------------------------------------------------------------------------</td>
</tr>
<tr>
<td>75</td>
<td>Zoom of the starting of the start and Stop test simulation for initial condition equal to 10</td>
</tr>
<tr>
<td>76</td>
<td>Simulation response for initial condition equal to 10</td>
</tr>
<tr>
<td>77</td>
<td>Zoom of the starting of the start and Stop test simulation for initial condition equal to 10</td>
</tr>
<tr>
<td>78</td>
<td>Comparison of several output pulley spin speeds for different initial conditions</td>
</tr>
<tr>
<td>79</td>
<td>Transitory effect of the initial conditions on the pulley rotary speed</td>
</tr>
<tr>
<td>80</td>
<td>Influence of the friction T1 on the response of the pulley when the accessories are excluded</td>
</tr>
<tr>
<td>81</td>
<td>Influence of the friction T1 on the response of the pulley when the accessories are included</td>
</tr>
<tr>
<td>82</td>
<td>Influence of the friction T1 on the response of the pulley during the start and the stop of the engine</td>
</tr>
<tr>
<td>83</td>
<td>Influence of T1 on the impact with the secondary springs during the run up with accessories excluded</td>
</tr>
<tr>
<td>84</td>
<td>Influence of T1 on the impact with the secondary springs during start and stop</td>
</tr>
<tr>
<td>85</td>
<td>Response of the system equipped with a ball bearing, instead of the journal bearing, when the accessories are excluded</td>
</tr>
<tr>
<td>86</td>
<td>Response of the system equipped with a ball bearing, instead of the journal bearing, when the accessories are included</td>
</tr>
<tr>
<td>87</td>
<td>Response of the system equipped with a ball bearing, instead of the journal bearing, during and the start and the stop of the engine</td>
</tr>
<tr>
<td>88</td>
<td>Response of the system equipped with a ball bearing, instead of the journal bearing, and a cover when the accessories are excluded</td>
</tr>
<tr>
<td>89</td>
<td>Response of the system equipped with a ball bearing, instead of the journal bearing, and a cover when the accessories are included</td>
</tr>
<tr>
<td>FIGURE</td>
<td>PAGE</td>
</tr>
<tr>
<td>--------</td>
<td>------</td>
</tr>
<tr>
<td>90</td>
<td>109</td>
</tr>
<tr>
<td>Response of the system equipped with a ball bearing, instead of the journal bearing, and a cover during and the start and the stop of the engine</td>
<td></td>
</tr>
<tr>
<td>91</td>
<td>110</td>
</tr>
<tr>
<td>Zooming on the start of the DDOS equipped with ball bearing and cover</td>
<td></td>
</tr>
<tr>
<td>92</td>
<td>111</td>
</tr>
<tr>
<td>Influence of T2 on the force G1 during the run up with accessories excluded</td>
<td></td>
</tr>
<tr>
<td>93</td>
<td>112</td>
</tr>
<tr>
<td>Influence of T2 on the force G1 during the run up with accessories included</td>
<td></td>
</tr>
<tr>
<td>94</td>
<td>113</td>
</tr>
<tr>
<td>Influence of T2 on the force G1 during the start and stop of the engine</td>
<td></td>
</tr>
<tr>
<td>95</td>
<td>114</td>
</tr>
<tr>
<td>Influence of T2 on the force G2 during the run up with accessories excluded</td>
<td></td>
</tr>
<tr>
<td>96</td>
<td>115</td>
</tr>
<tr>
<td>Influence of T2 on the force G2 during the run up with accessories included</td>
<td></td>
</tr>
<tr>
<td>97</td>
<td>116</td>
</tr>
<tr>
<td>Influence of T2 on the force G2 during the start and stop of the engine</td>
<td></td>
</tr>
<tr>
<td>98</td>
<td>117</td>
</tr>
<tr>
<td>Influence of the clearing G4 on the filtering during the run up with accessories excluded</td>
<td></td>
</tr>
<tr>
<td>99</td>
<td>118</td>
</tr>
<tr>
<td>Influence of the clearing G4 on the filtering during the run up with accessories included</td>
<td></td>
</tr>
<tr>
<td>100</td>
<td>119</td>
</tr>
<tr>
<td>Influence of the clearing G4 on the filtering during the start and stop of the engine</td>
<td></td>
</tr>
<tr>
<td>101</td>
<td>120</td>
</tr>
<tr>
<td>Zoom in on the influence of the clearance G4 at the start</td>
<td></td>
</tr>
<tr>
<td>102</td>
<td>121</td>
</tr>
<tr>
<td>Influence of the clearance G4 on the impact between pulley and the main spring during the run up with accessories excluded</td>
<td></td>
</tr>
<tr>
<td>103</td>
<td>121</td>
</tr>
<tr>
<td>Influence of the clearance G4 on the impact between pulley and the main spring during the run up with accessories included</td>
<td></td>
</tr>
<tr>
<td>104</td>
<td>122</td>
</tr>
<tr>
<td>Influence of the clearance G4 on the impact between pulley and the main spring during the start and stop of the engine</td>
<td></td>
</tr>
<tr>
<td>105</td>
<td>122</td>
</tr>
<tr>
<td>Influence of the clearance G4 on the impact between pulley and the secondary spring $k_{ss2}$ during the run up with accessories excluded</td>
<td></td>
</tr>
<tr>
<td>FIGURE</td>
<td>Description</td>
</tr>
<tr>
<td>--------</td>
<td>-----------------------------------------------------------------------------</td>
</tr>
<tr>
<td>106</td>
<td>Influence of the clearance G4 on the impact between pulley and the secondary spring $k_{ss2}$ during the run up with accessories included</td>
</tr>
<tr>
<td>107</td>
<td>Influence of the clearance G4 on the impact between pulley and the secondary spring $k_{ss2}$ during the start and stop of the engine</td>
</tr>
<tr>
<td>108</td>
<td>Shape of the ramp in the original design</td>
</tr>
<tr>
<td>109</td>
<td>Shapes of the modified ramps</td>
</tr>
<tr>
<td>110</td>
<td>Influence of the ramp on the pulley spin speed during the run up with accessories excluded</td>
</tr>
<tr>
<td>111</td>
<td>Influence of the ramp on the pulley spin speed during the run up with accessories included</td>
</tr>
<tr>
<td>112</td>
<td>Influence of the ramp on the pulley spin speed during the start and stop of the engine</td>
</tr>
<tr>
<td>113</td>
<td>Influence of the ramp on the force G1 during the run up with accessories excluded</td>
</tr>
<tr>
<td>114</td>
<td>Influence of the ramp on the force G1 during the run up with accessories included</td>
</tr>
<tr>
<td>115</td>
<td>Influence of the ramp on the force G1 during the start and stop of the engine</td>
</tr>
<tr>
<td>116</td>
<td>Original and modified ramps</td>
</tr>
<tr>
<td>117</td>
<td>Pulley speed of the modified DDOS with accessories excluded</td>
</tr>
<tr>
<td>118</td>
<td>Pulley speed of the modified DDOS with accessories included</td>
</tr>
<tr>
<td>119</td>
<td>Oscillations of the modified DDOS at the Start</td>
</tr>
<tr>
<td>120</td>
<td>Amplitude of the internal impacts of the modified DDOS with accessories excluded</td>
</tr>
<tr>
<td>121</td>
<td>Amplitude of the internal impacts of the modified DDOS with accessories included</td>
</tr>
<tr>
<td>FIGURE</td>
<td>PAGE</td>
</tr>
<tr>
<td>--------</td>
<td>------</td>
</tr>
<tr>
<td>122</td>
<td>Amplitude of the internal impacts of the modified DDOS at the Start .</td>
</tr>
<tr>
<td>123</td>
<td>Filtering of the first solution when the accessories are excluded . . . . .</td>
</tr>
<tr>
<td>124</td>
<td>Filtering of the first solution when the accessories are included . . . . .</td>
</tr>
<tr>
<td>125</td>
<td>Filtering ratio at 1000 rpm with accessories excluded . . . . .</td>
</tr>
<tr>
<td>126</td>
<td>Filtering ratio at 2000 rpm with accessories excluded . . . . .</td>
</tr>
<tr>
<td>127</td>
<td>Filtering ratio at 3000 rpm with accessories excluded . . . . .</td>
</tr>
<tr>
<td>128</td>
<td>Filtering ratio at 1000 rpm with accessories included . . . . .</td>
</tr>
<tr>
<td>129</td>
<td>Filtering ratio at 2000 rpm with accessories included . . . . .</td>
</tr>
<tr>
<td>130</td>
<td>Filtering ratio at 3000 rpm with accessories included . . . . .</td>
</tr>
<tr>
<td>131</td>
<td>Pulley speed of the DDOS with the pre-engagement spring when the accessories are excluded .</td>
</tr>
<tr>
<td>132</td>
<td>Pulley speed of the DDOS with the pre-engagement spring when the accessories are included .</td>
</tr>
<tr>
<td>133</td>
<td>Pulley speed of the modified DDOS with the pre-engagement spring at the start .</td>
</tr>
<tr>
<td>134</td>
<td>Amplitude of the impacts in the DDOS with the pre-engagement spring when the accessories are excluded .</td>
</tr>
<tr>
<td>135</td>
<td>Amplitude of the impacts in the DDOS with the pre-engagement spring when the accessories are excluded .</td>
</tr>
<tr>
<td>136</td>
<td>Amplitude of the impacts in the DDOS with the pre-engagement spring at the start .</td>
</tr>
<tr>
<td>137</td>
<td>Filtering of the current solution when the accessories are excluded .</td>
</tr>
<tr>
<td>138</td>
<td>Filtering of the current solution when the accessories are included .</td>
</tr>
<tr>
<td>139</td>
<td>Filtering ratio of the second solution at 1000 rpm with accessories excluded</td>
</tr>
<tr>
<td>FIGURE</td>
<td>PAGE</td>
</tr>
<tr>
<td>--------</td>
<td>------</td>
</tr>
<tr>
<td>140</td>
<td>Filtering ratio of the second solution at 2000 rpm with accessories excluded</td>
</tr>
<tr>
<td>141</td>
<td>Filtering ratio of the second solution at 3000 rpm with accessories excluded</td>
</tr>
<tr>
<td>142</td>
<td>Filtering ratio of the second solution at 1000 rpm with accessories included</td>
</tr>
<tr>
<td>143</td>
<td>Filtering ratio of the second solution at 2000 rpm with accessories included</td>
</tr>
<tr>
<td>144</td>
<td>Filtering ratio of the second solution at 3000 rpm with accessories included</td>
</tr>
<tr>
<td>145</td>
<td>Influence of the clearing G5 on the filtering during the run up with accessories off</td>
</tr>
<tr>
<td>146</td>
<td>Influence of the clearing G5 on the filtering during the run up with accessories on</td>
</tr>
<tr>
<td>147</td>
<td>Influence of the clearing G5 on the filtering during the start and stop of the engine</td>
</tr>
<tr>
<td>148</td>
<td>Zoom in on the influence of the clearance G5 at the start</td>
</tr>
<tr>
<td>149</td>
<td>Influence of the clearance G5 on the impact between pulley and the main spring during the run up with accessories off</td>
</tr>
<tr>
<td>150</td>
<td>Influence of the clearance G5 on the impact between pulley and the main spring during the run up with accessories on</td>
</tr>
<tr>
<td>151</td>
<td>Influence of the clearance G5 on the impact between pulley and the main spring during the start and stop of the engine</td>
</tr>
<tr>
<td>152</td>
<td>Influence of the clearance G5 on the impact between pulley and the secondary spring $k_{se2}$ during the start and stop of the engine</td>
</tr>
</tbody>
</table>
SUMMARY

The object of this thesis is to investigate the mechanical power requirement to the front end accessory belt drive system.

A mathematical model was developed to simulate the dynamic behavior of the system under normal operating conditions. The prototype model was built based on a belt drive system that is currently still being tested.

The purpose of this thesis is to assess the effects of the geometric design variables on the average performance of the front end belt drive system and how they contribute to the optimization of the current design.

The model was validated using data collected at an industrial company which designs and manufactures belts, tensioners and pulleys for the automotive industry.

Once the structure of the whole system was identified, it was schematized and studied using a discrete-system approach.

Finally, a sensitivity analysis was performed and potential solutions are sought to enhance the design and performance are proposed.
CHAPTER 1

INTRODUCTION

The focus of this research is to investigate a crankshaft decoupler mounted in an automotive front end accessory drive.

![Figure 1. Front end accessory drive](image)

In the automotive front end accessory drive, which will be referred to as FEAD, the crankshaft supplies, through a pulley, power to a front end belt which drives many critical parts including: the alternator, which supplies electrical power to the vehicle, the water pump,
which circulates the coolant into the engine, the air conditioning compressor, which circulates
the refrigerant in air conditioning system and other accessories. The layout of an automotive
FEAD depend on the engine mounts, safety and performance. A sample of automotive FEAD
is shown in Figure 1.

The motion of the crankshaft is not regular but it is made of continuous impulses, which are
given by the combustion and which cause high rotational accelerations(1). These irregularities
are transmitted to the front end, and, as a consequence, it affects the behavior of the whole
subsystem.

First, all the accessory pulleys are subjected to the high rotational accelerations as well as
the belt. Such irregular motions of each component involved in the front end transmits excessive
noise and vibration to other vehicle structures, to the vehicle occupants, and may also promote
the fatigue and failure of system components.

An other drawback is the belt slip. The continuous accelerations of the shaft, transmitted
to the pulley, create a sudden reversal of the belt tension which might make it slip on the pulley.
To avoid such a problem, the FEAD is provided with some tensioners that exert a pre load on
the belt.

The presence of the tensioners allow the belt not to slip. However, an increase of pre load
on the belt leads to an increase in friction between the pulley and the belt and higher energy
dissipation.
These effects are critical especially at the start, when the crankshaft pass through its first natural frequency which is lower than 800rpm. Therefore, considering that nowadays almost all the vehicles are equipped with a start and stop system, these issues are very common.

Belt Drive Systems in general have been subjected to many studies and many publications. At the beginning of their development, the attention was focused on the belt-pulley contact. These former studies led to the development of the creep theory (2).

Such theory was adequate to describe the behavior of flat belts, but it became obsolete as the multi-ribbed belts became widely adopted in automotive applications.

Thus, new approaches have been proposed. One of them was introduced by Firbank (3) who evidenced that the shear deformation along the radial direction in the arc of adhesion gives the determining drive behavior.

The shear deflection was studied by Childs and Parker (4) who assumed that the power loss in flat and V belt was due to the shear deflection itself.

It is very important to note that Gerbert in (5), (6), (7), (8), (9) studied extensively the pressure distribution in belts.

Similar studies were done by Hanson in (10) who demonstrated how the finite element method is reliable to compute the pressure distribution between belt and pulley.

Recently, Bechtel et al. reviewed the creep theory by introducing the inertial effect of the wound (11) The other important effect, in Belt Drive Systems, is the dynamic excitation due to transmission, which is analyzed in (12), (13), (14).
Although many studies on Belt Drive Systems have been published, the attention was never focused on the internal structure of the pulleys of the front end layouts.

This last point is interesting since a modern solution to reduce the vibration, and the problem linked to the transmission in general, is to interpose a decoupler between the pulley and the corresponding shaft.

The most simple decoupler adopted in serpentine belt drive systems is the alternator decoupler (15).

This kind of device allows the alternator to rotate at different speed from the alternator hub, hence it allows to decouple the alternator from the accessory drive system.

