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1 **A COMBINED NUMERICAL APPROACH FOR THE THERMAL ANALYSIS**
2 **OF A PISTON WATER PUMP**

3
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7 **ABSTRACT**

8 The paper proposes a numerical model for the investigation of a piston water pump
9 under different operating conditions. In particular, the lubricating system is analysed
10 and modelled. The study accounts for the lubrication and friction phenomena, heat
11 transfer, multiphase fluid approach and motion simulation.

12 A computational thermo fluid dynamics approach has been adopted to develop a
13 numerical tool able to simulate the behaviour of the oil during the machine working
14 phases. The CFD approach simulates the moving metal components by means of
15 moving meshes techniques; the friction phenomenon is estimated on the basis of
16 formulations available in literature. The numerical model evaluates the heat transfer
17 between moving metal parts and oil during the operating phases of the system.
18 Furthermore, the heat transfer between oil and environment is calculated, accounting for
19 conduction through the metal crankcase walls. A multiphase fluid approach is used for
20 the simulation of the oil and air mixing during the crank rotation.

21 The heat transfer coefficient predicted by the CFD approach are employed in a lumped
22 and distributed numerical model; the reliability and accuracy of the proposed numerical
23 approach is addressed and validated against experimental results. Experimental data
24 have been collected by means of a thermographic camera and thermocouples. Finally,
25 the tool's predictive capabilities are addressed by simulating different working
26 conditions.

27 **KEYWORDS:** heat transfer, friction, piston water pump, CFD, lumped parameter,
28 moving mesh.

29 **1. INTRODUCTION**

30 Water piston pumps are largely employed in many industrial applications, but they are
31 mainly used in the urban sector, fitted on drain cleaning trucks, waste bin washers and
32 road sweepers. Bigger size pumps are used for ship keels cleaning. Due to the machine
33 versatility, the pump can operate continuously or not; in addition, it can be used in cold
34 country as well as in hot places. Thus, it is fundamental to project the machine in order
35 to ensure the necessary heat transfer from the internal moving parts to the external
36 environment. Therefore, overheating must be avoided to guarantee the performance and
37 the lifetime of the pump itself for each working condition. A key role is played by
38 lubricant oil which is the transfer fluid that transmit the heat from cranks, rods, pistons
39 and crankshaft to the crankcase walls.

40 A great support can be offered to engineers by numerical simulations, in order to predict
41 heat transfer for different working conditions of many systems and components. In
42 particular, computational fluid dynamics models are largely employed to describe the
43 thermo fluid dynamics behaviour of various machines. Bhutta et al. [1] presented a
44 complete review of CFD analysis of heat exchangers. Different turbulence models and

45 velocity-pressure coupling schemes have been compared, for a wide variety of heat
46 exchanger architectures. In this regard, H. Mroue et al. [2] investigated the performance
47 of a heat exchanger equipped with six thermosyphons by means of a CFD approach
48 without simulating the two-phase change that occurs inside the thermosyphons; an
49 overview of numerical models used to investigate the condensation, evaporation and
50 boiling in these systems can be found in [3].

51 Within the context of centrifugal pumps, a critical review of different CFD models has
52 been presented by Shah et al. [4] in order to outline the most interesting areas of
53 research to improve the pump performance: cavitation analysis, diffuser pump analysis,
54 volute flow study and impeller-volute interaction.

55 In order to simulate the thermo fluid dynamics behaviour of the oil inside the crankcase,
56 different phenomena must be accounted for. Firstly, attention should be devoted to the
57 movement description of cranks, rods, pistons, and crankshaft that caused the oil-air
58 mixing (splash lubrication). Moving meshes give the possibility to the user to include
59 moving parts, based on equations well known in literature [5, 6]. This numerical
60 technique is expensive in terms of computational resources, but it ensures good
61 accuracy in modelling moving parts and solid fluid moving interfaces. Menéndez
62 Blanco and Fernández Oro [7], for instance, used this numerical approach to construct a
63 model of an air-operated piston pump for lubricating greases. Subsequently, it is
64 necessary to calculate the thermal energy introduced into the system by friction.
65 Different approaches can be found in literature [5, 6, 8, 9, 10, 11], referred to analysis of
66 engine pistons. Indeed, piston water pumps and engines present similar architectures of
67 pistons, crank mechanisms, rings. In particular, detailed descriptions of the friction
68 between the piston rings and the cylinder wall have been outlined by Cho and Moon [6]
69 and by Livanos and Kyrtatos [8], while Tateishi [9] proposed an empirical
70 approximation. One of the most applied equation in numerical modelling is the Chen
71 and Flynn correlation [10], used also by Hooper et al. [11] to successfully simulate a
72 stepped piston engine using one dimensional CFD approach. Once calculated the heat
73 released by friction, fluid properties and heat transfer models must be defined. The fluid
74 is described as a two phases mixture of air and oil; thus, the volume of fluid (VOF)
75 approach is used. Several examples of VOF simulations are available in literature
76 applied to different contexts. Jouhara et al. [12] simulated flow and heat transfer in a
77 thermosyphon: by means of VOF technique, evaporation and condensation were
78 accounted for as well as the interaction between gas and liquid. Lückmann et al. [13]
79 applied the numerical method to resolve the free-surface oil flow in a lubricant oil
80 pumping system of a reciprocating compressor.

81 In [14] a numerical approach has been used to predict the transient behaviour of a
82 lubrication in a wet clutch of a hydromechanical variable transmission; the volume of
83 fluid approach has been employed in the numerical model in order to determine the oil
84 distribution in the clutch region under different rotating velocities. A similar study was
85 conducted by Terzi et al [15] where a VOF approach has been used to determine the
86 lubrication flow within a multi-plate wet clutch.

87 Air and oil physical properties need to be updated on the basis of the temperature field:
88 while air property correlations are included in the library of the software, oil ones have
89 to be provided. Habchi et al. [16] developed and validated models of pressure and
90 temperature dependencies of standard oil properties. Heat transfer problems have been
91 widely simulated by means of numerical models, especially for heat exchangers [1].
92 Also heat transfer in cylinder walls has been largely studied: Rakopoulos et al. [17]

93 compared different heat transfer formulations. In this paper, dimensionless numbers [18,
94 19] are involved in correlations [20, 21] able to described the heat transfer coefficient in
95 a very simple way. Brucker and Majdalani [20] presented a comprehensive table of
96 Nusselt number expressions for different geometries, flow conditions and ranges of
97 validity. In particular, the equation proposed by Churchill and Chu [21] is used to
98 calculate the Nusselt number that characterized the heat transfer between the crankcase
99 walls and the environment. A similar approach has been successfully used by Bottazzi
100 et al. [22] to construct and to develop a numerical model able to simulate the thermo-
101 dynamics behaviour of a coffee roasting machine and, in particular, the heat transfer
102 from a hot air flow to coffee beans during toasting phases.

103 The aim of this study is the development of a numerical tool that can be used for the
104 investigation of lubricating system for piston water pump in order to design new
105 crankcase and to improve existing components.

106 Thus, the model is intended to predict the influence of the various parameters that
107 characterize the heat transfer between oil and metal parts, such as surface geometry,
108 temperature and oil mixing. The main goal of the numerical tool is to predict the
109 evolution of the temperature map in order to define the steady value for different
110 working conditions.

111 Finally, the accuracy of the numerical results of the proposed model are validated
112 against experimental data. The experimental measurements+ are collected by means of
113 thermocouples and a thermographic camera applied to a standard pump tested for
114 different working conditions.

115 2. CFD MODEL

116 Piston water pump are generally composed of three alternative pistons, with 120° of
117 angular displacement between each one. A complete numerical model of the pump can
118 be obtained joining three single models representing one piston. Thus, initially, a single
119 model regarding a crank, a rod and a piston is developed. Once prepared the geometry,
120 the mesh is constructed. As previously said, moving mesh technique is applied in order
121 to simulate the splash lubrication effects. Motion of each moving part need to be
122 modelled. Thus, energy dissipated due to the friction is estimated and introduced into
123 the system. For fluid modelling, a “Volume of Fluid” approach is used, in order to
124 describe the two phases mixture of oil and air. Oil properties are expressed as a function
125 of the temperature. Finally, heat transfer from metal moving part to the environment is
126 defined by means of dimensionless formulations. The implementation of all these
127 features is necessary in order to ensure a good accuracy of the model, but it determines
128 a high computational effort. In addition, the heat transfer phenomenon is a slow
129 mechanism that requires a long computational time. Thus, a 2D model is used in order
130 to obtain a model that can be usefully adopted by pump designers: indeed, the model
131 accuracy is important as well as the possibility to obtain the results in a reasonable time.
132 Once all the features are properly configured in the 3D single piston model, it is
133 possible to automatically scale from the 3D to a 2D model using a section plane that
134 includes the axis of the central piston and that is perpendicular to the pump base.

