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ON THE POWER DEMANDS OF ACCESSORIES ON AN AGRICULTURAL TRACTOR

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ABSTRACT

Recently, the number of accessories in vehicle powertrains has been significantly increased. These accessories are necessary for vehicle operation. Accessories are driven by the engine, so their power demands result in a reduction of the vehicle's useful power. For heavy-duty vehicles, the most demanding accessories are the alternator; fan drive; heating, ventilation, and air-conditioning compressor, and brake air compressor. Their power demands vary according to the engine speed, and this may lead to notable energy losses especially in certain conditions, such as engine idling. This paper aims to evaluate the accessories power demands on a tractor equipped with an engine rating of 192 kW. In this tractor, a data logger together with auxiliary sensors were installed to monitor their power demands. During idling conditions, accessories power demands reach 38 % of the power provided by the engine and this leads to non-negligible impact on fuel consumption. The fuel consumption caused by accessories and their potential fuel savings by introducing electrically-driven accessories into tractor powertrains were estimated. A potential fuel saving of 232 l per year could be attained. This may lead to a yearly saving of 278 million litres of fuel and 747 thousand tons of CO₂ in the US. These savings were calculated assuming that all tractors are used with the same duty cycle. Even if it may not occur, thus the actual savings may be significantly different than that reported in this study, it permits to estimate if farming may benefit from tractors with electrically driven accessories.

32

33

34 **KEYWORDS:** engine accessories; powertrain; real-world data; fuel economy; vehicle

35 electrification.

Nomenclature	
β_{fan}	Estimate per cent fan speed
γ_{HVAC}	HVAC compressor engagement
ΔT	Temperature gradient between actual and setpoint cabin temperatures
Δt_{OC}	Total sampling period in generic operative condition
η_{alt}	Alternator efficiency
τ_{HVAC}	Transmission ratio between the engine angular speed and the HVAC compressor
τ_{alt}	Transmission ratio between the engine angular speed and the alternator
τ_{BAC}	Brake air compressor ratio
τ_{fan}	Transmission ratio between the engine angular speed and the fan drive
APU	Auxiliary power unit
D_{OC}	Duty cycle coefficient in generic operating condition
\bar{D}_{OC}	Average duty cycle coefficient in generic operating condition
ECU	Electronic control unit
$FC_{*,OC}$	Average fuel rate of a generic accessory (*) in generic operating condition
FR	Engine fuel rate
GNSS	Global navigation satellite system
HVAC	Heating, ventilation and air-conditioning
I_{alt}	Alternator output current
ICE	Internal combustion engine
$M_{eng\%}$	Actual engine percent torque
M_{er}	Engine reference torque
$M_{f\%}$	Nominal friction percent torque
N_{OC}	Number of samples in generic operating condition
n_{HVAC}	HVAC compressor speed
n_{alt}	Alternator speed
n_{BAC}	Brake air compressor speed
n_{eng}	Engine speed
n_{fan}	Fan speed
OC	Operating condition according to GNSS
$P_{*,OC}$	The average power demand of a generic accessory (*) in generic operating condition
P_{alt}	Alternator power demand
P_{atm}	Atmospheric pressure
P_{aux}	Total accessory power demand
$P_{aux,OC}$	Average accessory power demand in generic operating condition
P_{BAC}	Brake air compressor power demand
P_{eng}	Engine power
$P_{eng,OC}$	Average values of P_{eng} in generic operating condition
P_{fan}	Fan power demand
p_{filt}	Air filter pressure
P_{HVAC}	HVAC power demand
p_{in}	Air intake pressure
p_{tb}	Trailer brake compressor pressure
p_{tk}	Trailer brake tank pressure
PNG	Parameter number group
PTO	Power take-off
SPN	Suspect parameter number
RHP	Rear hitch position
S_{HVAC}	HVAC switch status
T_{cab}	Actual cabin temperature
T_{fan}	Engine coolant temperature
t_{OC}	Duration of the recorded data of each operating state
T_{sp}	Setpoint cabin temperature
t_{pc}	Pulse period of each cycle

t_{wc}	Pulse width of each cycle
v_a	Ground-based speed
V_{bat}	Battery potential

36

1. Introduction

37 Heavy-duty diesel vehicles, including agricultural tractors, produce relatively great amounts
38 of nitrogen oxides (NO_x) and particulate matter (PM) compared with light-duty vehicles
39 (Brodrick et al., 2002). Environmental concerns, fuel prices, and emissions regulations are
40 pushing the engine industry to improve engine designs to reduce fuel consumption and
41 emissions, so that modern society can lower the usage of fossil fuels. To this goal, research
42 centres and vehicle manufacturers have been investigating the power demands of each
43 component of powertrains. A family of components which has gained a lot of attention in recent
44 years in heavy-duty vehicles are accessories (Baglione et al., 2007; Campbell & Kittelson,
45 2009). These are essential for ensuring the proper vehicle operation and operator comfort, but
46 they limit the vehicle's useful power.