The advantage is based on the fact that effective inertia of the alternator is by far the largest in a typical accessory drive system. For this reason, if the apparent inertia can be reduced, the dynamic tension fluctuation can also be greatly reduced. Hence, the negative effects of speed fluctuations are reduced.

This leads to the idea to interpose a decoupler between the shaft and the pulley, which could decouple the crankshaft, in order to filter as much as possible the irregularities transmitted to the accessories.

The decoupler, must be designed to allow the output pulley, which is the pulley usually linked to the drive shaft, to operate at higher speed or ”overrun” the crankshaft as it speeds up and slows down. This causes the pulley to be isolated from the torsional vibration due to the oscillations of the crankshaft.
A possible solution is the one designed and patented by Litens, called crankshaft decoupler(16).

The decoupler assembly includes, among others, a drive hub configured to be fixedly secured to the crankshaft, a pulley which is rotatably coupled to the hub, two shell springs which are operatively coupled between the pulley and the hub and a clutch element, which is disposed adjacent the inner clutch surface of the output pulley.

The clutch element is a coil spring having a plurality of coils extending helically. The two ends of the clutch element are called proximal end and distal end. The coil spring is supported by the radial rim element, such that the rim element of each spring shell mates with the contour of the helical coils. Furthermore, the proximal end is fixedly held in a slot obtained in the lower spring shell. The coils should have a rectangular section such that the grip secured is higher.

A rotational acceleration of the drive hub causes the coils to expand radially outwardly to frictionally couple the drive hub and the output pulley.

Conversely, as the drive hub decelerates the coils spring contract radially inwardly. In this way the grip between the coils and the inner clutch surface is released and the output pulley is enabled to overrun around the drive hub.

The solution is shown in Figure 2.

Another possible solution to obtain the decoupler assembly is given by Dayco Europe s.r.l. The decoupler is referred to as Damping Decoupling Over-running System, and it will be referred to as DDOS.
The system has both the function of torsional vibration damper and decoupler assembly.

While the damping behavior does not present particular problem, the decoupling behavior is still under development.

The main objective of this work was to create a mathematical model able to simulate its behavior in different working conditions and to use such a model to solve the issues the device still has.

Several studies on the dynamic of the system were carried out, using a software called Simdrive. The software allow the user to "build" a front end accessory drive and to simulate its behavior. The model made it possible for us to study the behavior of the whole front end.
However the software does not provide any information about the device itself and its internal components.

Therefore, starting with the geometry of the DDOS I studied The behavior and analyzed the static characteristic of the parameters which affect it and how. Secondly, a dynamic study of the behavior of all the internal components was made possible to write the equations of motion of all the parts contained in the DDOS, and finally, through the equations of motion, a numerical model helped us to reproduce the simulated behavior of the system.

The thesis is divided into seven chapters.

The first chapter describes the device, its working principle and its components.

The second chapter is dedicated to the static characteristic. In particular, in this part there is presented a method to obtain the static characteristic of the device numerically.

The next chapter is focused on the dynamic behavior of the device. Hence there are obtained the equation of motion and the model.

Then, a whole chapter will describe how the model is validated.

Once the model was verified, it was performed the Sensitivity Analysis to observe which are the parameters which affect the behavior of the DDOS and in which way.

Finally, considering the results from the analysis, a chapter was dedicated to the design review, where two feasible proposals are analyzed.
CHAPTER 2

DESIGN AND WORKING PRINCIPLE OF THE CRANKSHAFT DECOUPLER

This chapter provides a detailed description of the DDOS components, with a particular emphasis on the function of the overall system. The first step is devoted to develop a schematization of the actual structure in order to derive the equations of motion. The chapter is divided into three parts. First an overview is presented of current state of the system and then all the DDOS components are described in detail. All the geometrical data collected is given at the end of the chapter.

2.1 Design and working principle

An outlook and a section view of the system are shown in Figure 3.

The hub, which referring to Figure 3 is the green part, is fixedly secured to a drive shaft.

The pulley, which is the yellow component in Figure 3, is rotatably coupled to the drive hub and drivingly engaged with the belt.

A spiral spring is fixedly secured to the hub as well as two helical springs which are seated in the proper slots of the hub.

The spiral spring is radially compressed during the assembly and its outer surface is in contact with the inner surface of the pulley through three sliding pads.
Furthermore, there is another element of dry friction which cannot be seen from Figure 3. That is the friction due to the contact between the journal bearing and the adjacent internal surface of the pulley.

As the crankshaft speeds up, and once the friction is won, the edge of the spiral spring moves toward the cog of the pulley until they get engaged. Then the spiral spring transfers torque to the pulley according to the static characteristic of the device.

Conversely, as the crankshaft slows down, the spring first release the elastic energy stored, then, when the energy is enough to overcome the frictions, the edge of the spring gets far from the cog and allows the pulley to overrun the drive hub.

The two helical springs limits the relative displacement in both the directions.

The frictions, which affect the system, have both positive and negative aspects.
The positive point is the dissipation of kinetic energy, which itself has two advantages. First, it prevents the springs from failing due to a high impact shock. Second, since the shock due to the impact is lower the noise created by the impact is lower also. This last aspect, might seem not to be important, and this might be true considering just the life of the system, but the comfort of the final user is one of the main guidelines of all the car manufacturers, and noise is not comfortable at all.

The drawback of the dry friction is that, it couples pulley and hub. This means that it must be small enough to allow the pulley to overrun.

2.2 Description of components

The exploded view of Figure 4 shows all the components of the DDOS, which will be described later in more details.

TVD, in Figure 4, stands for Torsional Vibration Damper. It has the function to damp the torsional vibration which affect the crankshaft. A better view of TVD is shown in Figure 5. The decoupler is shown in Figure 6.

2.2.1 Spiral spring

The spiral spring, also called main spring, is seated in the main spring housing of the hub. It has a spring stiffness coefficient \( k_{ms} \). On its external surface, there are three holes where the sliding pads are fixed.

The free edge of the main spring gets in contact with the cog of the pulley.
In fact, the part of the spring included between the first and the last sliding pad does not behave like a spring but rather as a mass. Hence it makes sense to consider the last part of the spring as an equivalent inertia. Let $I^{*}_{ms}$ be the equivalent inertia of the main spring.

This can be obtained through the energy equation where $m^{*}_{ms}$ is the mass of the spring included between the three sliding pads. This is given by

$$T_{ms} = \frac{1}{2} \cdot m^{*}_{ms} \cdot \dot{x}^{2}_{ms}$$

(2.1)
Since the sliding pads, which are $s_{sp}$ thick, are in contact with the internal surface of the pulley, $R_{pi}$, the external surface of that part of the mass is parallel to the that surface. Furthermore the main spring has a constant section $s$ width.

Hence it is possible to assume, without making a big mistake, that the mass $m_{ms}$ moves at a distance $R_{ms}$ given by:

$$R_{ms} = R_{pi} - s_{sp} - \frac{s}{2} \quad (2.2)$$

Then the velocity can be expressed as:

$$\dot{x}_{ms} = R_{ms} \dot{\theta}_{ms} \quad (2.3)$$
Thus the kinetic energy becomes:

\[ T_{ms} = \frac{1}{2} \cdot m_{ms}^* \cdot (R_{ms} \dot{\theta}_{ms})^2 = \frac{1}{2} \cdot I_{ms}^* \cdot \dot{\theta}_{ms}^2 \]  

(2.4)

Hence the equivalent inertia of the main spring is found to be:

\[ I_{ms}^* = m_{ms}^* R_{ms}^2 \]  

(2.5)

The equivalent inertia of the main spring \( I_{ms}^* \) is the inertia of the sole mass \( m_{ms}^* \). The actual Inertial of the whole spiral spring is \( I_{ms} \).

2.2.2 Sliding pads

The sliding pads are 3 pads made of PA46mod(polymer compound). They slide on the inner surface of the pulley, creating dry friction. Each sliding pad is \( s_{sp} \) thick.
This dry friction contributes is indicated as T1 and it is proportional to $f_{sp}$, which is the friction coefficient between the pad and the surface, the elastic force normal to the internal surface of the pulley $F_n$ and the radius of the internal surface of the pulley $R_{pi}$.

The force normal to the surface is given by:

$$F_n = k_r \cdot \Delta r_i$$  \hspace{1cm} (2.6)

where $k_{tan}$ is the tangential stiffness of the main spring and $\Delta r_i$ is the initial load due to the radial compression of the main spring which is fitted inside the pulley.

However the value of the friction T1 is known from the static test.

### 2.2.3 Helical springs

The helical springs, Figure 7, are also called secondary springs. Their name is due to the fact that when the hub is moving forward to the pulley, the latter one gets in contact first with the spiral spring and then with these ones. There are two helical springs, each one has stiffness $k_{ss1}$ and $k_{ss2}$ respectively. The maximum deformations are $\Delta \theta_{ss1_{max}}$ and $\Delta \theta_{ss2_{max}}$. The axis of these springs is far $R_{ss}$ from the rotation axis. The potential energy stored in each helical spring is:

$$U = \frac{1}{2} \cdot k_{ss} \cdot x_{ss}^2$$  \hspace{1cm} (2.7)

where the displacement $x_{ss}$ might be expressed as:

$$x_{ss} = R_{ss} \cdot \theta_{ph}$$  \hspace{1cm} (2.8)
Then the potential energy becomes:

\[ U = \frac{1}{2} \cdot k_{ss} \cdot (R_{ss} \cdot \theta_{ph})^2 \]  

(2.9)

hence:

\[ U = \frac{1}{2} \cdot k^*_{ss} \cdot \theta^2_{ph} \]  

(2.10)

where the equivalent spring \( k^*_{ss} \) is:

\[ k^*_{ss} = k_{ss} \cdot R^2_{ss} \]  

(2.11)

They are fixed in the auxiliary spring housing, and they have a pin head on both ends.

Figure 7. Zoom on the secondary spring group
2.2.4 Pin head

The pin head, Figure 7, is located on any end of the secondary springs.

From a practical point of view its main function is to make the impact between pulley and secondary springs more uniform.

Although its inertia is negligible, it will be important in the dynamic analysis. The mass of the pin head is $m_{ph}$, and its center of mass is distant $R_{ss}$ to the rotation axis.

As done before for the main spring, it is possible to obtain the equivalent inertia of each pin head $I_{ph}$ through an energetic approach.

The kinetic energy of the pin head is:

$$T_{ph} = \frac{1}{2} \cdot m_{ph} \cdot \dot{x}_{ph}^2$$  \hspace{1cm} (2.12)

The velocity of the pin head is expressed by:

$$\dot{x}_{ph} = R_{ph} \cdot \dot{\theta}_{ph}$$  \hspace{1cm} (2.13)

Finally the kinetic energy becomes:

$$T_{ph} = \frac{1}{2} \cdot m_{ph} \cdot (R_{ph} \dot{\theta}_{ph})^2 = \frac{1}{2} \cdot I_{ph} \cdot \dot{\theta}_{ph}^2$$  \hspace{1cm} (2.14)
Hence the equivalent inertia of the pin head is:

\[ I_{ph} = m_{ph} R_{ph}^2 \]  \hfill (2.15)

2.2.5 Auxiliary spring housing

This housing, which can be made of either steel or plastic, is fitted in a slot of the hub, Figure 7. Its aim is to guide the spring movement. The dry friction between the secondary spring and its housing is negligible and it will be neglected.

2.2.6 Journal bearing

It is a cylindrical bearing which supports a cylindrical shaft, which in this case is the Inertia ring.

It is the simplest, lightest and least expensive type of bearing. Although these characteristics, it has a high load-carrying capability. It is interposed between the Inertia ring and the pulley. During the working, it generates the dry friction \( T_2 \), which is proportional to the radius of the bearing \( r_b \), the friction coefficient \( f_b \), and the radial load \( H_l \).

The radial load is the reaction force exerted by the inertia ring on the journal bearing and by the journal bearing on the pulley.

According to Figure 8, the torque provided by the pulley to the belt is proportional the difference between the two tensions \( F_1 \) and \( F_2 \):

\[ T = (F_1 - F_2) \cdot R \]  \hfill (2.16)
where $R$ is the radius of the pulley.

The sum of the two forces is the radial load, $H_l$:

$$
\vec{H}_l = \vec{F}_1 + \vec{F}_2
$$

(2.17)
According to Figure 9, the Radial load can be expressed by the Law of cosines as:

\[ H_l = \sqrt{F_1^2 + F_2^2 - 2 \cdot F_1 \cdot F_2 \cdot \cos(\alpha)} \] (2.18)

where \( \alpha \) is the wrap angle.

The dry friction \( T_2 \) is given by:

\[ T_2 = r_b \cdot f_b \cdot H_l \] (2.19)

hence:

\[ T_2 = r_b \cdot f_b \cdot \sqrt{F_1^2 + F_2^2 - 2 \cdot F_1 \cdot F_2 \cdot \cos(\alpha)} \] (2.20)

where \( F_2 \) is the tension imposed by the tensioner and \( F_1 \) is the tension which varies according to the external load.

### 2.2.7 Rubber ring

It is a rubber ring interposed between the hub and the inertia ring. Its internal surface is glued to the hub, while the external one is stuck to the inertia ring.

It is characterized by a stiffness \( k_{rr} \).

### 2.2.8 Inertia ring

It is a ring made of steel. It is fixed to the rubber ring. These two elements together realize the torsional dynamic damper, whose function is to reduce the amplitude of the tor-
sional vibration when the system gets in resonance. The inertia, $I_r$, completely define the component.

It is important to remind that in this thesis the aspect of the dynamic damper it was not really treated since the attention was focused mostly on the issues of the decoupler.

2.2.9 Pulley

The pulley has a particular geometry. It has an annular outer track defined between a pair of spaced apart, raised and parallel rims that seats the belt therein.

The inner surface $R_{pi}$ is hardened to prevent it from wearing due to the friction $T_1$.

On that surface there are a cog and a slope, as shown in Figure 10.

Figure 10. Zoom on the inner surface of the pulley
The cog gets in contact with the free edge of the main spring. When they are engaged the spring transfer torque from the crankshaft to the pulley.

The slope, which is seated before the cog in the direction of motion, radially compresses the spiral spring. In this way, the force exerted by the spring on the surface of the pulley increases as well as the dry friction.

The pulley has two appendices whose edges get in contact with the head pin of the secondary springs. These appendices, shown in Figure 11 as the yellow parts, move inside a circular slot obtained in the hub.

The pulley has its own inertia \( I_P \).

Actually to perform the right dynamic simulation of the DDOS the inertia of all the accessories, which are engaged to the pulley through the belt, are important and must considered.

Hence, it is necessary to define an equivalent inertia \( I_P^* \) of the system rather than the inertia of the pulley alone.