135

136 2.1 Motion model

137 The single piston model accounts for two moving parts: the rod and the piston. Both the
138 motions of the rod and the piston are simulated by means of moving mesh technique. In

139 the first case, two blocks are constructed: a fixed one, that is the void of the crankcase,
 140 and a moving one that accounts for the rod. Indeed, this last one is a box that includes
 141 the rod and that moves inside the fixed block. This movement is the rod motion and it is
 142 possible to define it with a geometrical analysis [5]. Referring to the layout of Fig. 1,
 143 rod position on a plane XY is described by Eq. 1 and Eq. 2:

$$144 \quad x_B = r_c \cdot \cos(\omega \cdot t - \pi) \quad (1)$$

$$145 \quad y_B = r_c \cdot \sin(\omega \cdot t - \pi)$$

146 (2)

147 where x and y are the position coordinates of the point B referred to the fixed system
 148 shown in Fig. 1; r_c is the eccentricity, i.e. the crank length, ω is the rotational velocity, t
 149 is the time, $-\pi$ is summed because the simulation starts when the piston is at the bottom
 150 dead centre (BDC). A roto-translation of rigid body is defined when the motion of a
 151 generic point J (Eq. 3) is known. Considering the generic point J , its movement respect
 152 the fixed coordinate system can be described as the vectoral sum of the translation
 153 velocity of a moving system and the rotational velocity referred to that system. The
 154 moving coordinate system is constructed with axes parallel to the ones of the fixed
 155 system and origin in B . The rotational axis coincides to the Y axis of the moving system.

$$156 \quad \vec{v}_{totJ} = \vec{v}_{transB} + \vec{\beta}_B \times \overline{BJ} \quad (3)$$

157 The components of the translational velocity of the moving system are expressed by Eq.
 158 4 and Eq. 5, while the angular velocity is calculated with Eq. 6.

$$159 \quad \dot{x}_B = -\omega \cdot r_c \cdot \sin(\omega \cdot t - \pi) \quad (4)$$

$$160 \quad \dot{y}_B = \omega \cdot r_c \cdot \cos(\omega \cdot t - \pi) \quad (5)$$

$$161 \quad \dot{\beta}_B = \lambda \cdot \omega \cdot \left(\frac{\cos(\omega \cdot t)}{\sqrt{1 - \lambda^2 \cdot \sin^2(\omega \cdot t)}} \right) \quad (6)$$

162 Deriving Eq. 1 and Eq. 2 respect time, it is possible obtain Eq. 3 and Eq. 4. With simple
 163 mathematical steps (see Appendix A) it is possible to determine Eq. 6, where λ is the ratio
 164 between crank and rod length. The software automatically applies Eq. 3 to all the cells of
 165 the moving mesh, once introduced Eq. 4 and Eq. 5 and Eq. 6.

166 Once the rod movement is detailed, piston motion is simulated. To do that, a second
 167 overset mesh is configured. As done for the rod, the void of the crankcase is used as
 168 fixed block, while a moving block accounts for the piston. The motion is a translation of
 169 a rigid body and it can be described on the basis of the motion equation of the small end
 170 connecting rod (point A in Fig. 1) obtained from the analysis of a generic crank-rod
 171 mechanism (see Appendix A).

$$172 \quad \dot{x}_A = r_c \cdot \omega \cdot \left[\sin \alpha + \frac{\lambda \cdot \sin 2\alpha}{2 \cdot \sqrt{1 - \lambda^2 \cdot \sin^2 \alpha}} \right] \quad (7)$$

173 The two moving blocks of the overset mesh zones are overlapping each other; thus, a
 174 third overset interface has to be configured in order to assign the correct behaviour to
 175 each cell that are positioned in the overlapping region between rod and piston blocks.

176

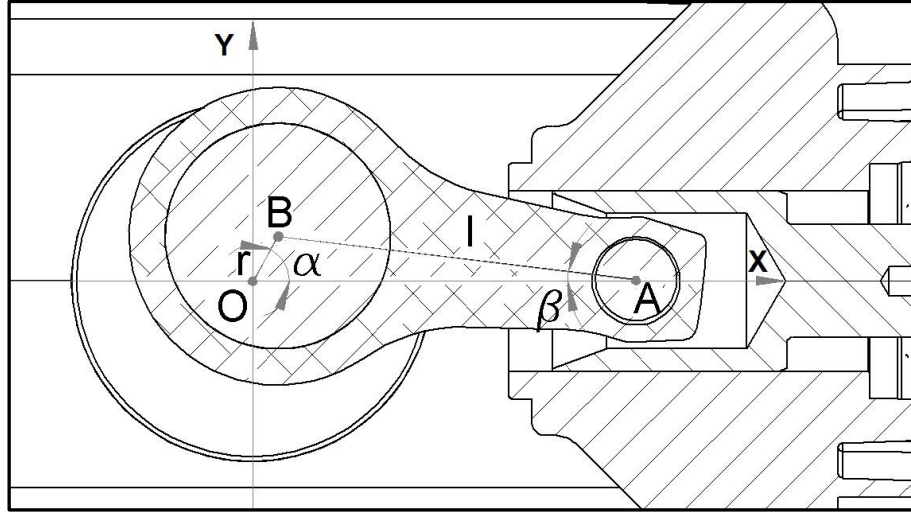


Fig. 1. Crank-rod mechanism, reference system.

2.2 Friction model

177

178

179

180 Friction analysis has a key role to assess the dissipated energy. It is very useful to assess
 181 at each contact surface the amount of energy that is released as heat. Unfortunately, no
 182 studies are available in literature that investigate friction evaluation on a piston water
 183 pump. On the contrary, there are some interesting works accounting for friction on an
 184 internal combustion engine [5, 6, 8, 9, 10, 11]. Water piston pumps and engines
 185 presented a quite similar architecture, in terms of crank-connecting rod mechanism,
 186 piston and rings. Dissipated energy due to friction in an engine, is frequently calculated
 187 as a whole, on the basis of energy balance, but in a few cases it is possible to found
 188 approximated correlation regarding the various contact surfaces.

189 According to Heywood [5], Eq. 8 can be used to calculate the friction force F_{f_rbe}
 190 referred to the contact between the connecting rod big end and the crankshaft, under the
 191 hypothesis of continuous oil film between the surfaces:

$$192 \quad F_{f_rbe} \approx (\pi \cdot d_{rbe} \cdot l_{rbe}) \cdot \mu_{oil} \cdot \left(\frac{\pi \cdot d_{rbe} \cdot \omega}{\bar{c}_{rbe}} \right) = \frac{\mu_{oil} \cdot \pi^2 \cdot d_{rbe}^2 \cdot l_{rbe} \cdot \omega}{\bar{c}_{rbe}} \quad (8)$$

193 Where d_{rbe} is the internal diameter of the connecting rod big end, l_{rbe} is the contact
 194 length (thickness of connecting rod big end), μ_{oil} is the oil dynamics viscosity, \bar{c}_{rbe} is
 195 the mean radial clearance. Once obtained the friction force by means of this
 196 approximated approach, it is possible to determine the related friction torque M_{f_rbe} (Eq.
 197 9) and the dissipated power P_{f_rbe} (Eq. 10):

$$198 \quad M_{f_rbe} = F_{f_rbe} \cdot d_{rbe} / 2 \quad (9)$$

$$199 \quad P_{f_rbe} = M_{f_rbe} \cdot \omega \quad (10)$$

200 The same approach can be applied to assess the dissipated power P_{f_rse} referred to the
 201 contact surface between the connecting rod small end and the pin. This contact, in fact,
 202 presents a similar geometry configuration (cylinder vs. cylinder contact surface) and an
 203 analogous lubrication condition. Another important contribution to the energy
 204 dissipation is the friction between the crankshaft and the two needle bearings. An
 205 approximated method to choose size the component is provided by producers. Each
 206 needle bearing supports a force F_{f_nb} :

$$207 \quad F_{f_nb} = p_{max} \cdot (\pi \cdot d_p^2 / 4) \cdot n_p / 2 \quad (11)$$

208 where p_{max} is the water maximum pressure, d_p the piston diameter, n_p the number of
 209 pistons. Obviously, the force is divided by two because there are two needle bearings.
 210 The related friction torque M_{f_nb} can be calculated by means of Eq.12:

$$211 \quad M_{f_nb} = 0.5 \cdot C_f \cdot d_{nb} \cdot F_{f_nb} \quad (12)$$

212 where C_f is the constant friction coefficient, value characteristic of the bearing
 213 architecture and tabulated by the producers, d_{bn} is the internal diameter of the
 214 component (where the crankshaft is connected). Thus, the related dissipated power can
 215 be assessed by Eq. 10.