47 To meet driver expectations in terms of greater performance and comfort, the number and
48 the size of accessories in vehicle powertrains have been increased in the last thirty years. Since
49 they demand power to operate, their power losses are also taken into account in all engine drive
50 cycle procedures (Hilliard & Springer, 1984). Among the others, the alternator; the fan drive;
51 the heating, ventilation, and air-conditioning (HVAC) compressor and the brake air compressor
52 are the most demanding accessories in heavy-duty vehicles (Hahn, 2008; Hendricks & O'Keefe,
53 2002).

54 On conventional tractors, accessories are belt-driven, which means they are driven by the
55 engine through the serpentine belt, so their speeds are linearly dependent by the engine
56 crankshaft speed. In this solution, accessories are controlled by the engine crankshaft speed and
57 not by the parameter they control (e.g. engine coolant temperature for the fan drive). For this

58 reason, accessories are sized to meet performance requirements at a certain engine speed, and
59 this may lead to two side effects (Campbell & Kittelson, 2009):

- 60 • **accessory overdrive:** when more power is delivered to an accessory than required
61 by the function.
- 62 • **parasitic loading:** when accessories demand power even when no useful output is
63 required.

64 These side effects lead to notable energy losses, especially in certain conditions such as
65 engine idling (Andersson, 2004; Hnatzuk et al., 2000). In this engine status, the impact of
66 accessories on engine fuel rate is large and a previous study reported a fuel rate increases up to
67 170 % when the HVAC compressor is engaged (Brodrick et al., 2002). Prolonged idle and high
68 accessory loads are causes for great parasitic energy losses, which can be a non-marginal
69 percentage of the delivered engine energy (Bass & Alfermann, 2003). Due to the above-
70 mentioned side effects, accessories are designed to control a bit their operations in certain
71 conditions, so their power demands can be reduced. For example:

- 72 • Alternators operate as a constant voltage source, so the delivered current is proportional
73 to the voltage differential with the battery. However, their efficiencies are typically low
74 (i.e. 60 %) (Perreault & Caliskan, 2004). Alternators do not include any strategy for
75 reducing parasitic loading when the battery reaches the full state-of-charge. A feasibility
76 study on controlling the engagement of the alternator with a magnetic clutch was carried
77 out, and up to 6 % of fuel-saving with respect to a conventional system was observed
78 (Sales, Sousa, Monteiro, & Rodrigues, 2018).
- 79 • The fan drives may integrate viscous couplings with ON/OFF or modulating behaviours
80 (Pierce & Shepherd, 1982). The coupling is controlled by the engine management
81 system (EMS), which controls the rotational speed of the fan as a function of the
82 temperatures of engine coolant, lubricating oil, etc. (Buning, 2010; Lee et al., 2010).

83 With respect to conventional fan drives, viscous couplings permit to reduce the fuel rate
84 up to 6 % and increase the useful engine power up to 10 % (Lee et al., 2011).

85 • The HVAC compressor can be disengaged through a magnetic clutch when the
86 temperature requirement is met or when the evaporator temperature is below a certain
87 value in order to prevent any HVAC damage (Shukla et al., 2018). The angular speed
88 of the compressor determines the flow rate of the refrigerant that circulates in the
89 system, and consequently, the heat transfer rate at the evaporator and condenser. The
90 HVAC compressor demand is about 4 % of the power delivered by the engine (Hahn,
91 2008). Since the HVAC cooler typically operates in highly transient conditions due to
92 the ON/OFF clutch control strategy, considerable energy saving was obtained by
93 adopting supervisory energy management algorithms (Zhang et al., 2016).