To find the equivalent inertia we use the energy method the total kinetic energy is:

\[
T_{\text{tot}} = \frac{1}{2} I_P \omega_P^2 + \frac{1}{2} I_{A/C} \omega_{A/C}^2 + \frac{1}{2} I_{Alt} \omega_{Alt}^2 + \frac{1}{2} I_{wp} \omega_{wp}^2
\]  

(2.21)

The rotational speeds of all the gears are linked by the following relations:

\[
v_t = \omega_P R_P = \omega_{A/C} R_{A/C} = \omega_{Alt} R_{Alt} = \omega_{wp} R_{wp}
\]  

(2.22)
Hence the gear ratios, which are defined as the ratio between the inlet rotational speed and the output rotational speed, are:

\[ i_{A/C} = \frac{R_{A/C}}{R_P} \]  
\[ i_{Alt} = \frac{R_{Alt}}{R_P} \]  
\[ i_{wp} = \frac{R_{wp}}{R_P} \]

Then the kinetic energy in Equation 2.21 can be expressed as:

\[ T_{tot} = \frac{1}{2} I_P \omega_P^2 + \frac{1}{2} I_{A/C} i_{A/C}^2 \omega_P^2 + \frac{1}{2} I_{Alt} i_{Alt}^2 \omega_P^2 + \frac{1}{2} I_{wp} i_{wp}^2 \omega_P^2 \]  

Finally assuming that:

\[ T_{tot} = \frac{1}{2} I_P \omega_P^2 \]  

the equivalent inertia is:

\[ I_P^* = I_P + i_{A/C}^2 I_{A/C} + i_{Alt}^2 I_{Alt} + i_{wp}^2 I_{wp} \]

2.2.10 Hub

The hub is the part of the DDOS which is fixed to the crankshaft. The hub has a main spring seat where the spiral spring is fitted in, and two secondary spring slots where the auxiliary spring housings are fitted in.
There is even a circular slot where the two appendices of the pulley run until they get in contact with the pin head. It is $r_{ss}$ far from the rotation axis. The hub has its own inertia $I_H$.

### 2.3 Internal clearances

Important features of the device, which must be identified and quantified to describe the behavior of the DDOS, are the internal clearances.

The internal clearances make the system highly non linear, hence to study the system as it would be linear, they will be taken into account as they would be generalized forces, each one with its direction, verse and amplitude which depend on the relative positions of each component of the whole system. This point will be fully treated in the equation of motion chapter.

However, since the internal clearances depend on the geometry of the components and how they are fixed together, in this chapter they will be quantified.

Referring to Figure 11 it is possible to identify five internal clearances:

- $G_1$, which is the clearance between the mass $m_{ms}$ and the cog of the pulley;
- $G_2$, which is the clearance to cover before the appendices of the pulley hit the pin heads 1, while the hub is moving forward to the pulley;
- $G_3$, which is the clearance to cover before the appendices of the pulley hit the pin heads 1, while the pulley is moving forward to the hub;
- $G_4$, which is the clearance to cover before the appendices of the pulley hit the pin heads 2, while the hub is moving forward to the pulley;
• **G5**, which is the clearance to cover before the appendices of the pulley hit the pin heads 2, while the pulley is moving forward to the hub.

Actually in Figure 11 shown just two internal clearances, G2 and G3. The point to make is in this particular model the values of the clearances G4 and G5 are the same as those of G2 and G3 respectively, and the clearance G1 is null because the main spring is in contact with the cog of the pulley.

Furthermore in picture Figure 11 it is possible to distinguish several parts:

• the pulley is the part colored in yellow as well as the appendices which hit the pin heads;

• the pin heads are grey;
• the auxiliary springs are blue;

• the green represents the slot of the hub where the appendices run;

• the black part is the main spring.

Clearly, the clearances change according to the position, hence it is sufficient to measure the clearances $G_1$, $G_2$, $G_3$, $G_4$ and $G_5$ from any position of the system. In this way the geometry of the device is fully known. The internal clearances are summarized in Table I.

<table>
<thead>
<tr>
<th>$G$</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>$G_1$</td>
<td>0 deg</td>
</tr>
<tr>
<td>$G_2$</td>
<td>36.5 deg</td>
</tr>
<tr>
<td>$G_3$</td>
<td>55.6 deg</td>
</tr>
<tr>
<td>$G_4$</td>
<td>36.5 deg</td>
</tr>
<tr>
<td>$G_5$</td>
<td>55.6 deg</td>
</tr>
</tbody>
</table>

2.4 Physical and geometrical data

All the data of the components are collected in Table II.
### TABLE II

**DATA COMPONENT**

<table>
<thead>
<tr>
<th>Component</th>
<th>Features</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Main spring</strong></td>
<td>$k_{ms}$</td>
<td>114.6 $Nm/deg$</td>
</tr>
<tr>
<td></td>
<td>$m_{ms}^*$</td>
<td>0.168 $Kg$</td>
</tr>
<tr>
<td></td>
<td>$R_{ms}$</td>
<td>0.04725 $m$</td>
</tr>
<tr>
<td></td>
<td>$I_{ms}$</td>
<td>0.0003751 $kg \cdot m^2$</td>
</tr>
<tr>
<td></td>
<td>$k_T$</td>
<td>235000 $N/m$</td>
</tr>
<tr>
<td></td>
<td>$c_{ms}$</td>
<td>0 $Nms/deg$</td>
</tr>
<tr>
<td><strong>Sliding pad</strong></td>
<td>$T_1$</td>
<td>15 $Nm$</td>
</tr>
<tr>
<td><strong>Secondary spring 1</strong></td>
<td>$k_{ss1}$</td>
<td>64540 $N/m$</td>
</tr>
<tr>
<td></td>
<td>$R_{ss}$</td>
<td>0.0385 $m$</td>
</tr>
<tr>
<td></td>
<td>$k_{ss1}^*$</td>
<td>95.66 $Nm/deg$</td>
</tr>
<tr>
<td></td>
<td>$c_{ss1}$</td>
<td>0 $Nms/deg$</td>
</tr>
<tr>
<td><strong>Pin head 1</strong></td>
<td>$m_{ph1}$</td>
<td>0.00205 $kg$</td>
</tr>
<tr>
<td></td>
<td>$I_{ph1}$</td>
<td>3.038 $\cdot 10^{-6}$ $kg \cdot m^2$</td>
</tr>
<tr>
<td><strong>Secondary spring 2</strong></td>
<td>$k_{ss2}$</td>
<td>64540 $N/m$</td>
</tr>
<tr>
<td></td>
<td>$R_{ss}$</td>
<td>0.0385 $m$</td>
</tr>
<tr>
<td></td>
<td>$k_{ss2}^*$</td>
<td>95.66 $Nm/deg$</td>
</tr>
<tr>
<td></td>
<td>$c_{ss2}$</td>
<td>0 $Nms/deg$</td>
</tr>
<tr>
<td><strong>Pin head 2</strong></td>
<td>$m_{ph2}$</td>
<td>0.00205 $kg$</td>
</tr>
<tr>
<td></td>
<td>$I_{ph2}$</td>
<td>3.038 $\cdot 10^{-6}$ $kg \cdot m^2$</td>
</tr>
<tr>
<td><strong>Journal bearing</strong></td>
<td>$r_b$</td>
<td>0.0675 $m$</td>
</tr>
<tr>
<td></td>
<td>$f_b$</td>
<td>0.15</td>
</tr>
<tr>
<td><strong>Pulley</strong></td>
<td>$R_{pi}$</td>
<td>0.05125 $m$</td>
</tr>
<tr>
<td></td>
<td>$R_{pe}$</td>
<td>0.0725 $m$</td>
</tr>
<tr>
<td></td>
<td>$I_P$</td>
<td>0.003077 $kg \cdot m^2$</td>
</tr>
<tr>
<td><strong>A/C gear</strong></td>
<td>$I_{A/C}$</td>
<td>0 $kg \cdot m^2$</td>
</tr>
<tr>
<td></td>
<td>$R_{A/C}$</td>
<td>0.055 $m$</td>
</tr>
<tr>
<td><strong>W/p gear</strong></td>
<td>$I_{W/p}$</td>
<td>0.001546 $kg \cdot m^2$</td>
</tr>
<tr>
<td></td>
<td>$R_{W/p}$</td>
<td>0.0525 $m$</td>
</tr>
<tr>
<td><strong>Alt gear</strong></td>
<td>$I_{Alt}$</td>
<td>0.00412 $kg \cdot m^2$</td>
</tr>
<tr>
<td></td>
<td>$R_{Alt}$</td>
<td>0.025 $m$</td>
</tr>
<tr>
<td><strong>Hub</strong></td>
<td>$I_H$</td>
<td>$kg \cdot m^2$</td>
</tr>
</tbody>
</table>
CHAPTER 3

STATIC CHARACTERISTIC

The static characteristic of the device is the characteristic Torque versus relative displacement. It provides information about torque transmitted, frictions, internal clearances, stiffness, maximum displacements. There are several ways to obtain the static characteristic. It can be obtained experimentally through a static test on a prototype. Conversely, it can be represented numerically. Starting with an equivalent scheme of the device, it is possible to write the equation of motion and then plot the curve displacement versus torque. In this chapter it is explained a numerical method is used to plot the static characteristic of the system. First of all, the attention will be focused on a different system whose behavior is described by a simple equation of motion. Once the equation of motion is know the characteristic displacement force is obtained. This basic system will be used to build an equivalent structure of the DDOS which allows to describe the behavior in static condition, i.e. the equivalent structure of the DDOS will be assembled putting several basic system in parallel.

3.1 Basic module

3.1.1 Structure

Consider the system shown in figure Figure 12.

It is characterized by the following parameters:

- stiffness k;
Figure 12. Basic module

- dry friction $T$;
- clearances $a, b, c$ and $d$.

This system is the "basic module" which will be used to assemble the whole structure of the DDOS.

3.1.2 Characteristic

Assuming $x$ in Figure 12 to be the displacement and $F$ is the force necessary to obtain that displacement, it is possible to write the characteristic of the system as force versus $x$.

From the equilibrium position $x = 0$, assuming that the system is moving forward, the spring compress until the elastic force stored by the spring is enough to win the static friction, $T$(figure Figure 13).
Figure 13. Trend of the force at the beginning of motion

Once the system lost its grip, it slips until the internal clearance $d$ is completely gone, Figure 14.

Figure 14. Force transmitted while the system is slipping

Next, if $x$ is still increasing, the spring starts again to compress, until it reaches its maximum compression. At that point the force transmitted tends to infinity, as shown in Figure 15.
The maximum value of forward displacement is: $x_{\text{max}} = b + d$ and at that value of $x$ the system is in the configuration shown in Figure 16.

From the last position the system starts moving backward, in this way the spring release the elastic force stored before, but when it reach its equilibrium position it starts compress again because of the static Friction. This behavior is shown in Figure 17.
Then when the elastic force stored is enough to win the static friction, the system starts slipping again until the whole internal clearance, $b + d$ is covered. Then the system continues moving forward until the spring reaches its maximum traction length, as shown in Figure 18.

Figure 17. Force transmitted during spring releasing and reloading

Figure 18. Force transmitted during moving backward
In this position, $x = -a - c$, the system is in the configuration illustrated in Figure 19.

![Figure 19. Basic module max tight](image)

Finally, from this position, imposing a displacement toward the top, the spring releases the force it stored during the previous traction, and then it loads again until the system starts slipping. The whole characteristic with the respective positions is shown in Figure 20.

The above characteristic is obtained running on an Excel sheet a Visual Basic function which reads relative displacement, stiffness $k$, friction $T$ and clearances as input, and gives the corresponding force $F$ as output.

Such VB function is listed as annex A.
3.2 Actual system

3.2.1 Structure

Considering that the mass properties do not affect the static characteristic and taking into account each component and its features described in the previous chapter, the schematic of the actual device becomes as shown in Figure 21.

In the picture T1 is the dry friction between the sliding pads and the internal surface of the pulley, T2 is the dry friction between pulley and the journal bearing, \( k_{ma} \) is the stiffness of the main spring and \( k_{sa} \) is the spring of the secondary spring. G1, G2, G3, G4 and G5 represent the internal clearances which were discussed in the first chapter.

According to Figure 21, the whole system is made up of four different subsystems each one bonded by a different colored dashed line.
Each subsystem can be obtained from the basic module studied before setting the proper parameters.

The first element in Figure 21, which is bounded by a red dashed line, can be obtained starting from the basic cell of Figure 12, setting the data collected in Table III.

**TABLE III**

**FEATURES OF THE FIRST MODULE**

<p>| | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>a</td>
<td>0</td>
</tr>
<tr>
<td>b</td>
<td>0</td>
</tr>
<tr>
<td>c</td>
<td>∞</td>
</tr>
<tr>
<td>d</td>
<td>∞</td>
</tr>
<tr>
<td>k</td>
<td>∞</td>
</tr>
<tr>
<td>T</td>
<td>T2</td>
</tr>
</tbody>
</table>
The second one is more complex than the previous since it has both spring and dry friction. However it is still possible to simulate the behavior of this element using the same approach used before. The data of this module are collected in Table IV.

\begin{table}[h]
\centering
\caption{Features of the second module}
\begin{tabular}{|c|c|}
\hline
\textbf{a} & 0 \\
\textbf{b} & \(G_3\) + max deformation secondary spring \\
\textbf{c} & \(G_2\) + max deformation secondary spring \\
\textbf{d} & \(G_1\) \\
\textbf{k} & \(k_{ms}\) \\
\textbf{T} & \(T_1\) \\
\hline
\end{tabular}
\end{table}

Finally the third and the fourth modules are obtained in the same way, setting for both the elements the data collected in Table V.

Then, setting the data properly it follows the characteristic shown in Figure 22.

### 3.3 Experimental curve

The static characteristic can be obtained experimentally also. The DDOS is mounted on a properly test stand, which rotate the pulley while the hub is stuck. During the relative motion it measures the Torque necessary to rotate the pulley. When the test is over it provides a curve like the one shown in Figure 23.

The experimental curve can be used to validate the numerical method.
TABLE V

FEATURES OF THE LAST TWO MODULES

<p>| | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>a</td>
<td>max deformation spring</td>
</tr>
<tr>
<td>b</td>
<td>max deformation spring</td>
</tr>
<tr>
<td>c</td>
<td>$G_2$</td>
</tr>
<tr>
<td>d</td>
<td>$G_3$</td>
</tr>
<tr>
<td>k</td>
<td>$k_{ss}$</td>
</tr>
<tr>
<td>T</td>
<td>0</td>
</tr>
</tbody>
</table>

Figure 22. Characteristic of the DDOS

Making a comparison between the curves Figure 23 and Figure 22, the curves do not match exactly each other, especially in the overruning zone.

This is mainly due to the surface defects of the inner surface of the pulley.

However, the two curves are close enough to consider the numerical solution reliable.
3.4 Radial Load

The static characteristic is not affected by the Radial Load.

However, it is interesting to evaluate its effect which occurs during the working condition.