216 In order to complete the friction evaluation, two additional dissipated power terms have
 217 to be accounted for: the first one is due to the contact surface between the seal placed in
 218 the cylinder wall and the ceramic piston part (P_{f_ring}) and the second one is referred to
 219 the friction between the journal box and the piston (P_{f_jb}). To calculate these two terms,
 220 several approaches referred to engine pistons are available in literature [8, 9] but in this
 221 case to consider the piston water pump as an engine is a poor approximation, due to the
 222 different ring kind and number for each piston and due to the different pressure curve
 223 during the cycle. A different approach can be based on efficiency analysis. The ratio
 224 between hydraulic (P_{hyd}) and mechanical (P_{mech}) power is the total efficiency of the
 225 pump η_{tot} :

$$226 \quad \eta_{tot} = \frac{P_{hyd}}{P_{mech}} = \frac{Q \cdot (p_{max} - p_{suc})}{\omega \cdot M_{max}} = \eta_{vol} \cdot \eta_{hm} \quad (13)$$

227 where Q is the flow rate, p_{suc} the pressure at the pump suction and M_{max} the maximum
 228 torque, referred to the maximum pressure p_{max} . The total efficiency η_{tot} is equal to the
 229 product of volumetric efficiency η_{vol} and hydromechanical efficiency η_{hm} :

$$230 \quad \eta_{vol} = Q / [3 \cdot \omega \cdot (2 \cdot r_c) \cdot (\pi \cdot d_p^2 / 4)] \quad (14)$$

231 while η_{vol} is defined as the ratio between the flow rate and the ideal geometrical flow
 232 rate, η_{hm} can be obtained on the basis of experimental data combining Eq. 14 with Eq.
 233 13. The total dissipated power P_{f_tot} can be calculated as:

$$234 \quad P_{f_tot} = P_{hyd} \cdot (1 - \eta_{hm}) = 2 \cdot P_{f_nb} + 3 \cdot (P_{f_rbe} + P_{f_rse} + P_{f_jb} + P_{f_ring}) \quad (15)$$

235 afterwards, subtracting the previously calculated terms of dissipated power, it is
 236 possible to estimate the sum of the two investigated terms. Based on producer's know-
 237 how, the ratio between the terms is fixed: thus, P_{f_ring} and P_{f_jb} can be separately
 238 assessed.

239 The total friction losses on the piston is calculated by both the proposed approach and
 240 the Chen and Flynn correlation [10] and the results are compared as a check. This
 241 empirical correlation is one of most used technique to estimate the total dissipated
 242 power due to the friction in combustion chamber simulation. Both the approaches
 243 provide results of the same order.

244 In the constructed model, the dissipated power terms are included as thermal flux from
 245 the contact surface to the fluid. The rod is made of aluminium, that is a good conductor;
 246 thus, the hypothesis of uniform energy distribution can be assumed and both P_{f_rbe} and
 247 P_{f_rse} are addressed to the external rod surface. A uniform energy distribution is also

248 supposed assigning P_{f_jb} to the part of the cylinder internal wall that is immersed in oil.
 249 P_{f_ring} is referred to the cylinder and the piston parts those work in contact with water
 250 and do not influence the oil behaviour: thus, P_{f_ring} is not included in the numerical
 251 model. P_{f_nb} , instead, must be accounted for in the overall numerical model of the pump
 252 but not in the single piston model.

253

254 2.3 Fluid model

255 The fluid inside the crankcase is modelled as a multiphase non reacting mixture by
 256 means of the Volume of Fluid approach. The spatial distribution of each phase at a
 257 given time is defined in terms of volume fraction. The Segregated Flow model is used
 258 to solve the conservation equations separated for each phase, except for the pressure
 259 field which is common. In this study, also the temperature field has to be accounted for;
 260 the model used is the Segregated Multi-Phase Temperature.

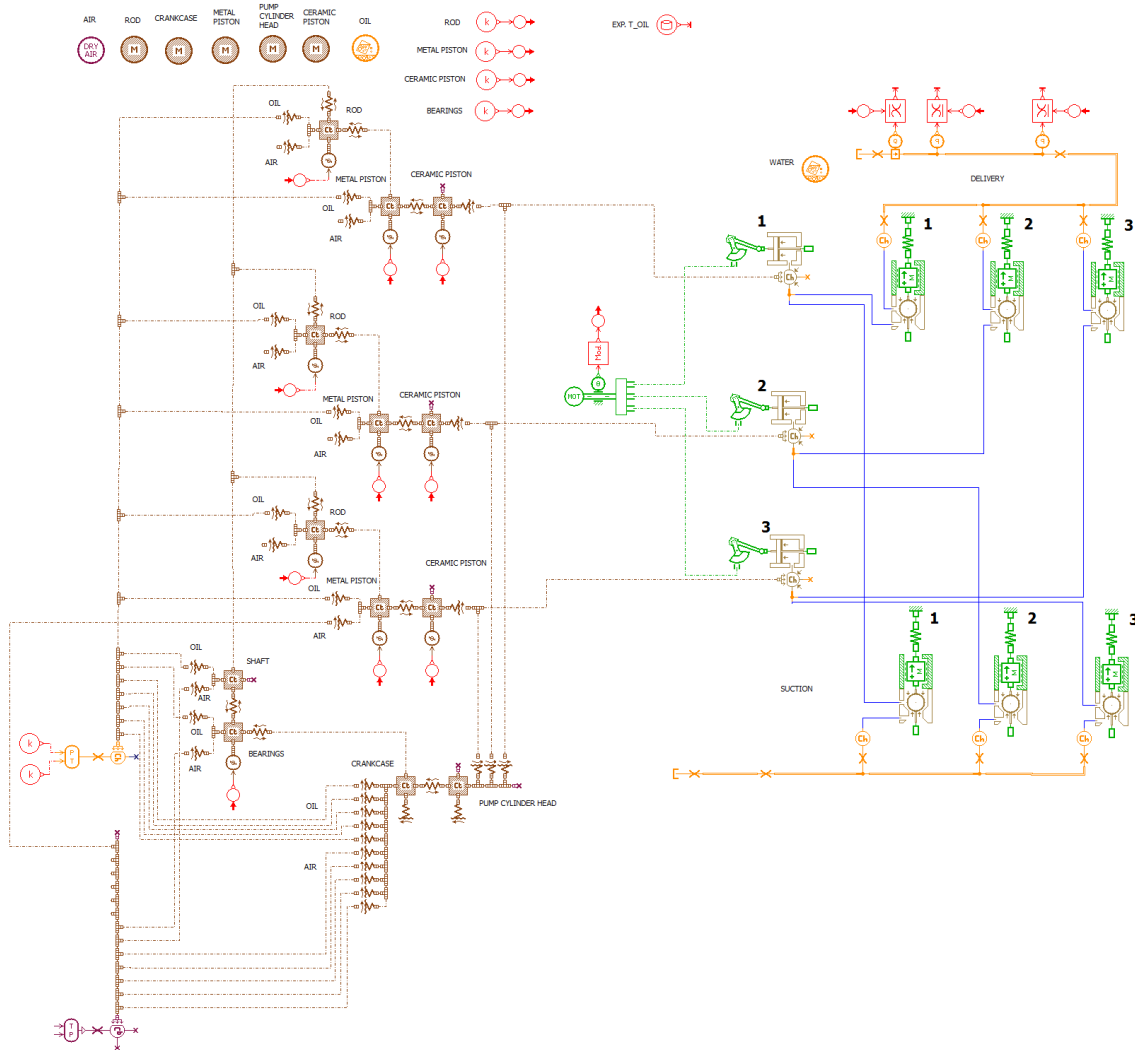
261 The two phases considered are air and lubricant oil. While the air physical properties are
 262 included in the software data base as temperature and pressure dependant, the oil ones
 263 must be provided by the user. The temperature influence on density and viscosity at
 264 atmospheric pressure can be obtained from the oil data sheet. In order to define the heat
 265 transfer, also oil thermal properties have been detailed. Brucker and Majdalani [20]
 266 proposed empirical correlations pressure and temperature dependant to calculate
 267 specific heat $c_{p_{oil}}$ and thermal conductivity k_{oil} of an oil similar to the one used in the
 268 piston water pump.

$$269 \quad k_{oil} = C_0 + C_1 \cdot \left\{ \left(V_{oil} / V_{oil_ref} \right) \cdot \left[1 - 0.101 \cdot \left(T_{oil} / T_{oil_ref} \right) \cdot \left(V_{oil} / V_{oil_ref} \right)^3 \right] \right\}^{-7.6} \quad (16)$$

$$270 \quad c_{p_{oil}} = \left[C_2 + C_3 \cdot \left(T_{oil} / T_{oil_ref} \right) \cdot \left(V_{oil} / V_{oil_ref} \right)^4 \right] / \rho_{oil} \quad (17)$$

271 The oil volume and temperature at actual conditions are V_{oil} and T_{oil} while V_{oil_ref} and
 272 T_{oil_ref} are related to reference values; C_0 , C_1 , C_2 and C_3 are empirical coefficients and
 273 ρ_{oil} is the oil density. During the working condition of the pump, the oil in the crankcase
 274 is constantly at atmospheric pressure; thus, the two equation can be simplified because
 275 the V_{oil} / V_{oil_ref} ratio is equal to 1. In fact, the pump has a breather plug and the model
 276 accounts for it by means of an air inlet at the atmospheric pressure (see Fig. 2). Thus,
 277 only air can enter the crankcase but both oil and air can exit. In particular, a very small
 278 amount of oil can exit from the breather plug, if it is thrown to the plug by the moving
 279 rod. All the other surfaces are considered as “wall” (no mass transfer is allowed by the
 280 boundaries between internal crankcase and the environment).