94 • The brake air compressor is part of an open-loop circuit and it is controlled to maintain
95 the pressure in the brake tank at a certain value. Thus, when no compressed air is needed,
96 the air is released into the atmosphere, named as “idling state” in the following. This
97 leads to an ON/OFF duty cycle, with prolonged circuit idling and significant energy
98 losses (Buning, 2010).

99 Increases in fuel efficiency can be achieved by decoupling the accessories from the engine;
100 thus, the impact of parasitic loadings and accessory overdrives can be reduced. This approach
101 is not straightforward on conventional tractors, where the secondary energy source is the
102 starting, lighting, and ignition battery, which is a low voltage electrical system, designed to
103 power up to 1 kW of electricity (Emadi, Williamson, & Khaligh, 2006). Therefore, heavy-duty
104 vehicle manufacturers have been investigating the potential fuel saving by adding different
105 power supplies in vehicles, allowing them to take advantages of their different characteristics
106 (Hamandi, Pèra, & Hissel, 2002). This is the principle of hybridisation, and two solutions have
107 been investigated on heavy-duty vehicles: powering the accessories with auxiliary power units

108 (APUs) or replacing belt-driven accessories with electrically-driven ones. APUs are usually
109 small diesel engines designed to power the accessories near their peak efficiencies. Thus, APUs
110 are more efficient than the main engine at idle in powering the accessories. APUs are usually
111 installed in heavy-duty trucks because these trucks are exposed to extremely prolonged idling
112 periods for maintaining the battery voltage and the cab temperature at the desired levels when
113 drivers sleep overnight inside the truck. APUs permit to run accessories even if the engine is
114 turned off, and a study reported that APUs permit to reduce idling fuel rate up to 80 % with
115 respect to belt-driven accessories (Lim, 2003).

116 Electrically-driven accessories are typically installed on hybrid vehicles, and they can be
117 completely controlled on-demand, so they can be switched ON and OFF, or speed up and down
118 (Moreda, Muñoz-García, & Barreiro, 2016). Even if part of the energy is lost for converting
119 mechanical energy into an electrical energy, several studies report that moving from belt-driven
120 accessories to electrically-driven ones permits to increase the fuel economy of vehicles in real
121 operating conditions. The advantages are vehicle dependent, and in a minivan an increase of
122 fuel economy up to 9.2 % was observed (Kluger & Harris, 2007), while on a hybrid bus a
123 reduction in the accessory power demand up to 36 % was observed (Campbell & Kittelson,
124 2009). This difference is probably caused by the fact that buses require larger HVAC
125 compressors than minivans. Considering that electrically-driven accessories could be run in
126 function of the actual performance requirement, accessories and ICEs could be downsized, and
127 this leads to a further lower fuel rate (Serrao et al., 2011).

128 The replacement of belt-driven accessories with electrically-driven ones seems to be the
129 best practical way to reduce tractors' fuel consumption and emissions, but it may lead to more
130 complicated powertrains. To design more efficient powertrains, a greater understanding of
131 where the fuel energy is demanded is of utmost importance. Thus, fuel-saving potentials should
132 be carefully evaluated on real-world data since the power demands of accessories are strongly

133 dependent on tractor operating conditions (Troncon et al., 2019). This data is not reported in
134 any study.

135 This paper aims to quantify the power demands of the accessories in agricultural tractors
136 under real-world operating conditions.

137 **2. Materials and methods**

138 The tests were carried out with a Steyr 6230 CVT tractor (CNH Industrial N.V., Amsterdam,
139 NL) during its regular use. Its engine main specifications are reported in Table 1. Data were
140 recorded for forty-five days at the Agricultural Farm of the University of Bologna (Cadriano,
141 BO, Italy). In total, the tractor was used for 137 *h*, and during that time the tractor travelled for
142 1212 *km*.

Maximum engine power	(<i>kW</i>)	192
Number of cylinders	(–)	6
Engine tier	(–)	4B
Transmission	(–)	Continuously variable transmission
Battery	(<i>V</i>)	14

143 **2.1 Accessories' specification**

144 Accessories are located at the front of the tractor, fixed on the engine block, and they are
145 driven by the engine through the serpentine belt.

146 The alternator is always engaged when the tractor is running, with a fixed transmission ratio.
147 The alternator current varies in order to keep the battery at a fixed voltage, and it depends on
148 the electrical loads. A voltage regulator controls the current distributed from the alternator to
149 the battery in order to control the charging process.