The radial load described in the first chapter affects the static characteristic since it changes the value of the dry friction $T_2$. Recall Equation 3.1.

$$T_2 = r_b \cdot f_b \cdot \sqrt{F_1^2 + F_2^2 - 2 \cdot F_1 \cdot F_2 \cdot \cos(\alpha)}$$ (3.1)
The total torque exerted by the pulley is given by the sum of all the forces exerted by each basic module. Let $T_a$, $T_b$, $T_c$ and $T_d$ be the torque exerted by the four modules respectively, the total torque provided by the pulley is given by the sum of all these contributes:

$$T = T_a + T_b + T_c + T_d$$  \hspace{1cm} (3.2)

According to what it was said, the contributes $T_a$ depends on the total torque $T$ itself.

$$T = f(T_b, T_c, T_d, T_a(T, F_2, R, r))$$  \hspace{1cm} (3.3)

To completely define the static characteristic the radial load must be taken into account also. There are two ways to do that, analytically and numerically. The first method consists on handle analytically the expression of the torque to obtain an expression in the form:

$$T = f(T_b, T_c, T_d, F_2, R, r)$$  \hspace{1cm} (3.4)

The other method consists on assuming an initial value for $F_2$, find the total torque $T$, compute the radial load $H_l$, compute the new value of friction torque $F_2'$ and finally compute the new transmitted torque $T'$. After several averages the series converges to a value which does not depends on the initial guess.
3.4.1 Analytical solution

In the Equation 3.2, the quantity $T_b + T_c + T_d$ depends only on the relative position of the pulley, hence:

$$T = D(\Delta \theta) + T_a$$  \hspace{1cm} (3.5)

replacing $T_a$ with Equation 3.1, it follows:

$$T = D + f \cdot r \cdot \sqrt{\left(\frac{T}{R} + F_2\right)^2 + F_2^2 - 2 \cdot \left(\frac{T}{R} + F_2\right) \cdot F_2 \cdot \cos(\alpha)}$$  \hspace{1cm} (3.6)

then:

$$\frac{(T - D)}{f \cdot r} = \sqrt{\left(\frac{T}{R} + F_2\right)^2 + F_2^2 - 2 \cdot \left(\frac{T}{R} + F_2\right) \cdot F_2 \cdot \cos(\alpha)}$$  \hspace{1cm} (3.7)

Hence:

$$\frac{(T - D)^2}{f^2 \cdot r^2} = \left(\frac{T}{R} + F_2\right)^2 + F_2^2 - 2 \cdot \left(\frac{T}{R} + F_2\right) \cdot F_2 \cdot \cos(\alpha)$$  \hspace{1cm} (3.8)

The above expression becomes:

$$\frac{T^2}{f^2 \cdot r^2} + \frac{D^2}{f^2 \cdot r^2} - 2 \cdot \frac{T \cdot D}{f^2 \cdot r^2} = \frac{T^2}{R^2} + 2 \cdot \frac{T \cdot F_2}{R} \cdot (1 - \cos(\alpha)) + 2 \cdot \frac{T \cdot F_2}{R} \cdot (1 - \cos(\alpha))$$  \hspace{1cm} (3.9)

Finally:

$$T^2(1 - \frac{f^2 r^2}{R^2}) + T(-2D - \frac{2F_2 f^2 r^2 \cdot (1 - \cos(\alpha))}{R}) + (D^2 - 2F_2 f^2 r^2 \cdot (1 - \cos(\alpha))) = 0$$  \hspace{1cm} (3.10)
and the solution is given by:

\[ T = \frac{-b \pm \sqrt{b^2 - 4ac}}{2a} \]  

(3.11)

where:

\[ a = (1 - f^2 r^2 \frac{R^2}{R^2}) \]  

(3.12)

\[ b = (-2D - 2F f^2 f^2 \cdot (1 - \cos(\alpha)) \frac{R}{R}) \]  

(3.13)

\[ c = (D^2 - 2F f^2 f^2 \cdot (1 - \cos(\alpha))) \]  

(3.14)

When \( \alpha = 180 \), \( F_1 \) and \( F_2 \) are parallel, hence:

\[ H_l = \frac{T}{R} + 2F_2 \]  

(3.15)

Assuming:

\[ T = \pm|T_a| + T_b + T_c + T_d \]  

(3.16)

the solution is:

\[ T = \pm f r \left| \frac{T}{R} + 2F_2 \right| + D \]  

(3.17)

hence:

\[ T = \frac{D \pm 2F_2}{1 \mp \frac{f r}{R}} \]  

(3.18)

The same result might be achieved substituting \( \alpha = 180 \) in the Equation 3.10.
3.4.2 Numerical solution

A different approach to take the radial load into account is the following: guess an initial value of $T_a$ and obtain the torque $T$, compute the value of the tension $F_1$ from the torque $T$ through the equation:

$$F_1 = \frac{T}{R} + F_2$$

(3.19)

then compute the radial load from the equation:

$$H_l = \sqrt{F_1^2 + F_2^2 - 2 \cdot F_1 \cdot F_2 \cdot \cos(\alpha)}$$

(3.20)

and compute the new value of $T_a$:

$$T_a = frH_l$$

(3.21)

Once the new value of the friction is known compute the new value of the torque exerted by the pulley. Iterating this procedure several times the torque tends to a value which does not depend on the initial guess.

In Figure 24 are plotted the static characteristics obtained through both the ways.
Figure 24. Actual static characteristic
CHAPTER 4

KINEMATIC AND DYNAMIC ANALYSIS

This chapter is focused on the equations of motion. Starting from the equivalent structure model the equation of motion are written in the configuration space first, and in the state-space next. In this way, such equations are implemented to assess the mechanical characteristics of the system and how the inertia properties affect its dynamic response.

4.1 Equivalent model

According to what was said in the description of components, the inertial properties which cannot be neglected, i.e. the inertial properties which affect the dynamic response of the system, are the inertia of the pulley, the inertia of the hub, the equivalent inertia of each pin head and the equivalent inertia of the main spring.

The equivalent model, considering the inertial properties, becomes the one shown in Figure 25.

The frictions T1 and T2 and the internal clearances make the system of figure Figure 25 high non linear.

Recall that T1 is the friction due to the contact between the pads and the inner surface of the pulley, while T2 is the friction due to the journal bearing which is proportional to the hub load.
The friction is always opposite to the motion, hence $T_1$ and $T_2$ change according to the relative velocity between pulley and main spring and pulley and hub respectively.(18).

The graph of both frictions is shown below in Figure 26.

On the other hand, clearances create a kind of "death zone" where they do not create any force, while, when the gap is covered, they cause contacts.

The effect of such contacts is a force which has almost the same trend of an impulse: it is null in the dead zone which is defined by the clearances and it tends to infinitive when the contact occurs.
The graphs of the force caused by each clearance are shown below.

Since all the forces generated by the non-linearities are known, it is possible to linearize the system replacing them with properly external generalized forces which have exactly the same trend.

The equivalent linear system so obtained is shown in Figure 30.
4.2 Configuration Space

The configuration space is the space of possible positions that a physical system may attain, possibly subject to external constraints.

The equations of motion in the configuration space can be found through the Lagrange Equations (19)

\[
\frac{d}{dt} \left( \frac{\partial T}{\partial \dot{x}_i} \right) - \frac{\partial T}{\partial x_i} + \frac{\partial U}{\partial x_i} + \frac{\partial F}{\partial \dot{x}_i} = Q_i
\]  

(4.1)
where $T$ is the kinetic energy of the whole system, $U$ is the potential energy, $F$ is the Rayleigh function, which is a measure of the power dissipated by non-conservative forces, $x_i$ is the $i$-th lagrangian coordinate and $Q_i$ is the sum of all the external generalized forces relative to the displacement $x_i$.

The lagrangian coordinates of the system are $\theta_p$, $\theta_{ms}$, $\theta_{ph1}$ and $\theta_{ph2}$.

According to Figure 30 the kinetic energy is

$$T = \frac{1}{2} I_p \dot{\theta}_p^2 + \frac{1}{2} I_{ms} \dot{\theta}_{ms}^2 + \frac{1}{2} I_H \dot{\theta}_H^2 + \frac{1}{2} I_{ph1} \dot{\theta}_{ph1}^2 + \frac{1}{2} I_{ph2} \dot{\theta}_{ph2}^2$$

(4.2)
Similarly the potential energy is found to be

\[ U = \frac{1}{2} k_{ms} (\theta_{ms} - \theta_H)^2 + \frac{1}{2} k_{ss1} (\theta_{ph1} - \theta_H)^2 + \frac{1}{2} k_{ss2} (\theta_{ph2} - \theta_H)^2 \]  

(4.3)

The Rayleigh dissipation function for the viscous dampers in Figure 30 is

\[ F = \frac{1}{2} c_{ms} (\dot{\theta}_{ms} - \dot{\theta}_H)^2 + \frac{1}{2} c_{ss1} (\dot{\theta}_{ph1} - \dot{\theta}_H)^2 + \frac{1}{2} c_{ss2} (\dot{\theta}_{ph2} - \dot{\theta}_H)^2 \]  

(4.4)

The expression of the forces \( Q_i \) in the Equation 4.1 can be obtained form the virtual work \( \delta L \) with respect to the lagrangian coordinate \( x_i \)(20). The virtual work generated by the virtual displacement \( \vec{\delta r}_k \) of the force \( \vec{F}_k \) is given by

\[ \delta W_k = \vec{F}_k \cdot \vec{\delta r}_k \]  

(4.5)

By writing the virtual displacement \( \vec{\delta r}_k \) as a function of the lagrangian coordinates \( x_i \)

\[ \vec{\delta r}_k = \frac{\partial \vec{r}_k}{\partial x_1} \delta x_1 + \frac{\partial \vec{r}_k}{\partial x_2} \delta x_2 + \cdots + \frac{\partial \vec{r}_k}{\partial x_n} \delta x_n = \sum_{i=1}^{n} \frac{\partial \vec{r}_k}{\partial x_i} \delta x_i \]  

(4.6)

the virtual work can be expressed as a function of the lagrangian coordinates

\[ \delta W = \sum_{i=1}^{n} \left( \sum_{k=1}^{n} \vec{F}_k \cdot \frac{\partial \vec{r}_k}{\partial x_i} \right) \delta x_i \]  

(4.7)
As stated above, the generalized forces are obtained from the virtual work $\delta W$ with respect to the lagrangian coordinate $x_i$, hence

$$Q_i = \frac{\partial \delta W}{\partial \delta x_i}$$

(4.8)

thus

$$Q_i = \sum_{k=1}^{n} F_k \frac{\partial r_k}{\partial x_i}$$

(4.9)

Referring to the system of Figure 30 the external generalized forces are

$$Q_i = T_2 \frac{\partial \theta_p}{\partial x_i} + T_2 \frac{\partial \theta_H}{\partial x_i} + G_1 \frac{\partial \theta_p}{\partial x_i} + G_1 \frac{\partial \theta_{ms}}{\partial x_i} + T_1 \frac{\partial \theta_p}{\partial x_i} + T_1 \frac{\partial \theta_{ms}}{\partial x_i}$$

$$+ G_2 \frac{\partial \theta_p}{\partial x_i} + G_2 \frac{\partial \theta_{ph1}}{\partial x_i} + G_4 \frac{\partial \theta_p}{\partial x_i} + G_4 \frac{\partial \theta_{ph2}}{\partial x_i} + T_{ext} \frac{\partial \theta_p}{\partial x_i}$$

(4.10)

thus developing the dot products according to the reference frame of Figure 30

$$Q_i = T_2 \frac{\partial \theta_p}{\partial x_i} - T_2 \frac{\partial \theta_H}{\partial x_i} + G_1 \frac{\partial \theta_p}{\partial x_i} - G_1 \frac{\partial \theta_{ms}}{\partial x_i} + T_1 \frac{\partial \theta_p}{\partial x_i} - T_1 \frac{\partial \theta_{ms}}{\partial x_i}$$

$$+ G_2 \frac{\partial \theta_p}{\partial x_i} - G_2 \frac{\partial \theta_{ph1}}{\partial x_i} + G_4 \frac{\partial \theta_p}{\partial x_i} - G_4 \frac{\partial \theta_{ph2}}{\partial x_i} - T_{ext} \frac{\partial \theta_p}{\partial x_i}$$

(4.11)

By performing the derivatives appearing in the Lagrange Equation 4.1 it follows the equation of motion of each lagrangian coordinate is found as follows

$$I_{ms} \ddot{\theta}_{ms} + k_{ms}(\theta_{ms} - \theta_H) + c_{ms}(\dot{\theta}_{ms} - \dot{\theta}_H) = -G_1 - T_1$$

(4.12)

$$I_{ph1} \ddot{\theta}_{ph1} + k_{ss1}(\theta_{ph1} - \theta_H) + c_{ss1}(\dot{\theta}_{ph1} - \dot{\theta}_H) = -G_2$$

(4.13)
\[ I_{ph2} \ddot{\theta}_{ph2} + k_{ss2}(\theta_{ph2} - \theta_H) + c_{ss2}(\dot{\theta}_{ph2} - \dot{\theta}_H) = -G4 \]  
(4.14)

\[ I_p \ddot{\theta}_p + c_{eq}(\dot{\theta}_p - \dot{\theta}_H) = T1 + T2 + G1 + G2 + G4 - T_{ext} \]  
(4.15)

The force \( T_{ext} \) and the rotary speed \( \dot{\theta}_H \) are both input of the system, while the forces \( T1, T2, G1, G2 \) and \( G4 \) are determined by negative feedback, as stated above. However it is convenient to consider them as input as well. In order to make the equation of motion more comprehensible it is useful to put all the input terms on the right-hand side. Hence

\[ I_{ms} \ddot{\theta}_{ms} + k_{ms} \theta_{ms} + c_{ms} \dot{\theta}_{ms} = -G1 - T1 + k_{ms} \theta_H + c_{ms} \dot{\theta}_H \]  
(4.16)

\[ I_{ph1} \ddot{\theta}_{ph1} + k_{ss1} \theta_{ph1} + c_{ss1} \dot{\theta}_{ph1} = -G2 + k_{ss1} \theta_H + c_{ss1} \dot{\theta}_H \]  
(4.17)

\[ I_{ph2} \ddot{\theta}_{ph2} + k_{ss2} \theta_{ph2} + c_{ss2} \dot{\theta}_{ph2} = -G4 + k_{ss2} \theta_H + c_{ss2} \dot{\theta}_H \]  
(4.18)

\[ I_p \ddot{\theta}_p + c_{eq} \dot{\theta}_p = T1 + T2 + G1 + G2 + G4 - T_{ext} + c_{eq} \dot{\theta}_H \]  
(4.19)

Let \( \vec{s} \) be the vector which contains all the lagrangian coordinates of the system

\[
\vec{s} = \begin{pmatrix}
\theta_{ms} \\
\theta_{ph1} \\
\theta_{ph2} \\
\theta_p
\end{pmatrix}
\]  
(4.20)
Equation 4.16, Equation 4.17, Equation 4.18, Equation 4.19 might thus be written in matrix form

$$ [M]\{\ddot{s}\} + [C]\{\dot{s}\} + [K]\{s\} = \vec{e} \quad (4.21) $$

The matrices $M, C, K$ are respectively the mass matrix, the damping matrix and the stiffness matrix.

Such matrices are

$$ M = \begin{bmatrix} I_{ms} & 0 & 0 & 0 \\ 0 & I_{ph1} & 0 & 0 \\ 0 & 0 & I_{ph2} & 0 \\ 0 & 0 & 0 & I_p \end{bmatrix} \quad (4.22) $$

$$ C = \begin{bmatrix} c_{ms} & 0 & 0 & 0 \\ 0 & c_{ss1} & 0 & 0 \\ 0 & 0 & c_{ss2} & 0 \\ 0 & 0 & 0 & c_{eq} \end{bmatrix} \quad (4.23) $$

$$ K = \begin{bmatrix} k_{ms} & 0 & 0 & 0 \\ 0 & k_{ss1} & 0 & 0 \\ 0 & 0 & k_{ss2} & 0 \\ 0 & 0 & 0 & 0 \end{bmatrix} \quad (4.24) $$
The vector $\vec{e}$, which contains all the input of the system, is

$$
\vec{e} = \begin{pmatrix}
c_{ms}\dot{\theta}_H + k_{ms}\theta_H - G1 - T1 \\
c_{ss1}\dot{\theta}_H + k_{ss1}\theta_H - G2 \\
c_{ss2}\dot{\theta}_H + k_{ss2}\theta_H - G4 \\
T1 + T2 + G1 + G2 + G4 + c_{eq}\dot{\theta}_H - T_{ext}
\end{pmatrix}
$$

(4.25)

### 4.3 State Space

The state of motion of a system is completely known when positions and velocities are known. Such variables taken together are thus the state variables of the system.