281



282

283

Fig. 2. Lumped parameter model, layout of the whole pump

284

285 2.4 Heat transfer model

286 Once calculated the dissipated power due to friction and modelled the two-phases fluid
 287 mixture, heat transfer must be defined. From the surfaces interested by power
 288 dissipation, the heat is transferred to the fluid. Heat transfer between the two phases are
 289 automatically included, as well as convection between the fluid and the crankcase
 290 internal walls. In order to account for the thermal power transferred (W_{cond}) through the
 291 walls due to the conduction phenomenon, Eq. 18 is used: T_{wall_int} and T_{wall_ext} are,
 292 respectively, the internal and external wall temperature, S_{wall_int} is the heat transfer
 293 surface and R_{wall} the wall thermal resistance.

$$294 \quad W_{cond} = (T_{wall_int} - T_{wall_ext}) \cdot S_{wall_int} / R_{wall} = (T_{wall_int} - T_{wall_ext}) \cdot S_{wall_int} / (s_{wall} / k_{wall}) \quad (18)$$

295 The wall thermal resistance R_{wall} has been evaluated based on the wall thickness s_{wall}
 296 and the thermal conductivity k_{wall} of the metal. Afterwards, the thermal power W_{conv} is
 297 dissipated towards the environment (natural convection) and it can be calculated as
 298 proposed in Eq. 19.

$$299 \quad W_{conv} = (T_{wall_ext} - T_{env}) \cdot S_{wall_ext} \cdot h_{wall_ext} \quad (19)$$

300 The environment is at atmospheric pressure and its temperature is T_{env} , the transfer
 301 surface is the external area S_{wall_ext} of the crankcase and h_{wall_ext} is the heat transfer
 302 coefficient. In order to define this parameter, a non-dimensional approach is used. It is
 303 possible to evaluate the Nusselt number Nu_{wall_ext} as a function of the Reynolds number
 304 Re_{wall_ext} and of the environment Prantl number Pr_{env} (see Apeendix B). Once obtained
 305 Nu_{wall_ext} , the heat transfer coefficient can be calculated according to Eq. 20, where
 306 l_{wall_ext} is a characteristic length of the transfer surface and k_{env} is the thermal
 307 conductivity of the environment.

$$308 \quad Nu_{wall_ext} = (h_{wall_ext} \cdot l_{wall_ext}) / k_{env} = f(Re_{wall_ext}, Pr_{env}) \quad (20)$$

309 The connecting function must be chosen based on the transfer surface shape, the flow
 310 conditions, the validity range of the non-dimensional numbers. Brucker and Majdalani
 311 [20] shown a comprehensive table of Nusselt number correlations for all these
 312 parameters. Equation 21 had been proposed by Churchill and Chu [21] and it was
 313 developed for natural convection from a planar surface and for $10^0 < Ra < 10^9$.

$$314 \quad Nu_{wall_ext} = 0.68 + \frac{0.67 \cdot Ra_{wall_ext}^{1/4}}{(1 + 0.67 \cdot Pr_{env}^{-9/16})^{4/9}} \quad (21)$$

315 where Ra_{wall_ext} is the Rayleigh number, obtained by multiplying the Grashof number,
 316 Gr_{wall_ext} , referred to the external crankcase surface and the Prandtl number, Pr_{env} , of the
 317 external ambient (see Appendix B). In order to calculate the Grashof number, the
 318 volumetric thermal expansion coefficient of the air b_{env} has to be taken into account:

$$319 \quad b_{env} = -\frac{1}{\rho_{env}} \cdot \left(\frac{\partial \rho_{env}}{\partial T_{env}} \right)_p = \frac{1}{\rho_{env}} \cdot \frac{p_{env}}{Rg \cdot T_{env}^2} = \frac{1}{T_{env}} \quad (22)$$

320 As proposed by Incropera and DeWitt [18], the air can be considered as an ideal fluid
 321 for the evaluation of the volumetric thermal expansion coefficient; thus, it can be
 322 assumed to be equal to approximately $1/T$, where T is the absolute temperature of the
 323 gas (see Eq. 22).

324

325 3. LUMPED PARAMETER MODEL

326 In order to obtain a complete analysis of the overall machine, a lumped and distributed
 327 parameter model is constructed. Indeed, the developed 2D CFD model designed to
 328 describe the thermo-fluid dynamic behaviour of the lubricating system but is not
 329 applicable for a detailed study of the pump due to the high computational effort. In
 330 other words, an overall CFD model that includes both lubricating system and pumping
 331 zone, will cause an high computational resource request and long-time simulations;
 332 thus, a lumped and distributed parameter approach is the best compromise between
 333 computational effort and results' accuracy in order to develop a model that can be able
 334 to show the results in an admissible time and ensuring a good predictive capability.

335 As depicted by Fig. 2, the model of the pump is constructed connecting two main parts:
 336 the pumping side, where the operating fluid, water, is addressed by the piston chamber
 337 evolution from the suction to the delivery, and the mechanical side, where the
 338 lubricating system is placed. The model accounts for the thermo-dynamics behaviour of
 339 both sides; in particular, the heat transfer between the pump cylinder head and the

340 water, the crankcase and the pump cylinder head, the lubricating fluid and the
341 crankcase, are included. There are parameters that can not be fixed on the basis of
342 geometrical or physical information: in order to obtain these data, such as the
343 convective heat transfer coefficient of each wall, the 2D CFD simulation is
344 fundamental. On the other hand, the lubricating system simulation requires to fix the
345 heat transferred from the crankcase to the pump cylinder head. Thus, the CFD
346 lubricating system model and the lumped parameter model of the pump are deeply
347 dependant each other and they need to be simultaneously developed.

348

349 3.1 Pumping side model

350 The piston chamber evolution of each of the three pistons, properly phased, is accounted
351 for by this part of the model. Particular care is devoted to the modelling of the opening
352 characteristic of the suction and delivery automatic valves by means of an accurate
353 geometrical definition; in addition to this, the spring displacement –force relationship
354 and the moving parts mass are also included. To complete the layout, the suction and
355 delivery line are considered, as well as the tank at the atmospheric pressure value and an
356 orifice used as pump load.

357 The hydraulic behaviour predicted by the model is tailored by means of experimental
358 data, in terms of load pressure and flow rate and volumetric efficiency. The heat transfer
359 between the pumping side and the mechanical side is permitted by means of the
360 crankcase – pump cylinder head contact interface and by the ceramic piston part – metal
361 piston part contact interface; these contact interfaces accounted for conduction, as well
362 as convection phenomena. In fact, while the pump cylinder head and the ceramic piston
363 part are cooled by the water flow, the crankcase and the metal piston part are in contact
364 with the oil and they are hooted by the power dissipation due to the friction, as
365 explained above. In addition, there is an amount of energy that is released as heat due to
366 the friction between the ceramic piston part and the seal; this thermal power P_{f_ring} is
367 included in the model and it is calculated by means of the approach described in the
368 Paragraph 2.2.

369 In order to define the thermal power transferred by the pump cylinder head towards the
370 environment, the heat transfer coefficient must be calculated. The numerical approach
371 employed is the same used for the crankcase in the CFD model, based on the Nusselt
372 number correlation of Eq. 21. As said above, the pump cylinder head is cooled by the
373 water flow. The internal geometry of the component is really complex, but, as an
374 approximation, it is possible to calculate the hydraulic diameter d_h and to consider the
375 convection phenomenon as referred to a turbulent flow in circular tubes; in other words,
376 the Nusselt number Nu_{head_int} is simulated by means of the Dittus-Boelter equation (as
377 shown by Incropera and DeWitt [18]):

$$378 \quad Nu_{head_int} = 0.023 \cdot Re_{head_int}^{4/5} \cdot Pr_{water}^{nc} \quad (23)$$

379 Where Re_{head_int} is the Reynolds number of the internal duct of the pump cylinder head,
380 using the hydraulic diameter d_h as characteristic length l (see Appendix B), Pr_{env} is the
381 water Prantl number and nc is a exponent equal to 0.4 for flow heating and 0.3 for flow
382 cooling. Eq. 23 is normally used for small temperature difference and for the range of
383 conditions: $0.6 \leq Pr \leq 160$, $Re \geq 10^4$, $l/d \geq 10$. Also the ceramic piston part is cooled by

384 water: the contact surface is a circular area that is moved inside the piston chamber. It is
 385 very difficult to define the flow condition. On the basis of the Reynolds number Re_{cer_p} ,
 386 it is not possible to recognize a fully developed turbulent flow; thus, a correlation
 387 validated for mixed condition on a flat plate and for $0.6 \leq Pr \leq 60$, $5 \cdot 10^5 < Re < 10^8$,
 388 (Incropera and DeWitt [18]), can be used in order to obtain an average value of Nusselt
 389 number Nu_{cer_p} , and, consequently, an average value of heat transfer coefficient:

$$390 \quad \overline{Nu}_{cer_p} = (0.037 \cdot Re_{cer_p}^{4/5} - 871) \cdot Pr_{water}^{1/3} \quad (24)$$

391 In the model, the conduction between two components is automatically calculated based
 392 on the material properties and the geometrical characteristic of the contact surface. In
 393 addition, it is possible to set a contact thermal resistance for cases where the surface
 394 roughness must be considered.