150 The fan drive integrates a modulating viscous coupling which continuously controls the fan
151 angular speed in function of the temperatures of engine coolant, transmission, and lubricating
152 oils. The viscous coupling is controlled in a closed control loop with a PID controller which

153 limits the temperature to set-point values. The PID governor can control the transmission ratio
154 between the engine crankshaft and the fan (τ_{fan}) in the range between 0.75 (maximum
155 modulation) and 1.25 (minimum modulation).

156 The HVAC compressor is a constant displacement compressor with a magnetic clutch
157 controlled by the HVAC controller to realise a variable cooling capacity. The cooling
158 requirement inside the cab is sensed by a thermistor that sends a signal to the HVAC controller,
159 which controls the engagement of the magnetic clutch in order to meet the cooling requirement
160 of the driver. Moreover, the HVAC controller disengages the clutch when the evaporator
161 temperature is below 3 °C, and it reengages the clutch when the temperature rises above 5 °C
162 in order to prevent any damage of the evaporator. Moreover, the driver can operate directly on
163 a switch placed into the cabin, choosing between four different configurations: one automatic
164 state, and three manual states with three different blower speeds. On automatic state, the
165 compressor can vary the mass flow rate of the refrigerant and the cooling requirement in order
166 to meet the temperature set-point controlled by the driver (Shukla et al., 2018). Fig. 1 shows
167 the HVAC compressor in the tractor architecture.

168 *Fig. 1:*
169

170 The brake air compressor is integrated into a circuit with a feedback signal coming from a
171 pressure transducer on the tank. When the tank pressure is below 7 bar, the compressor works
172 to add air to the tanks, otherwise the compressor releases air into the atmosphere.

173 **2.2 Acquisition system – CAN-Bus data logger**

174 The test was carried out according to the approach introduced by Molari et al. (2013). Tractor
175 CAN-Bus data were recorded through a stand-alone CAN-Bus data-logger optimized by CNH
176 Industrial. The CAN-Bus data logger automatically recorded all the CAN-Bus messages

177 anytime the tractor engine was turned on; thus, data were collected in totally uncontrolled
178 conditions in order to not interfere with any farming activities.

179 Auxiliary sensors were connected to the CAN-Bus data-logger to record information which
180 was not otherwise available into the CAN-Bus network. In particular:

- 181 • Alternator output current (I_{alt}): measured with a current sensor (HTA200S, LEM GmbH,
182 Europe) placed in the positive terminal of the alternator (Fig. 2).
- 183 • Trailer brake compressor pressure (p_{tb}): measured with a pressure sensor (PX3AG1BH
184 010BSAAX, Honeywell International Inc., Europe) placed in the compressor head (Fig.
185 3).
- 186 • Air intake pressure (p_{in}): measured with a pressure sensor (PX3AG1BH010BSAAX,
187 Honeywell International Inc., Europe) placed at the compressor inlet.
- 188 • Trailer brake tank pressure (p_{tk}): measured with a pressure sensor
189 (PX3AG1BH010BSAAX, Honeywell International Inc., Europe) placed in the
190 compressor tank.

191 Each auxiliary sensor was connected to the CAN-Bus data logger through a CSM ADMM
192 pro 2 (CSM GmbH, Filderstadt, Germany), which is a CAN-Bus interface for analogue signals.

193

194

Fig. 2

195

196

Fig. 3

197

198 For the purpose of this study, the CAN-Bus signals with the Suspect Parameter Numbers
199 (SPNs) and Parameter Group Numbers (PGNs) (ISO, 2012; SAE, 2013) reported in Table 2
200 were used for the analysis.