A state space is a space defined by a reference frame whose coordinates are the state variables of the system.

A state space representation is a mathematical model of a physical system as a set of input, output and state variables related by first order differential equations. This representation provides a convenient and compact way to model and analyze systems with multiple inputs and outputs.

The state equations are usually written in the form

$$
\begin{align*}
\ddot{z} &= A\dot{z} + B\ddot{u} \\
\dot{y} &= C\dot{z} + D\ddot{u}
\end{align*}
$$

(4.26)

where

- $\ddot{u}$ is the input of the system;
• $\vec{y}$ is the output of the system;

• $\vec{z}$ is the state vector, which is the vector which contains the state variables;

• $A, B, C, D$ are a set of four matrices which defined the system.

The state vector, as said, contains the state variables. Hence it is

$$\vec{z} = \begin{pmatrix} \dot{s} \\ s \end{pmatrix} \quad (4.27)$$

The inputs of the system are the crankshaft rotary speed $\dot{\theta}_H$ and the external load $T_{ext}$ exerted by the accessory. However, as said before, all the generalized forces which replace the non linearities, are considered input of the system as well.

Thus the input vector is

$$\vec{u} = \begin{pmatrix} \dot{\theta}_H \\ \theta_H \\ T_{ext} \\ T1 \\ T2 \\ G1 \\ G2 \\ G4 \end{pmatrix} \quad (4.28)$$

On the other hand the output vector is not determined by the system itself, but it depends on which variables are considered important to study. In this case the variables which are
observed and set as output of the state space are all the velocities and the displacements. Hence the output vector is equal to the state vector.

Thus

\[
\vec{y} = \begin{pmatrix}
\dot{\theta}_{ms} \\
\dot{\theta}_{ph1} \\
\dot{\theta}_{ph2} \\
\dot{\theta}_p \\
\theta_{ms} \\
\theta_{ph1} \\
\theta_{ph2} \\
\theta_p
\end{pmatrix}
\] (4.29)

Once the vectors \(\vec{u}, \vec{y}, \vec{z}\) are determined, the set of matrices A,B,C,D can be obtained introducing such vectors in the equation of motion written in the configuration space.

From Equation 4.21 it follows

\[
\{\ddot{s}\} = -[M]^{-1}[C]\{\dot{s}\} - [M]^{-1}[K]\{s\} + [M]^{-1}\vec{e}
\] (4.30)
The vector $\vec{e}$ can be expressed as a function of the input vector $\vec{u}$

$$\vec{e} = \begin{bmatrix} c_{ms} & k_{ms} & 0 & -1 & 0 & -1 & 0 & 0 \\ c_{ss1} & k_{ss1} & 0 & 0 & 0 & 0 & -1 & 0 \\ c_{ss2} & k_{ss2} & 0 & 0 & 0 & 0 & 0 & -1 \\ c_{eq} & 0 & -1 & 1 & 1 & 1 & 1 & 1 \end{bmatrix} \cdot \begin{bmatrix} \dot{\theta}_H \\ \theta_H \\ T_{ext} \\ T1 \\ T2 \\ G1 \\ G2 \\ G4 \end{bmatrix}$$

(4.31)

To make the notation easier to handle the above expression can be written as following

$$\vec{e} = [E]\{\vec{u}\}$$

(4.32)

By introducing Equation 4.32 in Equation 4.30 it follows

$$\{\ddot{s}\} = -[M]^{-1}[C]\{\dot{s}\} - [M]^{-1}[K]\{s\} + [M]^{-1}[E]\{\vec{u}\}$$

(4.33)

Hence Equation 4.26 becomes

$$\begin{bmatrix} \ddot{s} \\ \dot{s} \end{bmatrix} = \begin{bmatrix} -[M]^{-1}[C] & -[M]^{-1}[K] \\ I_{4x4} & 0_{4x4} \end{bmatrix} \cdot \begin{bmatrix} \dot{s} \\ s \end{bmatrix} + \begin{bmatrix} -[M]^{-1}[E] \\ 0_{4x8} \end{bmatrix} \cdot \vec{u}$$

(4.34)
On the other hand, since the output vector $\vec{y}$ is equal to the state vector $\vec{z}$ and it is not affected by the input vector $\vec{u}$ Equation 4.26 becomes

$$\begin{pmatrix} \dot{s} \\ s \end{pmatrix} = \begin{bmatrix} I_{8x8} \end{bmatrix} \cdot \begin{pmatrix} \dot{s} \\ s \end{pmatrix} + \begin{bmatrix} 0_{8x8} \end{bmatrix} \cdot \vec{u}$$

(4.35)

The set of matrices $A,B,C$ and $D$ are so known

$$A = \begin{bmatrix} -[M]^{-1}[C] & -[M]^{-1}[K] \\ I_{4x4} & 0_{4x4} \end{bmatrix}$$

(4.36)

$$B = \begin{bmatrix} -[M]^{-1}[E] \\ 0_{4x8} \end{bmatrix}$$

(4.37)

$$C = I_{8x8}$$

(4.38)

$$D = 0_{8x8}$$

(4.39)

### 4.4 Numerical model

Equation 4.34 and Equation 4.35 were computed using Simulink.

Simulink, developed by MAthWorks, is a commercial tool for modeling, simulating and analyzing multidomain dynamic systems. It allows to represents a mechanical system as a set of proper blocks. The Simulink model made is shown in Figure 31.

The above model has 5 main blocks

- State-Space block
Figure 31. Simulink model of the DDOS

- Rotary speed block
- Experimental data filter block
- External load block
- Non linearities block

The State-Space block belongs to Simulink library, Figure 32.

It computes the output $\vec{y}$ from the input vector $\vec{u}$ through the set of matrices $A, B, C, D$. Such matrices are defined in a M-file script, and are called back when the model is opened.

The rotary speed block, Figure 33, loads the crankshaft rotary speed from a MAT-file and converts it from rpm to rad/s.

The change of unit of measure from rpm to rad/s is necessary because all the coefficients in the matrices A, B, C, D are in the SI while the input velocity is commonly expressed in rpm.
The experimental data filter block, Figure 34, loads the experimental pulley rotary speed and filters it.

The importance of such block will be clear in the next chapter dealing with the validation of the model. However it is better to explain now what this block does. The block reads the experimental pulley speed and filters it according to a lowpass filter. The lowpass filter attenuates all the measure whose frequency is high than a threshold frequency, which in this
case is 500 rad/s. Before and after the filter, there are two gains which convert the input speed from rpm to rad/s and from rad/s to rpm, respectively.

The external load block, Figure 35, simulates the external load exerted on the pulley by the accessories of the the front-end.
The block reads the pulley rotary speed as input and provides the torque exerted on it as output.

The block has three lookup tables which represent the accessories of the front-end. Each lookup table is defined by characteristic load [Nm] versus speed [rpm] of each accessory. The load exerted by each accessory depends on its rotary speed. Assuming that such speed is equal to the rotary speed of the pulley, the torque of each accessory is the output of the relative lookup table.

After each of them there is a gain which is the gear ratio of the respective accessory.

All the torques so obtained are then summed and this last signal is routed through a switch block.

Such block pass through the first input when the second input satisfies the criterion, otherwise it pass through the third input. The inputs of the switch block are, as it is shown in Figure 35, zero, time and external load, respectively. The pass-through criterion is input time greater or equal than a threshold, which must be defined.

Such value of threshold represents the instant in which the accessories switch on.

Acting on this value it is possible to include or exclude the external load due to the accessories.

The last main block is the non linearities block, Figure 36.

The block has the vector $\vec{y}$ as input and provides the values of the generalized forces $T_1$, $T_2$, $G_1$, $G_2$, $G_4$ as output.
There are two subsystems which compute the forces due to frictions, T1 and T2, and the forces due to internal clearances, G1, G2 and G4.

The first subsystem is shown in detailed in Figure 37.

The input of the block are velocities and positions of the components which define the system, the crankshaft rotary speed and the torque exerted by the accessories.

The signals relative to the velocities of the two pin heads are terminated since they do not affect the frictions.

On the other hand the pulley rotary speed is routed away from the rotary speeds of the crankshaft and of the main spring, to obtain the two relatives speeds which determine the sign of the forces T1 and T2.

Each of them are computed in its own block.

The block T1 compute the force T1, Figure 38.
The relative angular speed $\dot{\theta}_{ms} - \dot{\theta}_p$ becomes important once it is integrated and considering its sign, using a look up table which determines the sign of the force $T_1$.

The value of such force is computed in the block called ramp friction, Figure 39.

Here, the value of the friction exerted by the pads on the inner surface of the pulley, $T_1$, which is considered constant, is added to a force which comes from the lookup table. Such force represents the increment of friction due to the presence of a ramp before the cog on the inner surface of the pulley. Such value depends on both the slope of the ramp and the relative position between the main spring and the pulley, $\theta_{ms} - \theta_p$.

On the other hand, the force $T_2$ is computed in the block $T2$, Figure 40.

The sign of such force is computed, as before, by integrating the relative velocity $\dot{\theta}_H - \dot{\theta}_p$ taking into account its sign.
On the other hand, the value of the force, which depends on the external load is determined in the block bushing friction T2, Figure 41.

The inputs in this block are the external load, the tension imposed by the belt tensioner and the wrap angle.

These data are routed through the law of cosines

\[
H_l = \sqrt{F_1^2 + F_2^2 - 2 \cdot F_1 \cdot F_2 \cdot \cos(\alpha)} \tag{4.40}
\]

where \(F_1\) and \(F_2\) are the tensions at the ends of the belt and \(\alpha\) is the wrap angle, to compute the hub load \(H_l\).

Then, the so obtained hub load is multiplied by two gains which represents the friction coefficients and the bushing radius.
The other subsystem in Figure 36 is the internal clearances block, Figure 42. This block supplies the generalized forces due to internal clearances as output.

The positions of each mass of the system are combined to obtain the relative displacements \((\theta_{ms} - \theta_p), (\theta_{ph1} - \theta_p)\) and \((\theta_{ph2} - \theta_p)\).

Such relatives displacements enter in three different lookup tables which provides the force generated according to the relative position.

The three look up tables are defined by three tables, whose graphs are shown in Figure 27, Figure 28 and Figure 29, which are obtained implementing the respective VB function on an Excel spreadsheet.

Such VB functions are listed as annex B.

All the data of the system are collected in an M-file called \(DDOS\_data\), which is called back when the Simulink model is opened.

Such M-file is listed as annex C.
Figure 40. Block of friction T2

Figure 41. Bushing friction block
Figure 42. Internal clearances block
CHAPTER 5

EXPERIMENTAL VALIDATION

Verification and Validation is the process of checking that a product, service or system satisfies the specified requirements specifications and that it fulfills its intended purpose. The Experimental validation described in this chapter assess the numerical response of the model is consistent with the experimental data. There are four different test which can be used to valid the model. They are the static test, which it was described in the chapter of the static characteristic, the run up with test with all the accessories of the front-end turned off, the run up test with all the accessories turned on and the start and stop test. For each one of the tests listed above it will performed a numerical simulation, and the results of such simulations, which are the time histories of rotative speed and positions, will be compared to the results of the experimental tests.

5.1 Static test

The static characteristic was described in chapter three.

The static characteristic can be produced numerically, by constraining the displacement of the hub and exerting an external torque on the pulley. Such torque, which has a sinusoidal trend, must be slow enough not to be affected by the inertial effects.

In order to do that the simulink model of Figure 31 must be modified.

The new one is shown in figure Figure 43.
As seen before, the static characteristic is defined as the plot relative displacement between pulley and hub versus torque exerted.

Therefore the velocities are terminated as well as all the displacement apart from the pulley displacement. It is useful to terminate all the non requested amounts, because it allows to save memory and it makes the simulation faster.

The non linearities block, which produces the effects of the non linearities is the same of the previous one as well as the State-Space block. However, the initial conditions defined inside the State-Space block are changed since now, at the beginning of the simulation both velocities and initial positions are zero.
The rotary velocity input block, which simulates the rotary speed of the crankshaft, is changed. Indeed, as the hub of the device is constraint, its velocity and position must be zero. This new block is shown in Figure 44.

![Figure 44. Crankshaft rotary speed for static test simulation](image)

The external torque block is changed too.

During the working condition the external torque depends on the pulley rotary speed. On the contrary, during a static test the torque is exerted by the testing machine, hence it does not depend on the pulley rotary speed anymore.

The new external torque block is shown in Figure 45. The torque is generated by the sine wave block. Such block allows to set both amplitude and frequency. The amplitude is chosen to be 120Nm while the frequency was set to be 0.2rad/s.

The result of the simulation is shown in Figure 46.
Unfortunately a direct comparison between the curve so obtained and the actual static characteristic is not possible, since the static test machine does not supply any numerical data which can be loaded on Matlab. However, it is still useful to graphically compare the two curves in order to find analogies and differences. The actual static characteristic is shown below in Figure 47.

The numerical curve is much more clear than the actual one which presents several irregularities, especially in the dead-angle zone. This is mainly due to the irregularities of the inner surface of the pulley which affect the contact with the sliding pads. Due to this irregularities the friction is not constant. Nevertheless it is not possible, to take into account these effects, at least for what it concerns this model. A FE model might be able to describe this phenomenon, but the simulation time would clearly increase.

Furthermore, the numerical result shows two undulatory motions, typical of a mass-spring system, at about 10.5 in overstress and -67 in overrunning. Such motions are generated by the elastic response of the springs at such points. Indeed, both in overstress and overrunning, the pulley is stuck until the external load is lower than the friction force. When the external
load overcomes the threshold, the system starts moving and suddenly cross the whole dead zone reaching the springs. During this phase, the system store enough kinetic energy to make springs oscillate.

On the contrary, the actual static test machine, does not apply an external torque, since it is measured with a Strain Gauge, but it imposed the pulley to move very slowly. In this way the pulley does not store kinetic energy and the impact with the springs is soft enough to make the springs not to oscillate.

However, there are some common points between the two curves which it is worth to underline.

First of all the aspects of both are more or less the same.

Then, a deeper analysis yield to the same maximum displacements.
Finally dead-angle in overrunning and overstress in Figure 46 are almost the same of the relative in Figure 47.

5.2 **Run up test with accessories removed**

The Run up curve is a kind of test in which the crankshaft rotary speed goes from 800 rpm to almost 5000 rpm very slowly. It lasts about 90s, even though this time varies from company to company.
So that the device fixed securely to the crankshaft pass through all the possible frequencies, and it is possible to verify whether or not a resonance occurs.

The time history of the crankshaft rotary speed is shown in Figure 48.

![Figure 48. Time history of the crankshaft rotary speed with accessories removed](image)

During this test the accessories are turned off, hence they do not exert any torque on the output pulley.

However, they still affect the motion of the pulley since, it is operatively coupled to each of the belt driven accessory component by the belt. Hence, the equivalent inertia of the pulley is affected by the inertia of each component operatively coupled to it.
Experimentally, both the time history of the crankshaft and of the output pulley are obtained through a pulse ring.

Setting this hub spin speed as input speed, the simulink model compute the response of the pulley. In Figure 49 is shown the result of such simulation together with the experimental result.