395

396 3.2 Mechanical side model

397 This part of the model focuses on the heat transferred between lubricating fluid and
 398 mechanical components by means of convection; the model includes also the
 399 conduction between the parts in contact. As mentioned above, each conduction interface
 400 requires the geometrical parameters and the involved material properties. The
 401 dissipation of mechanical power due to friction is considered by means of the approach
 402 described in the Paragraph 2.2 in the CFD model; more in details, P_{f_rbe} and P_{f_rse} are
 403 addressed to the rod as well as P_{f_jb} is referred to the metal piston part and P_{f_nb} , regards
 404 the needle bearings.

405 The lubricating fluid is composed of air and oil. The lumped parameter approach
 406 normally does not let the user to model a multiphase fluid; thus, two different virtual
 407 volumes (one of air and one of oil) are employed. The sum of the two volume is equal
 408 to the internal volume of the crankcase. Each volume can transfer heat with all the
 409 components that are in contact with the lubricating fluid, by means of various interfaces
 410 of area S_{eff} :

$$411 \quad S_{eff} = nd \cdot S_{geo} \quad (25)$$

412 Where S_{geo} is the geometrical area and nd is a coefficient of covered area, obtained from
 413 the CFD simulation and equal to the surface average of the oil mass fraction (nd_{oil}) and
 414 the air mass fraction (nd_{air}). Thus, each contact interface is divided in two surfaces, one
 415 of area S_{eff_oil} referred to the oil and one of area S_{eff_air} referred to the air.

416 The thermal power transferred from the crankcase to the environment is described by
 417 means of the same approach (Eq. 21) used in the CFD model, while the heat transferred
 418 between the lubricating fluid (both air and oil) and each crankcase wall is modelled by
 419 means of the Eq. 24. Each wall of the crankcase is considered as a flat plat and is
 420 characterized by different geometry and oil/air distribution; the flow is described by a
 421 Reynolds number too low for a fully developed turbulent condition, thus, the Nusselt
 422 number correlation (Eq. 24) proposed seems to be a good approach to obtain the related
 423 heat transfer coefficient. In fact, air and oil are continually mixed inside the crankcase
 424 by the moving parts, but the fluid does not reach an average velocity sufficiently high to
 425 be in turbulent condition. For the same reason, also the Nusselt number referred to the

426 contact surface between the bearings and the lubricating fluid is calculated with the
427 same approach (Eq. 24). The surface is the area between the bearing external
428 circumference and the shaft external circumference. The heat transferred from the shaft
429 to the lubricating fluid is accounted for by means of an approach validated for rotating
430 cylinder in a cross flow (Incropera and DeWitt [18]):

$$431 \quad \overline{Nu}_{sh_oil} = 0.193 \cdot Re_{sh_oil}^{0.618} \cdot Pr_{oil}^{1/3} \quad (26)$$

432 Where Re_{sh_oil} is the rotational Reynolds number of the oil dragged by that shaft (see
433 Appendix B) and Pr_{oil} is the Prandtl number of the oil. This correlation is used for the
434 range of conditions $0.7 \leq Pr$, $4 \cdot 10^3 \leq Re \leq 4 \cdot 10^4$ and it can be employed for both oil and
435 air. For the contact surface between the rod and the lubricating fluid, the heat transfer
436 coefficient is obtained by the 2D CFD simulation, as well as the one referred the
437 interface between the metal piston part and the lubricating fluid. The 2D CFD model is
438 also used, as said above, to calculate the coefficients of covered area for both oil and air,
439 regarding all the considered contact surfaces.

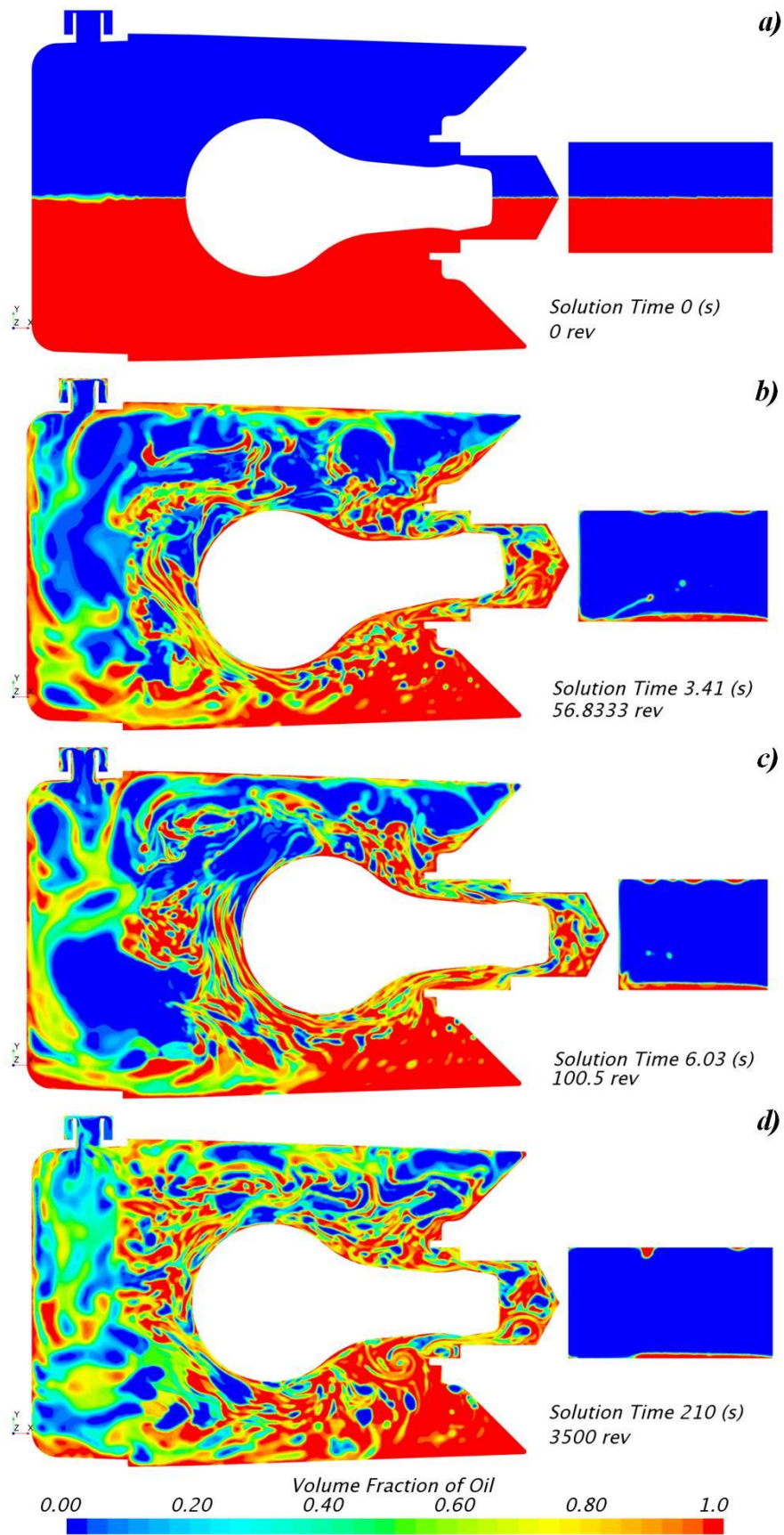
440 4. CFD MODEL RESULTS

441 The results of the CFD model of the lubricating system in terms of heat transfer
442 coefficient and oil and air distribution are then employed in the lumped and distributed
443 numerical model.

444 In the CFD model, the rotational speed used is one thousand rpm and the employed time
445 step is 0.1 millisecond; thus, angular time step is smaller than one degree. A breather plug
446 is included in the geometry in order to maintain the atmospheric pressure of the fluid
447 volume inside the crankcase. . Afterwards, the initial temperature value is set equal to the
448 ambient temperature, while the initial oil and air distribution is shown in Fig. 3a. The oil
449 mass fraction is equal to the 50% of the volume.

450 On the right side of the picture, a rectangular shape can be noticed that is a fictitious
451 volume separated from the main volume by the piston. This volume is requested by the
452 overset mesh technique in order to correctly describe the piston movement, but it is not
453 referred to the real cylinder. In fact, in the real machine, on this side of the piston there is
454 water, that is to say, the pumped fluid. This fictitious volume has the same initial pressure,
455 temperature and air-oil distribution of the main volume but it is physically separated from
456 the crankcase volume, thus, the air and oil in this region do not influence the fluid
457 dynamic behaviour of the crankcase volume. In order to highlight this point, an open
458 boundary is included in the simulation at the left side of the fictitious volume: after few
459 crankshaft revolutions the volume is almost full of air at the environment conditions.