Table 2 – Signal identifications

Name	SPN	PNG	Definition	Id.	Unit
Engine speed	190	61444	Revolution speed of the engine crankshaft	n_{eng}	rpm
Ground-based speed	1859	65097	Actual ground speed of the tractor	v_a	$m\ s^{-1}$
Battery potential	168	65271	Battery voltage	V_{bat}	V
Engine reference torque	544	65251	Peak torque used as a reference value for all indicated engine torque parameters	M_{er}	$N\ m$
Actual engine percent torque	513	61444	Engine torque as a per cent of reference engine torque. The value includes the torque developed in the cylinders required to overcome friction	$M_{eng\%}$	$[-]$
Nominal friction-percent torque	514	5398	Torque contribution of frictional and thermodynamic losses of the engine itself, pumping torque loss and the losses of fuel, oil, and cooling pumps	$M_{f\%}$	$[-]$
Fan speed	1639	65213	Revolution speed of the fan associated with the engine coolant system	n_{fan}	rpm
Estimate percent fan speed	975	65213	The ratio of the fan drive (current speed) to the fully engaged fan drive (maximum fan speed)	β_{fan}	$[-]$
Engine coolant temperature	110	65262	The temperature of the liquid in engine the engine cooling system	T_{fan}	$[-]$
HVAC compressor engagement	1351	65198	Engagement of the air compressor: 0 when the compressor is engaged, and 1 when the compressor is disengaged	γ_{HVAC}	$[-]$
HVAC switch status	7853	64993	State of the control switch inside the cab. There is an automatic airflow ratio modulation state, and three fixed airflow rate levels named as “Speed1”, “Speed2”, and “Speed3” hereafter	S_{HVAC}	$[-]$
Actual cabin temperature	170	65269	The temperature inside the cab	T_{cab}	$^{\circ}C$
Setpoint cabin temperature	1691	57344	Temperature set by the operator in the cab	T_{sp}	$^{\circ}C$
Engine fuel rate	183	65266	Amount of fuel consumed per unit of time	FR	$l\ h^{-1}$

201

202 All the data recorded when the tractor was run for less than 300 s in a day were excluded

203 from the analysis, which could be caused by a non-operational use of the tractor, such as

204 downloading data from the CAN-Bus data-logger. The recorded signals were interpolated at
205 0.1 s through a spline interpolation algorithm.

206 2.3 Signal post-processing

207 The engine power (P_{eng}) was calculated with the equation (1):

$$P_{eng} = \left(\frac{2 \pi n_{eng}}{60} \right) \cdot M_{er} \cdot \frac{(M_{eng\%} - M_{f\%})}{100} \quad (1)$$

208

209 The alternator power demand (P_{alt}) was calculated with the equation (2):

$$P_{alt} = \frac{(V_b \cdot I_{alt})}{\eta_{alt}} \quad (2)$$

210 η_{alt} being the alternator efficiency and it was calculated with equation (3):

$$\eta_{alt} = a_{alt} + b_{alt} n_{alt} + c_{alt} n_{alt}^2 + d_{alt} n_{alt}^3 \quad (3)$$

211 a_{alt} , b_{alt} , c_{alt} and d_{alt} being regression coefficients calculated by fitting the efficiency curve

212 of the alternator (provided by the manufacturer). n_{alt} is the alternator angular speed and it was

213 calculated with equation (4):

$$n_{alt} = n_{eng} \tau_{alt} \quad (4)$$

214 τ_{alt} being the transmission ratio between the engine crankshaft, and the alternator shaft.

215 The fan power demand (P_{fan}) was calculated with the equation (5):

$$P_{fan} = a_{fan} n_{fan} + b_{fan} n_{fan}^2 + c_{fan} n_{fan}^3 \quad (5)$$

216 a_{fan} , b_{fan} and c_{fan} being regression coefficients calculated from test bench data. n_{fan} is the

217 fan angular speed.

218 The HVAC compressor power demand (P_{HVAC}) is dependent on refrigerant charge level,

219 blower speed, actual ambient temperature, and compressor speed; however, the compressor

220 speed is much more influential than the others (Joudi et al., 2003; Macagnan et al., 2013). To
 221 limit the number of sensors to install in the tractor, a pragmatic approach was adopted in this
 222 study, where P_{HVAC} was calculated with the equation (6), considering only the HVAC
 223 compressor speed (n_{HVAC}) (Zhong, 2018), and γ_{HVAC} :

$$P_{HVAC} = (a_{HVAC} + b_{HVAC} n_{HVAC} + c_{HVAC} n_{HVAC}^2) \gamma_{HVAC} \quad (6)$$

224 a_{HVAC} , b_{HVAC} , and c_{HVAC} being regression coefficients calculated from test bench data. n_{HVAC}
 225 the HVAC compressor angular speed and it was calculated with equation (7):

$$n_{HVAC} = n_{eng} \tau_{HVAC} \quad (7)$$

226 τ_{HVAC} the transmission ratio between the engine crankshaft and the HVAC compressor pulley.