![Graph showing comparison between numerical and experimental result](image)

Figure 49. Comparison between numerical and experimental result of a run up speed with accessories removed

Figure 50 and Figure 51 show a zoom on the constant part of the graph and a zoom on the speed ramp respectively.
The results is positive since the two results are very close. However, the experimental pulley spin speed present much more peaks than the numerical one. This effect might be due to an observational error. Indeed, the precision of a pulse ring usually leads to have a sample rate of about 100 Hz. Hence all the measure characterized by a frequency greater than this value do not make much sense. Hence, by filtering the signal through a lowpass filter, it is possible to make the pulley speed cleaner.

Above, in Figure 52 both the speed are plotted.

The filtered pulley spin speed is plotted in Figure 53 together with the numerical result.

Figure 54 and Figure 55 show respectively a zoom of the result in the first part of the test, when the speed is constant, and on the second part, when the speed increases.

Figure 50. Numerical and experimental response of the pulley moving at constant mean speed with accessories removed
Figure 51. Numerical and experimental response of the pulley on the speed ramp with accessories removed

Figure 52. Comparison between experimental and filtered pulley spin speed during the run up with no external load
The positive feelings are confirmed by the very close response shown in Figure 54 and Figure 55.

5.3 **Run up test with accessories included**

This test is quite similar to the previous one.

The crankshaft rotary speed increase from 800rpm to about 5000rpm crossing all the frequency spectrum very slowly.

However this time, the accessories are turned on, hence they exert a torque on the pulley. Such torque is a function of the pulley speed itself and of the characteristic of each component operatively coupled to the output pulley.
Figure 54. Numerical and filtered response of the pulley moving at constant mean speed with no accessories

Figure 55. Numerical and filtered response of the pulley on the ramp with accessories removed
The time history of the crankshaft spin speed is shown in Figure 56.

![Figure 56. Time history of the crankshaft spin speed with accessories included](image)

Setting such speed as input speed and turning on the accessory, by setting 0 as threshold in the signal switch in the external load block, it is possible to compute the numerical response of the pulley during the run up test with accessories turned off.

The result of the simulation is shown in Figure 57 together with the experimental result obtained in similar conditions.

Obviously it is difficult to analyze the result.
Figure 57. Comparison between numerical and experimental result of a run up speed with the external load

However, by zooming on the first part of the graph and one on the middle of the ramp, they are obtained Figure 58 and Figure 59 respectively.

Both the results evidence that the experimental result has much more peaks than the numerical one. For this reason it might be interesting to filter the experimental data as done before and observe if whether the results get closer or farer.

The speed is filtered by a lowpass filter with a threshold set to be $500 \text{rad/s}$ as the previous test. The result is shown in Figure 60 together with the experimental curve.

Figure Figure 61 shows the graphs of both numerical and filtered pulley speed.

Although the experimental speed has been filtered, the graph is not clear enough yet to allow a comparison. Therefore it is necessary and useful to make a zoom.
Figure 58. Numerical and experimental response of the pulley moving at constant mean speed with accessories included

Figure 62 and Figure 63, which are both relative to the run up test with accessories on, confirm the reliability of the simulink model for this test.

5.4 Start and Stop test

The Start and Stop test is different from all the others seen so far. It consists on a test in which the engine, therefore the crankshaft as well, starts rotating when it is off. Hence the crankshaft spin speed goes from 0rpm to 800rpm. Then the mean spin speed remains constant for about 10 seconds. After that, the engine is switched off and the crankshaft spin speed goes to 0rpm again.

In Figure 64 is shown the time history of the crankshaft speed for one start and stop.
This test is particularly interesting because it allows the study of the pulley output during the initial phase.

Indeed, as stated in the introduction of this thesis, during the initial phase there is a phenomenon of resonance, hence the displacement of the pulley is very elevated.

However, it is important to limit such displacement as much as possible to prevent the belt from slipping.

By setting the time history shown in Figure 64 as input in the simulink model, it yields the following dynamic response of the pulley.

Figure 65, Figure 66, Figure 67 show that the experimental and the numerical speeds of the output pulley are very close.
However, it is interesting to filter the experimental data with a lowpass filter set to 500 rad/s as it was done for the previous simulations.

The two speeds, experimental and filtered, are plotted in Figure 68.

Such figure is not clear enough to show the difference between the two graphs, since they have too many peaks. Therefore, it is useful to focus on both the first part of the test and the ending part of the test, which are shown respectively in Figure 69 and Figure 70.

These plots show how, by filtering the experimental speed with a lowpass filter the amount of peaks decreases sharply. It might mean that several peaks are not really peaks of the speed of the pulley but are simply errors of measurement.
Figure 61. Comparison between numerical and filtered result of a run up speed with accessories included

The filtered pulley rotary speed is shown together with the numerical response in the graph of Figure 71.

The two responses are even closer than before. This is shown better through the zooms during the starting and the ending of the test which are shown in Figure 72 and Figure 73 respectively.

The zooms of the response show an interesting phenomenon.

The pulley speed obtained numerically and the filtered one are almost the same both during the ending and when the crankshaft mean speed is constant.

On the other hand, the amplitude of the speed variations during the starting is slightly different.
Such difference of amplitude attenuates as the simulation goes on until the two speeds match almost exactly.

This is a matter of fact that the phenomenon which makes the two speeds different is transitory, hence it is strongly affected by the initial condition of the system.

In all the simulations performed so far, it was assumed that at the beginning of the test, the pulley and the main spring were in contact.

However this is not the only feasible initial condition, since the system might be in the "dead-angle" zone.
Recall that the initial conditions of all the components of the system are set in the State-Space block, while the angular distances between each component and the respective spring are defined through lookup tables.

Assuming that, at the beginning of the Start and Stop test simulation, the pulley is 10 far from the main spring, the simulation yields to the following result.

A zoom of the starting shows how the speeds are much closer to the experimental one than the previous simulation, when the pulley and the main spring were assumed to be in contact.

The same work can be done by setting the initial distance between the pulley and the main spring equal to 15.
In this case the result of the simulation is

A zoom of the starting of such simulation is shown in Figure 77.

At this point, it is interesting to plot several responses of the pulley, each one with the relative initial condition, on the same graph in order to evaluate how much and in which way the initial conditions affect the output pulley spin during the starting. Such graph is shown in Figure 78.

The above figure show that after a certain point the effect of the initial conditions attenuated and all the responses converge.

By making a zoom in on the first part of the graph, Figure 79, it is possible to see clearly the different responses of the pulley due to the different initial conditions.

The responses of the output pulley, obtained numerically, for different time histories in input shown in Figure 53, Figure 61, Figure 71 and the respective zooms in on the most interesting part of each graph, evidence that the simulink model is able to simulate the dynamic behavior of the system.

5.5 Root-Mean-Square Deviation

The Root Mean Square Deviation is a reliable index of the difference between values predicted by a model and the values actually observed.

It is frequently used to determine if whether a model fit the experimental data or not.

The Root Mean Square Error is defined as the square root of the mean square error

\[ \theta_{RMS} = \sqrt{\frac{\sum (\hat{\theta}_{1,i} - \hat{\theta}_{2,i})^2}{n}} \]  

(5.1)
The Root Mean Square errors of each simulation are collected in Table VI.

TABLE VI
ROOT MEAN SQUARE ERRORS

<table>
<thead>
<tr>
<th>θ_{RMS,off}</th>
<th>9.3 rpm</th>
</tr>
</thead>
<tbody>
<tr>
<td>θ_{RMS,on}</td>
<td>11.5 rpm</td>
</tr>
<tr>
<td>θ_{RMS,SKS}</td>
<td>15.3 rpm</td>
</tr>
</tbody>
</table>
Figure 64. Time history of the crankshaft spin speed during the Start and Stop test

Figure 65. Numerical and experimental pulley angular speed during the Start and Stop test
Figure 66. Numerical and experimental pulley angular speed during the initial phase

Figure 67. Numerical and experimental pulley rotary speed during the final phase
Figure 68. Experimental and filtered pulley angular speed

Figure 69. Comparison between filtered and experimental response of the pulley during the initial phase
Figure 70. Comparison between filtered and experimental response of the pulley during the final phase

Figure 71. Filtered and numerical pulley speed during the Start and Stop test
Figure 72. Zoom of the filtered and numerical response during the initial phase

Figure 73. Zoom of the filtered and numerical response during the final phase
Figure 74. Simulation response for initial condition equal to 10

Figure 75. Zoom of the starting of the start and Stop test simulation for initial condition equal to 10
Figure 76. Simulation response for initial condition equal to 10

Figure 77. Zoom of the starting of the start and Stop test simulation for initial condition equal to 10
Figure 78. Comparison of several output pulley spin speeds for different initial conditions

Figure 79. Transitory effect of the initial conditions on the pulley rotary speed
CHAPTER 6

SENSITIVITY ANALYSIS

Sensitivity Analysis (SA) is the study of how the variation in the output is affected to different variations in the input of the model. It is a technique for systematically changing variables in a model to determine the effects of such changes. The SA, which is performed in this chapter, is divided into three parts. First it is necessary to define the output which describe the behavior of the system. Then, the same must be done for the input parameter which may affect the functioning. Such parameter, will then be collected in a table, and, finally, using the simulink model, all the simulation relative to each case of study will be performed. The results of such simulations will be the guideline for the design review

6.1 Output parameter of the analysis

The task of the DDOS is to filter the vibration of the crankshaft.

The ideal decoupler is the one who provides torque to the front-end accessory drive in only one direction. Such device would filter completely the vibration of the crankshaft.

Unfortunately, such system does not exist, since there will always be at least a very small fluctuation in the velocity of the pulley.

However, it is very important to make such fluctuation as small as possible.
The first important parameter to appraise the functioning of the system is, thus, the *filtering ratio*, which is the ratio between the amplitude of the speed fluctuation of the crankshaft and of the pulley.

Furthermore, the experimental tests registered strange noises when the springs hit the pulley. This is one of the issues which affect the device, hence it must be studied carefully. Unfortunately, the noise emitted during an impact is not easy at all to quantify. However, this obstacle can be avoided by performing a qualitative study of the phenomenon. Indeed, according to the structure of the simulink model, the impact between one of the springs and the pulley occurs when there is clearance between them.

At that point, thus, the relative external generalized force increases suddenly. Assuming that, higher is the variation of the force, bigger is the amplitude of the impact, such force can be a qualitative meter to make a comparison between the nominal and the changed case.

Finally, the parameter to observe in order to appraise the functioning of the system are the spin speed of the pulley and the graphs of the external generalized forces which supply the effect of the internal clearances.

### 6.2 Input parameter of the analysis

The physical variables, which characterize the simulink model are the friction $T_1$, the friction $T_2$, the stiffness of the main spring $k_{ms}$ and those of the secondary springs $k_{ss1}$ and $k_{ss2}$, the values of the internal clearances and the inertia of the components.
However, although all the variables listed above can be varied in the simulink model, not all of them will be taken into account during the Sensitivity Analysis.

Indeed, only the changes, which represents feasible solution, will be considered.

The friction $T_1$ is due to the relative motion between the pulley and the main spring. Its nominal value is equal to 15 Nm. It depends on the number of the sliding pads interposed between the two surfaces, the material of the pads, hence the friction coefficient, and the radial force exerted by the main spring. This is the most important factor which affect the friction.

Therefore, by varying the radial compression during the assembly it is possible to vary $T_1$. In the simulations $T_1$ will be assumed to be equal to 0 Nm, 5 Nm, 10Nm and 25 Nm.

However, it is important to underline that, such friction is strongly affected by a lot of factors, which cannot be taken into account with the simulink model, like temperature, surface roughness and geometrical irregularities of the components.

For this reason the variation of this friction is not feasible.

On the other hand, a very important variable, which can be modified, is the friction $T_2$.

Such force, which is proportional to the radial load, is strongly affected by the physical and geometrical characteristics of the journal bearing.

An interesting case of study is represented by the solution with a ball bearing, which supports the radial load, instead of the bushing.

This technological solution makes both the friction coefficient and the distance of the force application point, much smaller.

The ball bearing considered is the NSK 6910 DDU.
Furthermore, it was performed a simulation in which the friction T2 was given by the sum of the ball bearing contribute and a constant load.

Such constant value might represents the effect of a cover which constrains the pulley not to move along its axis.

In the simulation the constant load was assumed equal to 5Nm.

The last variables to analyze are the internal clearances and the stiffness of the secondary springs.

The simulations performed during the experimental validation showed that the secondary springs did not work.

Therefore, it will not be studied neither the case in which the internal clearances increase, nor the case in which the internal clearances does not vary while the stiffness increase.

On the other hand, it is interesting to valuate the output when the internal clearings decrease.

First, it was analyzed the influence of the clearance G4, by assuming different values of such amount, while all the other parameters were equal to the original design.

Next, it was carried out a study on the influence of the stiffness of the secondary spring $k_{ss2}$, which is relative to the clearings G4 and G5. As said before, such springs did not work, therefore, the values of such internal clearances were set small enough to allow them to transmit force.

Finally, the analysis focused on the effect of the variation of the ramp, which is seated before the cog of the pulley, on the the behavior of the DDOS.
6.3 Influence of T1

The influence of the friction T1 on the spin speed of the pulley is shown in Figure 80, Figure 81 and Figure 82.

Such graphs compare the rotary speed of the pulley for different values of T1.

In order to have a better filtering, the velocity must fluctuate as less as possible, or, at least, the fluctuation of the modified system must be lower than the original design one.

Overall, the graphs show that the behavior of the DDOS worsen as the friction decreases. Focusing the attention on the run up tests, this effect is more visible when the accessories are off. The reason is that, when the accessories are on, the friction T2, which is proportional to the radial load, increases. Therefore, this increment compensates the reduction of T1.

On the other hand, the Start and Stop simulation show that when the friction T1 is null or very small, the speed fluctuation are very high and they occur when the mean speed of the crankshaft is constant. Such variations can explained through the analysis of the forces exerted during the impacts.

Figure 83 and Figure 84 show the force G4 during the run up with accessories off, and the start and stop, respectively.

The first plot shows that the pulley impacts the secondary spring in over-running several times.

By making a comparison between Figure 80 and Figure 83, it is possible to see that the impacts occur at the same time of the high speed variations. This point explains why the filtering effect worsen.
Figure 80. Influence of the friction T1 on the response of the pulley when the accessories are excluded

Figure 81. Influence of the friction T1 on the response of the pulley when the accessories are included
Figure 82. Influence of the friction T1 on the response of the pulley during the start and the stop of the engine

Figure 83. Influence of T1 on the impact with the secondary springs during the run up with accessories excluded
The second graph, on the other hand, shows that after a certain time the secondary springs start working and make the system unstable.

Actually, the friction T1 is a very difficult to quantify exactly, since it is affected by many external factors, like: temperature, surface roughness, geometrical tolerances.

Therefore, the analysis done so far is useful to understand the behavior of the DDOS, but it does not supply any feasible changes to bring.

6.4 Influence of T2

A feasible change, which can be applied to the system, concerns the friction parameter T2.
The variation of such variable is made by replacing the journal bearing in the original design with the ball bearing *NSK 6910 DDU*. The data to substitute in the simulink model are the friction coefficient \( f = 0.0013 \) and the radius \( r = 34\text{mm} \), which are collected in the catalogue.

![Graph](image)

**Figure 85.** Response of the system equipped with a ball bearing, instead of the journal bearing, when the accessories are excluded

The pulley speed of such system is plot, together with the response of the original DDOS, in Figure 85, Figure 86 and Figure 87.