460 While the rod and the piston position at the BDC (bottom dead centre) are shown in the
461 Fig. 3a and in the Fig. 3d, Fig. 3c depicts the machine in the TDC configuration (top
462 dead centre) and an arbitrary angular position is chosen in Fig. 3b.



463

464

465

Figure 3- Volume fraction of oil in the crankcase, referred to a) initial condition; b) after 3.41 s c) after 6.03 s; d) after 210 s.

466 Fig. 3b displays the air-oil volume fraction after 3.4 seconds; the oil and air are mixed
467 but a separation between the fluids can be still identified. A similar phenomenon can be
468 noticed also in Fig. 3c, i.e. simulation time equal to 6 seconds which corresponds to 100
469 revolutions. The last picture, Fig. 3d, shows the oil volume fraction distribution when
470 the system has reached a steady state condition, i.e. simulation time 210 s; the air and
471 the oil are completely mixed.

472 In all the presented pictures, there is a no recirculating zone where oil is almost fixed, in
473 the right-lower side, under the cylinder. A good oil recirculation is one of the most
474 important goal for lubricating system design, so, if this behaviour will be confirmed also
475 by the three-dimensional simulations, the crankcase geometry should be modified in
476 order to avoid it. During the transient period, a small amount of oil can escape through
477 the breather plug, as confirmed by experimental test, but after a few seconds these oil
478 losses are no more observable. The oil amount in the fictitious volume, starting from the
479 initial value, in a few revolutions decreased rapidly. Only a thin oil layer is still
480 observable in the steady-state phase. As said above, this fictitious volume has no
481 relation with the real cylinder, because the pumped fluid is water.

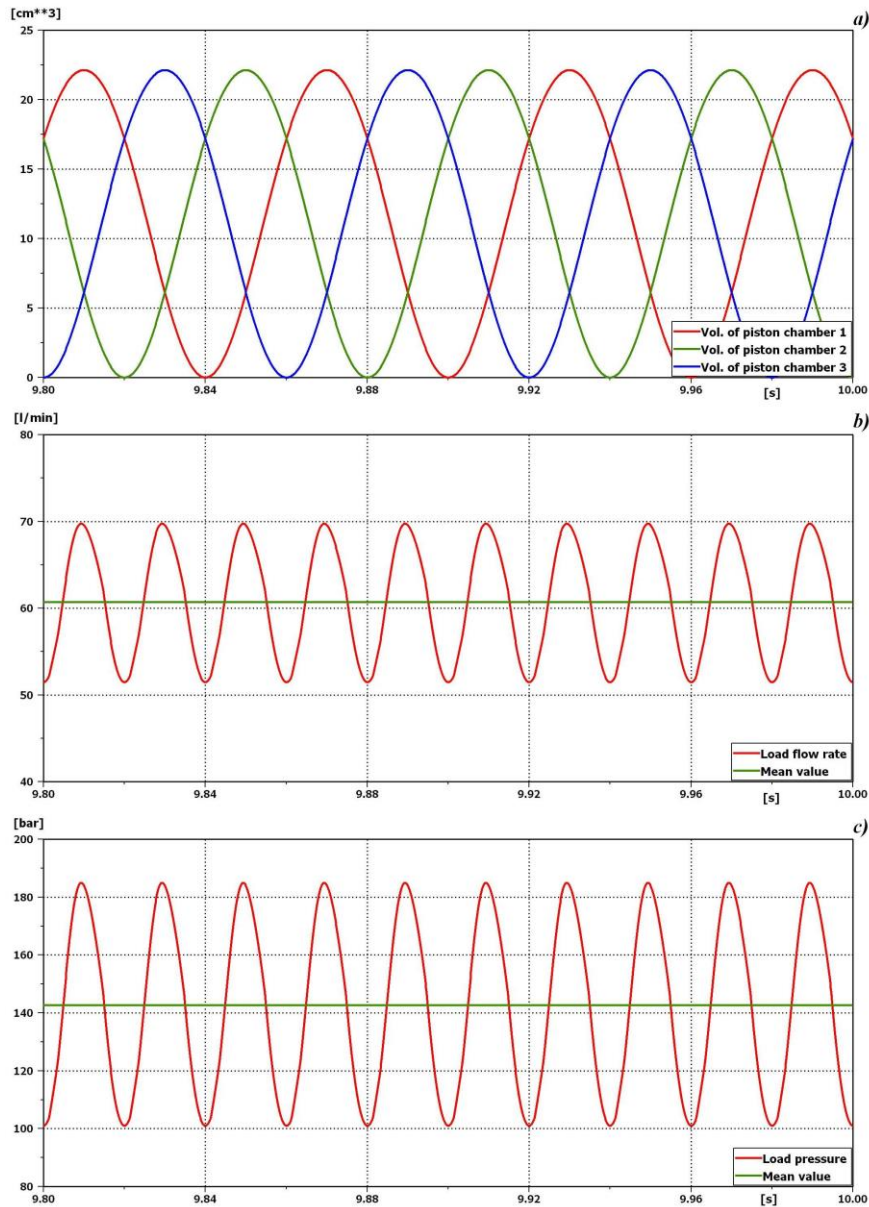
482 The results obtained from this 2D CFD model are compared to experimental
483 measurements and a good agreement is obtained from a qualitatively point of view.

484 In fact, it is not possible to strictly compare the numerical data achieved from the 2D
485 CFD model to the experimental data: converting from 3D to 2D, the crankcase walls
486 parallel to the model plane are neglected. In other words, the crankcase area able to
487 transfer heat form the fluid inside to the environment is different from the one of the
488 real geometry. The thermal power introduced in the model due to friction is reduced to
489 account for this consideration. In fact, the time duration of the numerical thermal
490 transient is minor that the real one, but the numerical mean value of the oil temperature
491 in steady-state condition is quite close to the experimental value.

492 Thus, the 2D model can not be used to predict exactly the punctual temperature
493 evolution of the lubricating system but the qualitatively good agreement between
494 numerical results and measurements lets the user to usefully employ the model to
495 estimate the heat transfer coefficient and the air-oil distribution of each surface. These
496 data are introduced in the lumped and distributed parameter model to obtain a predictive
497 model of the pump. This approach, based on the use of a 2D CFD model and a lumped
498 parameter model, has a computational effort minor than a complete 3D CFD model;
499 thus, the combined approach demonstrated to be a reliable tool to achieve the numerical
500 results with good accuracy.

501 5. LUMPED PARAMETER MODEL RESULTS

502 The lumped and distributed numerical model of the whole pump is tailored in two steps.
503 Firstly, the pumping side is accounted in the analysis and the measurements are
504 compared with the numerical results in terms of load pressure and flow rate. More in
505 details, the discharge coefficient and the friction parameter of each valve are introduced
506 and regulated in order to obtain a good agreement between numerical and experimental
507 data. Particular care is devoted to the angular phasing of the three pistons: in Fig. 4a the
508 volume evolution of the three piston chambers are shown. Figs. 4b and 4c depicts the
509 instantaneous and the mean values of load pressure and flow rate. The curves are very
510 close to the measurements; thus, the model is able to describe the fluid dynamics
511 behaviour of the pumping side and it is possible to calculate the volumetric efficiency,
512 that is equal to the experimental value and higher than 90%.