The graphs show that the fluctuation of the speed increased. This result is concordant with what was seen before, analyzing the influence of the friction \( T_1 \).
Indeed, the previous analysis shows that a very low value of friction cause the pulley to oscillate.

Therefore, the oscillations of the pulley might be reduced increasing the friction T2.

Figure 88, Figure 89 and Figure 90 show the response of the system equipped with the same ball bearing of the previous case and an additional constant value equal to 5Nm.

Such constant value, might be the contribute due to the axial force exerted by a cover on the pulley. This cover is fixed to the hub and it axially constrains the pulley.

Overall the filtering of the DDOS with this design increased, even though the influence of the T2 is less evident when the accessories are off.

However, the comparison with the original response evidences that, the oscillations of the pulley are still greater, especially at the starting, as shown better in Figure 91.

This design is particulary interesting, since it allows to solve one of the issues of the DDOS.

Indeed, the hub and the pulley are not perfectly coaxial. Therefore, to allow them to be coupled they are designed with a certain tolerance, so that there is always a minimum clearance between them. Otherwise, it might verify the situation in which, the outer diameter of the hub is larger than the inner diameter of the pulley.

However, the clearance cause the pulley to oscillates around the axis of the hub. This movement of the pulley creates noise and wearing.

The ball bearing, however, solve the problem of the non coaxiality and all the issues related to it.
Figure 86. Response of the system equipped with a ball bearing, instead of the journal bearing, when the accessories are included

Figure 87. Response of the system equipped with a ball bearing, instead of the journal bearing, during and the start and the stop of the engine
Figure 88. Response of the system equipped with a ball bearing, instead of the journal bearing, and a cover when the accessories are excluded

Figure 89. Response of the system equipped with a ball bearing, instead of the journal bearing, and a cover when the accessories are included
Figure 90. Response of the system equipped with a ball bearing, instead of the journal bearing, and a cover during and the start and the stop of the engine.

The forces exerted during the impacts between the pulley and the free edge of the main spring are shown in Figure 92, Figure 93, Figure 94.

Such figures compare the forces G1 of the original design with those one exerted when the ball bearing is interposed between pulley and hub.

The first graph is obtained by assigning to the crankshaft a run up motion with the accessories off. Such figure, shows the impacts which occur when T2 is reduced, while in the original design there were not any shock.
The influence of T2, on the force G1, is less evident when the accessories are switched on. Indeed, as shown in figure Figure 93, the main spring is always engaged with the pulley. Therefore, the force G1 is never null.

It is worth to recall that, the analysis of the forces G1, G2 and G4 was carried out to obtain information about the amplitude of the shocks. However, from those graph, an impact occurs only when the respective force is impulsive.

Finally, during the start and the stop of the engine, the forces of each design are quite close. However, the DDOS with the new design worsen slightly respect to the original one.

The analysis of the impacts is completed studying the force G2, which is the force exerted when the pulley and the the secondary spring engaged.
The results are shown in Figure 95, Figure 96 and Figure 97.

The graphs of the force G2 are reliable for the force G4 as well, since the two secondary springs are exactly the same.

The graphs show that the secondary springs do not work during the run up test, whether the accessories are on or off.

On the other hand, the behavior worsens during the start and the stop. Indeed, the study of the new design evidences the presence of two impacts, while the springs were completely at rest in the original design.

According to the analysis carried out so far, the system might be improved by increasing the load exerted by the cover.

Figure 92. Influence of T2 on the force G1 during the run up with accessories excluded
6.5 Influence of G4

G4 is the angular distance between the pulley and the secondary spring $k_{ss2}$ when the pulley engaged the main spring in over-stress.

According to the results of the analysis computed on the original design, the secondary springs never engage the pulley.

Therefore, to observe how this parameter affects the behavior of the DDOS it is necessary to reduce that clearing.

Figure 98, Figure 99 and Figure 100, show the rotary speed of the pulley for several values of G4.
The first two graphs allow to study the filtering of the DDOS when the accessories are off and on respectively.

It is quite clear that, this parameter does not influence the filtering that much, at least as it remains positive. Indeed, a negative value of G4 would mean that the respective secondary spring engaged the pulley before the main spring.

On the other hand, the behavior of the DDOS at the start, worsens as much as the clearance G4 decreases. This influence is shown in Figure 100 and Figure 101. This last picture is a zoom in on the start.

In Figure 102, Figure 103 and Figure 104 is plotted the force G1 for different values of G4. Overall, the influence of G4 on the amplitude of G1 is negligible.
However, the number of the impacts increases as the clearance decreases.

On the other hand, the graph relative to the run up with accessories on, does not show impulses. Indeed, in this case the main spring is always engaged, since it is exerting torque. This figure evidences that, smaller is the clearing $G_4$, lower is the value of $G_1$.

This behavior is concordant to the geometry of the device. Indeed, as the secondary spring engages the load exerted by the pulley on the main spring reduces. The amplitudes of the impacts between the pulley and the secondary spring $k_{s_2 s_2}$ are shown in Figure 105, Figure 106 and Figure 107.

Such graphs are not reliable to quantify the intensity of the shocks which occur.

However, they are useful to see that smaller is the distance $G_4$, bigger is the force $G_4$ exerted at the impact.
The sensitivity analysis of the angular distance \( G_5 \) on the outputs of the system is shown in annex D.

### 6.6 Influence of \( k_{ss2} \)

The last two sensitivity analysis show how the behavior of the DDOS is affected by the reduction of the internal clearances \( G_4 \) and \( G_5 \).

Such clearings establish the instant in which the secondary spring \( k_{ss2} \) engages respectively in over-stress and over-running.

However, it is interesting to observe the influence of the stiffness of such spring on the system.
6.7 **Influence of the ramp**

In chapter dedicated to the description of the components, the pulley, which is one of the most important part of the DDOS, was deeply described.

However, it was not paid much attention on the slope seated before the cog, which, during the functioning, engages the main spring.

The interesting point is that, before the impact occurs, the free edge of the main spring climbs this inclined surface. Therefore, the spring compress radially.

Such compression, increases the normal force, which is exerted by the spring on that surface. Thus, the friction, which is given by the relation

\[ F = N \cdot f \]  

(6.1)
increases as well.

| $f$ is the friction and $N$ is the normal force, which is given by |

$$N = k_\tau \cdot \delta r$$  \hspace{1cm} (6.2)

where $k_\tau$ is the stiffness of the spring in the radial direction and $\delta r$ is the radial compression.

Both $k_\tau$ and $f$ are constant.

Furthermore, the distance between the point where the friction is applied and the rotation axis is

$$l = (R_{pi} - \delta r)$$  \hspace{1cm} (6.3)
where \( R_{pi} \) is the radius of the inner surface of the pulley.

Therefore, the friction torque due to the ramp, which is

\[
\delta T_1 = F \cdot l 
\]  

becomes

\[
\delta T_1 = k_r f (R_{pi} - \delta r) 
\]  

hence

\[
\delta T_1 = \delta T_1(\delta r) 
\]
Figure 100. Influence of the clearing G4 on the filtering during the start and stop of the engine

The last relation means that the increment of the friction torque depends on only the radial compression.

The shape of such surface, which is shown in Figure 108, is obtained from the drawing of the original design.

The area under the curve represents the kinetic energy dissipated by the increment of friction.

This is an interesting point, which can be used to improve the functioning of the DDOS.

Indeed, by increasing the area, which underlays the curve, it is possible to dissipate more kinetic energy. Thus, it is possible to soften the impact.
Such area was modified in three ways. First, the actual ramp was replaced with a constant slope one. Then it was considered the case in which the slope increases, and finally it was studied the influence of the length, by considering a longer ramp.

Figure 109 show the shapes of each case studied.

The effect of the variation of the ramp on the filtering is shown in Figure 110, Figure 111 and Figure 112, where the pulley spin speeds, relative to each change, are plotted.

The responses of the pulley are not affected by the new shapes of the ramp.

On the other hand, the advantages gave by such alteration, concerns the internal impacts.

Indeed, as shown in Figure 110, Figure 111 and Figure 112, the amplitude of the impacts, between pulley and main spring, reduce.
Figure 102. Influence of the clearance G4 on the impact between pulley and the main spring during the run up with accessories excluded

Figure 103. Influence of the clearance G4 on the impact between pulley and the main spring during the run up with accessories included
Figure 104. Influence of the clearance G4 on the impact between pulley and the main spring during the start and stop of the engine.

Figure 105. Influence of the clearance G4 on the impact between pulley and the secondary spring $k_{ss2}$ during the run up with accessories excluded.
Figure 106. Influence of the clearance G4 on the impact between pulley and the secondary spring $k_{ss2}$ during the run up with accessories included.

Figure 107. Influence of the clearance G4 on the impact between pulley and the secondary spring $k_{ss2}$ during the start and stop of the engine.
Figure 108. Shape of the ramp in the original design

Figure 109. Shapes of the modified ramps
Figure 110. Influence of the ramp on the pulley spin speed during the run up with accessories excluded

Figure 111. Influence of the ramp on the pulley spin speed during the run up with accessories included
Figure 112. Influence of the ramp on the pulley spin speed during the start and stop of the engine

Figure 113. Influence of the ramp on the force G1 during the run up with accessories excluded
Figure 114. Influence of the ramp on the force G1 during the run up with accessories included

Figure 115. Influence of the ramp on the force G1 during the start and stop of the engine
The Sensitivity Analysis, which was performed in the previous chapter, shows the influence of limited number of parameters on the output of the system. The information so obtained might be used to improve the functioning of the system. In this final chapter, two possible solutions will be explained. The aim of such solutions is to solve the issue which affect the actual device.

7.1 First solution

The first solution proposed, includes a ball bearing instead of the journal bearing.

In this way, by replacing the bushing with the bearing NSK 6910 DDU, the friction T2, which is proportional to the radial load, decreases sharply.

This reduction, which is greater when the accessories are off, causes the system to oscillate more than the original design. Therefore, to counterbalance this diminution, the friction T2 is increased by adding a constant contribute, which is equal to 5 Nm. Such constant value of friction, might be exerted on the pulley by a cover, which is fixed to the hub, and a cone spring, which is interposed between cover and pulley.

The changes listed above, guarantee an acceptable filtering and make the assembly of the DDOS easier. Indeed, the ball bearing allow to mount hub and pulley coaxially without clearance, which, like in the original design, generate noise during the functioning.
However, the current design does not attenuate the noise generated during the impacts between the pulley and the main spring. Such noise is proportional to the kinetic energy of the system when the pulley hits the spiral spring.

Thus, the noise might be attenuated by dissipating the kinetic energy before the collision. A feasible solution is represented by the increment of the friction $T_1$. This effect can be obtained by changing the shape, the length and the slope of the ramp seated before the cog.

The modified ramp is shown in Figure 116 together with the original ramp.

![Figure 116. Original and modified ramps](image)

The response of the pulley, compared to the nominal case, is shown in Figure 117, Figure 118 and Figure 119, for run up with accessories off, run up with accessories on and start and stop, respectively.
The graphs show that both the filtering, whether the accessories are on or off, and the oscillations at the start are almost the same of the original design.

On the other hand, the amplitude of the impacts reduced, as Figure 120, Figure 121 and Figure 122 show.

Finally, Figure 123 and Figure 124, show the filtering throughout the ramp whether the accessories are on or off. Indeed, in such figures are plotted the velocities of the crankshaft and the pulley.

From the graph, it was possible to obtain the filtering ratio at different values of speed. Such filtering ratio are collected in Table VII and Table VIII.
Figure 118. Pulley speed of the modified DDOS with accessories included

Figure 119. Oscillations of the modified DDOS at the Start
Figure 120. Amplitude of the internal impacts of the modified DDOS with accessories excluded

Figure 121. Amplitude of the internal impacts of the modified DDOS with accessories included
Figure 122. Amplitude of the internal impacts of the modified DDOS at the Start

Figure 123. Filtering of the first solution when the accessories are excluded
Figure 124. Filtering of the first solution when the accessories are included

Figure 125. Filtering ratio at 1000 rpm with accessories excluded
Figure 126. Filtering ratio at 2000 rpm with accessories excluded

Figure 127. Filtering ratio at 3000 rpm with accessories excluded
Figure 128. Filtering ratio at 1000 rpm with accessories included.

Figure 129. Filtering ratio at 2000 rpm with accessories included.
### TABLE VII

FILTERING RATIO OF THE FIRST SOLUTION, ACCESSORIES EXCLUDED

<table>
<thead>
<tr>
<th>Rotary speed [rpm]</th>
<th>filtering original design</th>
<th>filtering current solution</th>
</tr>
</thead>
<tbody>
<tr>
<td>1000</td>
<td>1.75%</td>
<td>1.75%</td>
</tr>
<tr>
<td>2000</td>
<td>0.27%</td>
<td>0.25%</td>
</tr>
<tr>
<td>3000</td>
<td>0.042%</td>
<td>0.05%</td>
</tr>
</tbody>
</table>

### TABLE VIII

FILTERING RATIO OF THE FIRST SOLUTION, ACCESSORIES INCLUDED

<table>
<thead>
<tr>
<th>Rotary speed [rpm]</th>
<th>filtering original design</th>
<th>filtering current solution</th>
</tr>
</thead>
<tbody>
<tr>
<td>1000</td>
<td>2.35%</td>
<td>1.75%</td>
</tr>
<tr>
<td>2000</td>
<td>0.55%</td>
<td>0.25%</td>
</tr>
<tr>
<td>3000</td>
<td>0.13%</td>
<td>0.05%</td>
</tr>
</tbody>
</table>

### 7.2 Second Solution

The second solution proposed concerns the presence of a spring, with a relative low stiffness, which engages the pulley before it hits the free edge of the main spring.

This solution allows to reduce the noise due to the internal collisions, since a big amount of kinetic energy is stored by the pre-engagement spring in the form of elastic energy.

In order to simulate the presence of such spring, one of the two secondary springs, $k_{ss2}$ was assumed to be the pre-engagement spring.
The spring was set to be 50 Nm/rad stiff, and the angular distance between such spring and the pulley was set in such a way that $k_{ss2}$ engages the pulley 2 before the main spring.

The ball bearing is still present as well as the constant contribute of friction due to the cone spring.

Conversely, the ramp is not higher and longer than the original ramp, like in the previous solution, but is only linear. The reason, is that, in this solution, the ramp does not have to dissipate kinetic energy, since the impact is attenuated by the spring.

The results, in term of filtering, are shown in Figure 131, Figure 132 and Figure 133, where the pulley spin speed, of the modified DDOS, is plotted together with the results of the original design.
The filtering throughout the ramp, whether the accessories are on or off, improved slightly. On the other hand, the oscillations of the pulley speed at the start increased, in comparison to the original design.

However, this solution is interesting because it allows to completely avoid the noise due to the impact between pulley and main spring. Indeed, when the pulley hit the spiral spring it has already lost most of its kinetic energy to compress the secondary spring $k_{ss2}$. From Figure 134 to Figure 136 show the amplitude of the force $G_1$, exerted during the impacts.
Figure 132. Pulley speed of the DDOS with the pre-engagement spring when the accessories are included

Figure 133. Pulley speed of the modified DDOS with the pre-engagement spring at the start
The force is always null except during the run up with accessories on. Indeed, in this case, below a certain value of speed, the secondary spring $k_{ss2}$ is able to provide enough torque to drive the accessories, while, over such threshold, the main spring engages the pulley and starts working.