513

514 Figs. 4. Pumping side analysis: a) Phasing of piston chamber volume; b) load flow rate;
 515 c) load pressure

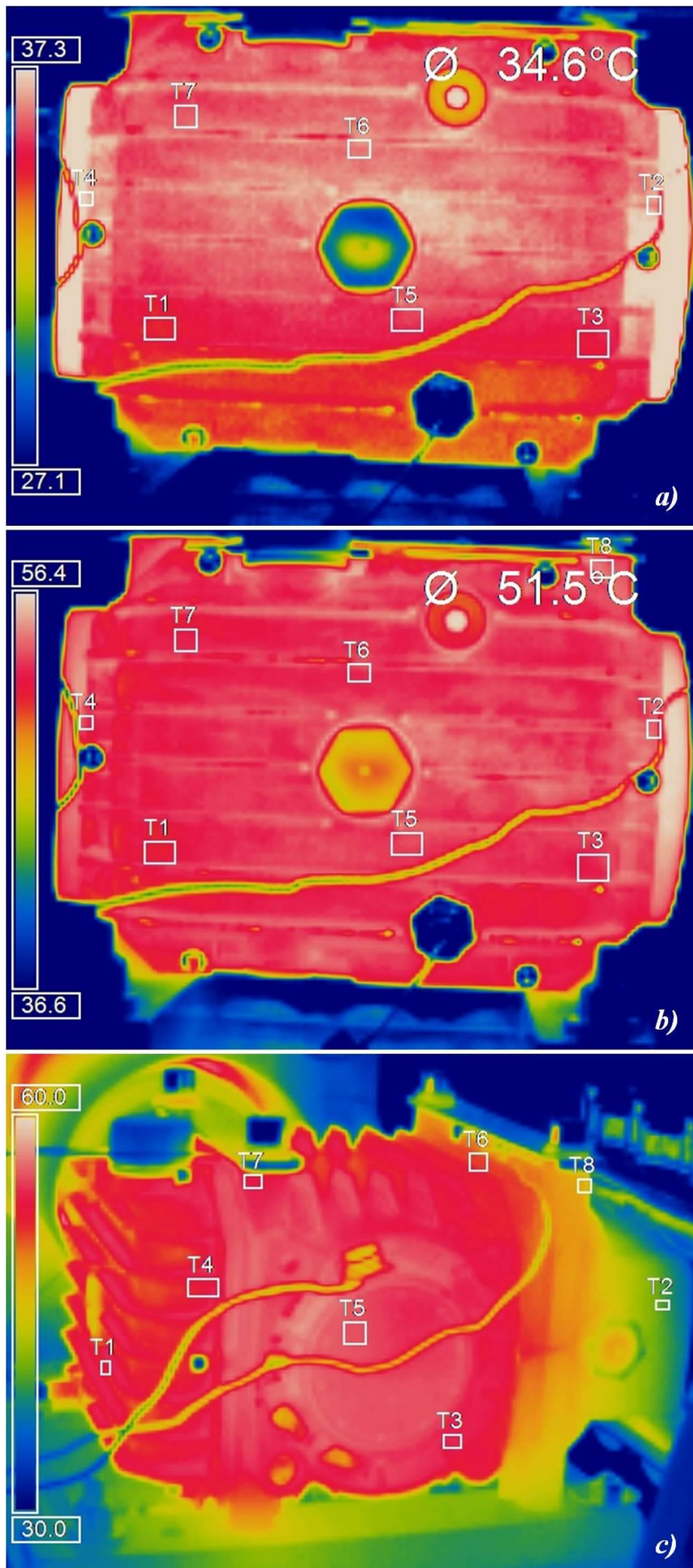
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517 Once the pumping side is tailored and validated, the mechanical side of the pump model
 518 can be completed. In particular, the heat transfer coefficient and the air-oil distribution
 519 of each surface of the internal geometry of the crankcase are achieved from the 2D CFD
 520 model and they are employed in the lumped parameter model in order to enhance the
 521 accuracy.

522 The validation of the whole pump model is achieved comparing the numerical results
 523 with the measurements carried out by means of thermocouples type K, placed in
 524 different positions of the crankcase, on the external walls and in the internal oil volume.
 525 The experimental oil temperature curve has been obtained as a mean of the
 526 measurements carried out and it has been used to tailor the numerical model. Water and
 527 air temperature are monitored, and the ambient temperature is recorded too. Both
 528 transient and steady-state operations are considered.

529 Afterwards, a thermographic camera is adopted in order to obtain a complete
530 temperature distribution of the external walls of the machine. The device used is a
531 Optris PI 600 thermocamera characterized by a spectral range of 7.5-1.3 μm , a
532 temperature range from -20°C to 900°C and an optical resolution of 160×120 pixel; the
533 frequency is 120 Hz.

534 Fig. 5 shows the thermal images in different positions: the temperature reported at the
535 top of each image (e.g., 34.6°C in Fig. 4a) is relative to the average value of all the pixel
536 that compose the T1 probe box. In the case shown in Figs. 5a and 5b, the camera is
537 positioned in front of the crankcase cover, the opposite part of the pump cylinder head
538 side. By monitoring the heating transient, it is possible to observe how the hottest parts,
539 the needle bearings and the shaft, progressively transfer thermal power from the middle
540 plane to the upper and the lower side, until the wall is almost at the same temperature
541 (in steady-state condition, see Fig. 5b). This consideration about the uniformity of the
542 temperature confirms that the lumped parameter approach can be used to describe the
543 system with a good accuracy: indeed, if the temperature distribution on each component
544 is uniform, the error due to the description of each part as an numerical element
545 characterized by a single temperature value, is very limited. Fig. 5c shows the whole
546 crankcase from a different view in the steady state condition: in particular, the cooling
547 effect of the pump cylinder head (where the water flows) can be observed on the left
548 side. The effect is restricted to a narrow zone but it can not be neglected: for this reason,
549 the lumped and distributed parameter model is referred to the whole pump and it
550 account for both thermal power dissipation between the crankcase and the environment
551 and between the crankcase and the pump cylinder head.



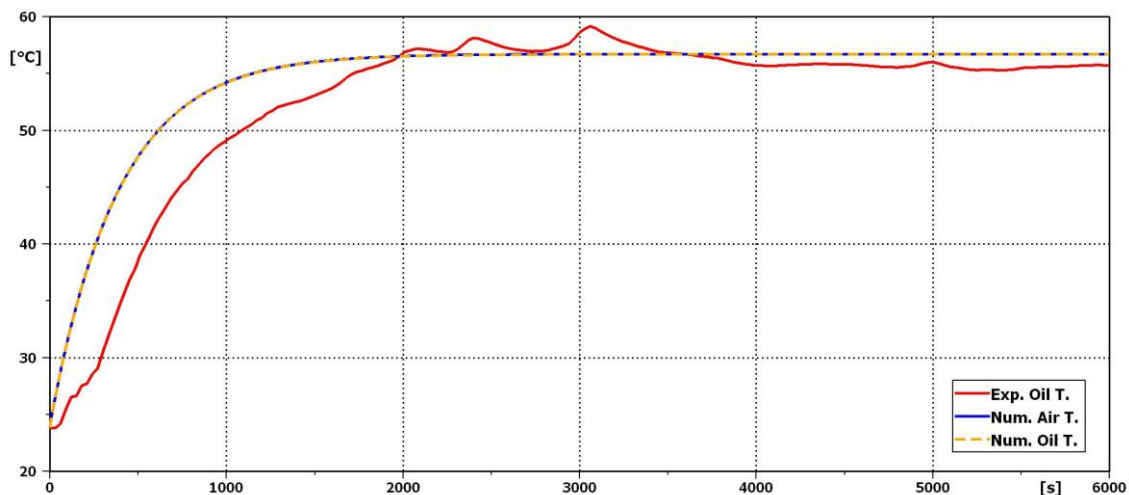
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553

554

Figs. 5. Thermographic analysis: a) crankcase cover after a few seconds; b) crankcase cover in steady-state condition; c) whole pump in steady-state condition

555 As mentioned above, the experimental campaign is carried out to tailor and validate the
 556 numerical model of the pump. Fig. 6 shows the comparison between the measured oil
 557 temperature and the numerical one, as well as the numerical air temperature. More in
 558 details, the model can not consider a multiphase fluid; thus, two separated fluids are
 559 included and so, two numerical temperatures are obtained. Each phase is able to
 560 exchange thermal power with the surfaces which are in contact, on the basis of the
 561 air/oil distribution obtained from the 2D CFD model. Two volumes are used, one for the
 562 air and the other one for the oil. Each volume is equal to the 50% of the internal volume
 563 of the crankcase. Nevertheless, in Fig. 6 it is possible to observe that the two numerical
 564 temperatures are perfectly overlapped, as a consequence of the model reliability. In fact,
 565 even if air and oil have no direct interfaces in the model, they are in contact with the
 566 same surfaces and it seems physically correct that the two curves are equal. In
 567 particular, due to the strongly different thermo physical properties of the two fluids, the
 568 thermal equilibrium is mainly influenced by the oil. Thus, the comparison is based on
 569 the oil temperature: the agreement is excellent for the steady-state condition (the error is
 570 around the 2%) but the numerical curve increases faster than the experimental one in the
 571 transient phase. This is due, on the one hand, to the 3D effect of the heat transfer
 572 phenomenon, that the lumped model can not consider, and on the other hand, to the
 573 employed data logger. In fact, in order to remove the noise from the signal, the data
 574 logger automatically applies a moving average to the raw data. This increases the
 575 quality of the signal, but it introduces a delay. Both the 3D effect and the signal
 576 treatment influence are more significant during the transient phase than the steady-state
 577 phase. In order to enhance the lifetime of the pump, it is very important to avoid too
 578 high temperature when the machine operates continuously; thus, it is possible to accept
 579 a quite poor agreement between numerical and experimental data in the transient phase
 580 because, in the steady state condition, the agreement is very good. The overheating risk
 581 regards only the mechanical side of the pump. In fact, observing Figs. 4 and 6, the
 582 hydraulic transient of the pumping side is strongly minor than the thermal transient of
 583 the mechanical side: after a few seconds, the load flow rate and pressure are in steady
 584 state condition, while the oil temperature of the lubricating system requires more than
 585 4000 s to be stable. Thus, the lubricating system temperature does not influence directly
 586 the operating point of the pump: the pumping side temperature is fixed by the water
 587 flow.



588

589

Figure 6. Comparison between measured and calculated oil temperature

590 6. CONCLUSIONS

591 This paper has presented a numerical approach for the prediction of the thermo fluid
592 dynamics behaviour of a piston water pump. Particular care has been devoted to the
593 lubricating system model and to the heat transferred from the internal crankcase to the
594 environment. A 2D CFD model of the system has been constructed, accounting for the
595 thermal power released by friction, the mixing of the two fluids (oil and air) in the
596 crankcase volume, the moving parts (rod and piston) described by means of the overset
597 mesh technique.

598 The outputs of the 2D CFD model, in terms of heat transfer coefficient and air/oil
599 distribution of each surfaces, have been passed to a lumped and distributed parameter
600 model of the whole pump, properly designed to describe both the operating point of the
601 pumping side and the thermal condition of the mechanical side. Conduction and
602 convection phenomena between the pump cylinder head and the crankcase, and between
603 the crankcase and the environment have been included. The model has been tailored and
604 validated using experimental data carried out by means of two different measurements
605 technique: thermocouples analysis and thermography. The employed thermocamera has
606 highlighted that the temperature of each component has a uniform distribution,
607 confirming the most important hypothesis for the use of the lumped parameter
608 approach.

609 The numerical results, in terms of oil temperature, have been compared with the
610 acquired data and a good agreement has been found, especially in the steady-state
611 condition. Simulation and measurements have confirmed that the water flow has a
612 cooling effect on the pumping side and the temperature in this zone is fixed by the
613 water. In fact, the operating point of the pump is not influenced by the thermal transient
614 of the lubricating system. On the other hand, even if the crankcase is partially cooled by
615 the conduction between the component itself and the pump cylinder head, this
616 phenomenon is not sufficient to maintain the mechanical side temperature under the
617 overheating limit without adopting a lubricating system.

618 Combing the 2D CFD lubricating system model and the lumped parameter model of the
619 whole pump, the user can achieve all the information needed to properly design the
620 machine and in particular the lubricating system. The approach is able to ensure a good
621 accuracy in an acceptable time: both the operating point and the thermo fluid dynamics
622 behaviour of the pump are described and the computational effort is minor than the one
623 referred to a complete 3D CFD model.

624

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627 contribution to this work for both the experimental analysis and the simulation
628 approach.

629

630

631

632 LIST OF NOTATIONS

a	Thermal diffusivity	m^2/s
Ar	Cross section area	m^2
b	Volumetric thermal expansion coefficient	K^{-1}
\bar{c}	Mean radial clearance	m
C	Coefficient	-
C_0	First coefficient of k_{oil} correlation	$0.053\text{W}/(\text{m}^*\text{K})$
C_1	Second coefficient of k_{oil} correlation	$0.026\text{W}/(\text{m}^*\text{K})$
C_2	First coefficient of c_{poil} correlation	$1.17*10^6\text{J}/(\text{m}^3*\text{K})$
C_3	Second coefficient of c_{poil} correlation	$0.39*10^6\text{J}/(\text{m}^3*\text{K})$
c_p	Specific heat	$\text{J}/(\text{kg}^*\text{K})$
d	Diameter	m
F	Force	N
g	Gravitational constant	m/s^2
Gr	Grashof number	-
h	Heat transfer coefficient	$\text{W}/(\text{m}^2*\text{K})$
k	Thermal conductivity	$\text{W}/(\text{m}^*\text{K})$
l	Length	m
M	Torque	N^*m
n	Number of	-
nc	Convection exponent	-
nd	Coefficient of covered area	-
Nu	Nusselt number	-
p	Pressure	Pa
P	Power	W
Per	Perimeter	m
Pr	Prandtl number	-
Q	Flow rate	m^3/s
r	Radius	m
R	Thermal resistance between the internal and the external case wall	$\text{K}^*\text{m}^2/\text{W}$
Ra	Rayleigh number	-

Re	Reynolds number	-
Rg	Perfect gas law constant	J/(mol*K)
s	Wall thickness	m
S	Surface area	m ²
t	Time	s
T	Temperature	K
v	Velocity	m/s
V	Volume	m ³
W_{cond}	Thermal power transferred through the wall	W
W_{conv}	Thermal power transferred between the wall and the environment	W
y	Position referred to the Y axis	m
\dot{y}	Velocity referred to the Y axis	m/s
x	Position referred to the X axis	m
\dot{x}	Velocity referred to the X axis	m/s
α	Angle between crank and piston axis	rad
β	Angle between rod and piston axis	rad
$\dot{\beta}$	Rod rotational speed	rad/s
η	Efficiency	-
λ	Ratio between crank and rod length	-
ρ	Density	Kg/m ³
μ	Dynamic viscosity	Pa*s
ω	Crankshaft rotational speed	rad/s

633 Subscripts

A	Point A - connecting rod small end
air	Air
B	Point B - connecting rod big end
c	Crank
cer	Ceramic
eff	Effective
env	Environment
ext	External
geo	Geometrical
f	Friction

<i>h</i>	hydraulic
<i>head</i>	Pump cylinder head
<i>hm</i>	Hydro-mechanical
<i>hyd</i>	Hydraulic
<i>int</i>	Internal
<i>jb</i>	Journal box
<i>J</i>	Point J – generic rod point
<i>max</i>	Maximum value
<i>mech</i>	Mechanical
<i>nb</i>	Needle bearing
<i>oil</i>	Oil
<i>p</i>	Piston
<i>r</i>	Connecting rod
<i>ref</i>	Reference value
<i>ring</i>	Seal placed in the cylinder wall
<i>rbe</i>	Rod big end
<i>rse</i>	Rod small end
<i>suc</i>	Suction
<i>sh</i>	Shaft
<i>tot</i>	Total
<i>vol</i>	Volumetric
<i>wall</i>	Crankcase wall
<i>water</i>	Water

634

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 695

696 APPENDIX A

$$\alpha = \omega \cdot t \quad (\text{A.1})$$

$$\sin \beta = (r_c / l_r) \cdot \sin \alpha = \lambda \cdot \sin \alpha \quad (\text{A.2})$$

$$\sin^2 \beta = \lambda^2 \cdot \sin^2 \alpha \quad (\text{A.3})$$

$$\sin^2 \beta + \cos^2 \beta = \lambda^2 \cdot \sin^2 \alpha + \cos^2 \beta \quad (\text{A.4})$$

$$\cos \beta = \sqrt{1 - \lambda^2 \cdot \sin^2 \alpha} \quad (\text{A.5})$$

$$x = (r_c + l_r) - r_c \cdot \cos \alpha - l_r \cdot \cos \beta = r_c \left[(1 + 1/\lambda) - \cos \alpha - (1/\lambda) \cdot \sqrt{1 - \lambda^2 \cdot \sin^2 \alpha} \right] \quad (\text{A.6})$$

$$\dot{x} = \frac{dx}{dt} = \frac{dx}{d\alpha} \cdot \frac{d\alpha}{dt} = r_c \cdot \omega \cdot \left[\sin \alpha + \frac{\lambda \cdot \sin \alpha \cdot \cos \alpha}{\sqrt{1 - \lambda^2 \cdot \sin^2 \alpha}} \right] = r_c \cdot \omega \cdot \left[\sin \alpha + \frac{\lambda \cdot \sin 2\alpha}{2 \cdot \sqrt{1 - \lambda^2 \cdot \sin^2 \alpha}} \right] \quad (7)$$

$$\beta = \arcsin(\lambda \cdot \sin \alpha) \quad (\text{A.7})$$

$$\frac{d\beta}{dt} = \frac{1}{\sqrt{1 - \lambda^2 \cdot \sin^2 \alpha}} \cdot \lambda \cdot \cos \alpha = \lambda \cdot \frac{\cos \alpha}{\cos \beta} \quad (\text{A.8})$$

$$\dot{\beta} = \frac{d\beta}{dt} = \frac{d\beta}{d\alpha} \cdot \frac{d\alpha}{dt} = \lambda \cdot \omega \cdot \frac{\cos \alpha}{\cos \beta} = \lambda \cdot \omega \cdot \frac{\cos \alpha}{\sqrt{1 - \lambda^2 \cdot \sin^2 \alpha}} \quad (6)$$

697

698 APPENDIX B

$$699 \operatorname{Re}_{\text{wall_ext}} = (v_{\text{env}} \cdot \rho_{\text{env}} \cdot l_{\text{wall_ext}}) / \mu_{\text{env}} \quad (\text{B.1})$$

$$700 \operatorname{Pr}_{\text{env}} = \mu_{\text{env}} / (\rho_{\text{env}} \cdot a_{\text{env}}) \quad (\text{B.2})$$

$$701 \operatorname{Gr}_{\text{wall_ext}} = g \cdot b_{\text{env}} \cdot (T_{\text{wall_ext}} - T_{\text{env}}) \cdot l_{\text{wall_ext}} \cdot \rho_{\text{env}} / \mu_{\text{env}} \quad (\text{B.3})$$

$$702 \operatorname{Ra}_{\text{wall_ext}} = \operatorname{Pr}_{\text{env}} \cdot \operatorname{Gr}_{\text{wall_ext}} \quad (\text{B.4})$$

$$703 d_h = 4 \cdot Ar / Per \quad (\text{B.5})$$

$$704 \operatorname{Re}_{\text{sh_oil}} = (\omega \cdot \rho_{\text{oil}} \cdot d_{\text{sh}}) / \mu_{\text{oil}} \quad (\text{B.6})$$