Finally in Figure 137 and Figure 138 are plotted the rotary speed of the crankshaft and the pulley throughout the ramp when the accessories are off and on respectively.

The filtering ratio of the current solution for different values of speed are collected in Table VII and Table VIII, respectively, when the accessories are off and on.
Figure 135. Amplitude of the impacts in the DDOS with the pre-engagement spring when the accessories are excluded

Figure 136. Amplitude of the impacts in the DDOS with the pre-engagement spring at the start
Figure 137. Filtering of the current solution when the accessories are excluded

Figure 138. Filtering of the current solution when the accessories are included
Figure 139. Filtering ratio of the second solution at 1000 rpm with accessories excluded

Figure 140. Filtering ratio of the second solution at 2000 rpm with accessories excluded
Figure 141. Filtering ratio of the second solution at 3000 rpm with accessories excluded

Figure 142. Filtering ratio of the second solution at 1000 rpm with accessories included
Figure 143. Filtering ratio of the second solution at 2000 rpm with accessories included

Figure 144. Filtering ratio of the second solution at 3000 rpm with accessories included
### TABLE IX

FILTERING RATIO OF THE SECOND SOLUTION, ACCESSORIES EXCLUDED

<table>
<thead>
<tr>
<th>Rotary speed [rpm]</th>
<th>filtering original design</th>
<th>filtering current solution</th>
</tr>
</thead>
<tbody>
<tr>
<td>1000</td>
<td>1.75%</td>
<td>1.5%</td>
</tr>
<tr>
<td>2000</td>
<td>0.27%</td>
<td>0.23%</td>
</tr>
<tr>
<td>3000</td>
<td>0.042%</td>
<td>0.03%</td>
</tr>
</tbody>
</table>

### TABLE X

FILTERING RATIO OF THE SECOND SOLUTION, ACCESSORIES INCLUDED

<table>
<thead>
<tr>
<th>Rotary speed [rpm]</th>
<th>filtering original design</th>
<th>filtering current solution</th>
</tr>
</thead>
<tbody>
<tr>
<td>1000</td>
<td>2.35%</td>
<td>1.75%</td>
</tr>
<tr>
<td>2000</td>
<td>0.55%</td>
<td>0.3%</td>
</tr>
<tr>
<td>3000</td>
<td>0.13%</td>
<td>0.06%</td>
</tr>
</tbody>
</table>
Ultimately, over the thesis I went through the developing of a numerical model and its validation, the sensitivity analysis of such a model and the optimization of its parameters in order to improve the performance of the mechanical device.
APPENDICES
Appendix A

VISUAL BASIC FUNCTION TO PLOT THE STATIC CHARACTERISTIC

Option Explicit

Function force(x As Double, x0 As Double, a As Double, b As Double, c As Double, D As Double, k As Double, f As Double) As Double
    Dim s As Double, a1 As Double
    a1 = f / k
    s = x - x0

    If (s > 0) Then
        If (x < -a - c - (a + c) / 100000) Then
            force = -10000
        ElseIf (x < -c + a1) Then
            force = k * (x + c)
        ElseIf (x < D + a1) Then
            force = f
        ElseIf (x < D + b + (D + b) / 100000) Then
            force = -10000
        End If
    Else
        If (x < 0) Then
            force = k * x
        Else
            force = 0
        End If
    End If
End Function
Appendix A (Continued)

    force = k * (x - D)

Else

    force = 10000

End If

Else

    If (x > D + b + (D + b) / 100000) Then

        force = 10000

    ElseIf (x > D - a1) Then

        force = k * (x - D)

    ElseIf (x > -c - a1) Then

        force = -f

    ElseIf (x > -a - c - (a + c) / 100000) Then

        force = k * (x + c)

    Else

        force = -10000

    End If

End If

End If

End Function
Public Function Radial_load(x As Double, x0 As Double, 
D As Double, rb As Double, Rp As Double, f As Double, 
T2 As Double, alpha As Double)

Dim a As Double, b As Double, c As Double, s As Double

a = (1 - f ^ 2 * rb ^ 2 / Rp ^ 2)
b = (-2 * D - 2 * T2 * f ^ 2 * rb ^ 2 * (1 - Cos(alpha)) / Rp)
c = (D ^ 2 - 2 * T2 ^ 2 * f ^ 2 * rb ^ 2 * (1 - Cos(alpha)))

s = x - x0

If s > 0 Then
    Radial_load = (-b + (b ^ 2 - 4 * a * c) ^ 0.5) / 2 / a
Else
    Radial_load = (-b - (b ^ 2 - 4 * a * c) ^ 0.5) / 2 / a
End If
Appendix A (Continued)

End Function

Public Function Hl(c As Double, alpha As Double, Rp As Double, T2 As Double) As Double

    Hl = ((c / Rp + T2) ^ 2 + T2 ^ 2 - 2 * (c / Rp + T2) * T2 * Cos(alpha)) ^ (0.5)

End Function

Public Function bushing_torque(H As Double, f As Double, rb As Double, x As Double, x0 As Double) As Double

    Dim s As Double

    s = x - x0

    If (s > 0) Then
        bushing_torque = H * f * rb
    Else
        bushing_torque = -H * f * rb
    End If
Appendix A (Continued)

End If

End Function
Appendix B

LOOK-UP TABLE FUNCTIONS

Option Explicit

Public Function \( T_1(v \text{ As Double}, k \text{ As Double}) \text{ As Double} \)

Dim s As Double

\( s = 1 / k \)

If (\( v \leq -s \)) Then
\( T_1 = -1 \)
ElseIf (\( v \leq s \)) Then
\( T_1 = k \times v \)
Else
\( T_1 = 1 \)
End If

End Function
Public Function G_1(x As Double, G1 As Double, k1 As Double) As Double

If (x < G1) Then
    G_1 = 0
Else
    G_1 = k1 * (x - G1)
End If

End Function

Public Function G_2(x As Double, G2 As Double, G3 As Double, k2 As Double) As Double

If (x < -G3) Then
    G_2 = k2 * (x + G3)
ElseIf (x < G2) Then
    G_2 = 0
End If
Else
G_2 = k2 * (x - G2)
End If

End Function

Public Function Hubload(F1 As Double, F2 As Double, 
alpha As Double) As Double
Hubload = (F1 ^ 2 + F2 ^ 2 - 2 * F1 * F2 * Cos(alpha)) ^ 0.5
End Function

Public Function G_4(x As Double, G4 As Double, G5 As Double, 
k2 As Double) As Double
If (x < -G5) Then
G_4 = k2 * (x + G5)
ElseIf (x < G4) Then
Appendix B (Continued)

\[ G_4 = 0 \]

\textbf{Else}

\[ G_4 = k2 \times (x - G4) \]

\textbf{End If}

\textbf{End Function}

Public Function Ramp(R As Double, Rp As Double, k As Double, f As Double) as Double

\[ \text{Ramp} = f \times k \times (Rp - R) \times R \]

\textbf{End Function}
Appendix C

**DDOS_DATA**

The M-file where all the data of the system are collected is listed below.

```matlab
%%DDOS_data
clear all
close all
clc

%%Data of the system from excell sheet
%%Geometrical and physical data of the components

%%Main spring

%%Equivalent inertia of the main spring [Kgm2]
Ims=xlsread('DDOS_9','Data','c5');

%%main spring stiffness[Nm/rad]
kms=xlsread('DDOS_9','Data','c7');

%%main spring equivalent viscous damping[Nms/rad]
cms=xlsread('DDOS_9','Data','c8');
```
%main spring tangential stiffness[N/m]

k_tau=xlsread('DDOS_9','Data','c9');

%sliding pad

%dry friction coefficient between pad and pulley

fsp=xlsread('DDOS_9','Data','c10');

%dry friction between pad and pulley[Nm]

T1=xlsread('DDOS_9','Data','c11');

%Secondary spring1

%inertia of the pinhead [Kgm2]

Iph1=xlsread('DDOS_9','Data','c19');

%secondary spring stiffness[Nm/rad]

kss1=xlsread('DDOS_9','Data','c15');

%secondary equivalent viscous damping[Nms/rad]

css1=xlsread('DDOS_9','Data','c16');

%Secondary spring2
Appendix C (Continued)

%inertia of the pin head [Kgm2]
Iph2=xlsread('DDOS_9','Data','c27');

%secondary spring stiffness [Nm/rad]
kss2=xlsread('DDOS_9','Data','c23');

%secondary equivalent viscous damping [Nms/rad]
css2=xlsread('DDOS_9','Data','c24');

%Journal bearing

%radius of the bushing [m]
rb=xlsread('DDOS_9','Data','c29');

%dry friction coefficient between pad and pulley
fb=xlsread('DDOS_9','Data','c30');

%Pulley

%external radius of the pulley [m]
Rpi=xlsread('DDOS_9','Data','c32');

%internal radius of the pulley [m]
Rpe=xlsread('DDOS_9','Data','c34');

%inertia of the pulley [Kgm2]
IP=xlsread('DDOS_9','Data','c35');
%equivalent viscous damping[Nms/rad]
ceq=xlsread('DDOS_9','Data','c36');
%equivalent inertia of the pulley[Kgm2]
IPeq=xlsread('DDOS_9','Data','c46');
%wrap angle of the pulley[deg]
alpha=xlsread('DDOS_9','Data','c48');
%tension imposed by the tensioner[N]
lowtensionbelt=xlsread('DDOS_9','Data','c49');

%M,C,K matrices in the configuration space

M=[Ims 0 0 0
   0 Iph1 0 0
   0 0 Iph2 0
   0 0 0 IPeq];
Appendix C (Continued)

\[ \text{Cdamp=} \begin{bmatrix} \text{cms} & 0 & 0 & 0 \\ 0 & \text{css1} & 0 & 0 \\ 0 & 0 & \text{css2} & 0 \\ 0 & 0 & 0 & \text{ceq} \end{bmatrix}; \]

\[ \text{K=} \begin{bmatrix} \text{kms} & 0 & 0 & 0 \\ 0 & \text{kss1} & 0 & 0 \\ 0 & 0 & \text{kss2} & 0 \\ 0 & 0 & 0 & 0 \end{bmatrix}; \]

\[ \text{F}_B= \begin{bmatrix} \text{cms} & \text{kms} & 0 & -1 & 0 & -1 & 0 & 0 \\ \text{css1} & \text{kss1} & 0 & 0 & 0 & 0 & -1 & 0 \\ \text{css2} & \text{kss2} & 0 & 0 & 0 & 0 & 0 & -1 \\ \text{ceq} & 0 & -1 & 1 & 1 & 1 & 1 & 1 \end{bmatrix}; \]

\%State-space matrices

\[ \text{A=} \begin{bmatrix} -\text{inv(M)}*\text{Cdamp} & -\text{inv(M)}*\text{K} \end{bmatrix}; \]
Appendix C (Continued)

\[
\begin{bmatrix}
1 & 0 & 0 & 0 \\
0 & 1 & 0 & 0 \\
0 & 0 & 1 & 0 \\
0 & 0 & 0 & 1
\end{bmatrix}
\begin{bmatrix}
0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0
\end{bmatrix} = 0.
\]

\[B = \text{inv}(M) \cdot F_B = \begin{bmatrix}
0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & 0 & 0 & 0
\end{bmatrix};\]

\[C = \begin{bmatrix}
1 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\
0 & 1 & 0 & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & 1 & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 1 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 1 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & 1 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & 0 & 1 & 0
\end{bmatrix};\]
0 0 0 0 0 0 0 1];

D=zeros(8);

% lookup table

%G2
data=xlsread('DDOS_9','G2');
angle_G2=data(1:end,1)';
G2=data(1:end,2);

%G4
data=xlsread('DDOS_9','G4');
angle_G4=data(1:end,1)';
G4=data(1:end,2);

%G1
data=xlsread('DDOS_9','G1');
angle_G1=data(1:end,1)';
G1=data(1:end,2);

%T
Appendix C (Continued)

data=xlsread('DDOS_9','T');
angle_T=data(1:end,1)';
T=data(1:end,2);

%A/C torque

data=xlsread('DDOS_9','A|C');
speed_AC=data(1:end,1)';
Torque_AC=data(1:end,2);

%A/C torque

data=xlsread('DDOS_9','Alt');
speed_Alt=data(1:end,1)';
Torque_Alt=data(1:end,3);

%wp torque

data=xlsread('DDOS_9','wp');
speed_wp=data(1:end,1)';
Torque_wp=data(1:end,6);
Appendix C (Continued)

i_Alt=xlsread('DDOS_9','Data','c64');  \%Alternator gear ratio
i_AC=xlsread('DDOS_9','Data','c66');  \%Air-conditioning gear ratio
i_wp=xlsread('DDOS_9','Data','c65');  \%water pump gear ratio

\%Ramp

data=xlsread('DDOS_9','ramp');

angle_DeltaT1=data(1:end,2);

DeltaT1=data(1:end,4);
Appendix D

INFLUENCE OF G5

G5 is the angular distance between the pulley and the secondary spring in over-running. Since the secondary springs never engages in the original design, G5 influences the behavior of the DDOS in only it decreases.

In figures Figure 145, Figure 146 and Figure 147, is plotted the Pulley spin speed for different values of G5.

The filtering is not affected by this parameter whether the accessories are off or on. Conversely, the response of the pulley at the start worsen slightly as G5 goes down. The oscillations are better shown in figure Figure 148.

The analysis of the force G1 yields to results very close to those observed in the study of the influence of G4.

Figures Figure 149 and Figure 150 are the graphs of G1 respectively where the accessories are off and on during the run up of the engine.

The effect of the variation does not affect the amplitude of the impacts. However, the reduction of G5 corresponds to an increment of the contacts, at least when the accessories are excluded.

On the other hand, when the DDOS is transferring torque to the front-end accessory, the pulley is always engaged.
The variation of G5, instead, makes both numbers and amplitude, of the impacts during the start, bigger, as figure Figure 151 shows clearly.

The study of the force G4 explains the presence of such impacts.

Indeed, while the graph of G4 is always null during the run up, it has several negative impulses at the start. It means that the secondary springs $k_{ss2}$ engages several times in over-running during the start. This trend is shown in figure Figure 152.

Figure 145. Influence of the clearing G5 on the filtering during the run up with accessories off
Figure 146. Influence of the clearing G5 on the filtering during the run up with accessories on

Figure 147. Influence of the clearing G5 on the filtering during the start and stop of the engine
Appendix D (Continued)

Figure 148. Zoom in on the influence of the clearance G5 at the start

Figure 149. Influence of the clearance G5 on the impact between pulley and the main spring during the run up with accessories off
Appendix D (Continued)

Figure 150. Influence of the clearance G5 on the impact between pulley and the main spring during the run up with accessories on

Figure 151. Influence of the clearance G5 on the impact between pulley and the main spring during the start and stop of the engine
Figure 152. Influence of the clearance G5 on the impact between pulley and the secondary spring $k_{ss2}$ during the start and stop of the engine
CITED LITERATURE


VITA

NAME: Stefano Medei

DATE OF BIRTH: 05/17/1987

PLACE OF BIRTH: Treia (MC)

CITIZENSHIP: Italian

EDUCTION: B.S. in Mechanical Engineering, Polytechnic of Turin, 1st Faculty of Engineering, 2009

M.S. in Mechanical Engineering, Polytechnic of Turin, 1st Faculty of Engineering, 2011

AWARDS: EDISU Fellowship, Torino, Italy, 2006-2011