227 The brake air compressor power (P_{BAC}) is dependent by the compressor outlet flow rate,
 228 which in turn depends on its swept volume (V_{BAC}), and its angular speed (n_{BAC}); the inlet
 229 pressure on the air intake manifold (p_{in}); and, finally, the compressor outlet pressure ($p_{tb} +$
 230 p_{atm}), where p_{atm} is the atmospheric pressure. Considering an adiabatic process, P_{BAC} was
 231 calculated with the formula (8):

$$P_{BAC} = n_{BAC} \frac{k}{k-1} p_{in} V_{BAC} \left[\left(\frac{p_{tb} + p_{atm}}{p_{in}} \right)^{\frac{k-1}{k}} - 1 \right] \frac{1}{\eta_{BAC}} \quad (8)$$

232 k being the heat capacity ratio and η_{BAC} being the compressor efficiency. n_{BAC} was calculated
 233 with equation (9):

$$n_{BAC} = n_{eng} \tau_{BAC} \quad (9)$$

234 τ_{BAC} being the transmission ration between the engine crankshaft and the pulley of the brake
 235 air compressor. The brake air compressor operates in idle state when p_{tb} is lower than 0.1 bar,
 236 otherwise it operates in the load state.

237

238 In Fig. 4, the characteristic curve of each accessory is reported as a function of n_{eng} . The
239 fan drive, and the alternator can operate on an area due to their embedded load control strategies.
240 In particular, for the alternator, the area is given by a set of constant voltage curves, where the
241 upper boundary curve was obtained with a voltage differential of 13 V ($\pm 0.5\%$). For the fan
242 drive, the area where the fan operates is limited by the two extreme values of τ_{fan} . On the other
243 hand, the HVAC compressor runs on a characteristic curve according to n_{eng} , and the brake air
244 compressor runs on the two different characteristics curves.

245 *Fig. 4*
246

247 The total accessory power demand (P_{aux}) was calculated with equation (10):

$$P_{aux} = P_{alt} + P_{fan} + P_{HVAC} + P_{BAC} \quad (10)$$

248

249 Since the power losses of the tractor are strongly dependent on the operating conditions
250 (OCs), the recorded data were classified according to the classification scheme developed by
251 Mattetti et al. (2020). In this classification scheme, three different operating conditions were
252 introduced:

- 253 • *idle*: standing tractor with no use of both PTOs, and with $n_{eng} < 850 \text{ rpm}$.
- 254 • *moving*: moving tractor with no use of PTOs, and three-point linkages.
- 255 • *field work*: tractor operating on the field when a repetitive pattern of a pass, headland
256 turn and pass was observed.

257 In Fig. 5, a portion of the P_{eng} signal classified with the before mentioned tractor
258 classifications is reported.

259 *Fig. 5*
260

261 For each operating condition, the average values of all the above-mentioned signals were
 262 calculated. All these values will be denoted with the subscript OC (e.g., $P_{eng,OC}$ denotes the
 263 average values of P_{eng} in a generic operating condition). For intermittently used accessories,
 264 which are the HVAC and the brake air compressor, their duty cycles for each operating
 265 condition (D_{OC}) were calculated with the equation (11):

$$D_{OC} = \frac{t_{wc}}{t_{pc}} \quad (11)$$

266
 267 t_{pc} being the pulse period of each cycle, t_{wc} being the pulse width of each cycle and different
 268 operating conditions (Fig. 6).

269 *Fig. 6*

270
 271 The average duty cycle (\bar{D}_{OC}) was calculated for each operating condition with the equation
 272 (12):

$$\bar{D}_{OC} = \frac{1}{N_{OC}} \sum_{i=1}^{N_{OC}} D_{OC,i} \quad (12)$$

273
 274 being N_{OC} the number of cycles of each operating condition.

275 Finally, for each operating condition, the fuel consumed by each accessory ($FC_{*,OC}$) was
 276 calculated with equation (13):

$$FC_{*,OC} = \frac{P_{*,OC}}{P_{eng}} \int_0^{t_{oc}} FR_{OC} \cdot dt \quad (13)$$

277

278 t_{OC} being the duration of the recorded data during each operating condition, $FR_{*,OC}$ and
279 $P_{*,OC}$ being the engine power and fuel rate for each operating condition, respectively. The
280 asterisk in the subscript standing for the consumed fuel or power of a generic accessory.

281 **3. Results and discussion**

282 The discussion will start by reporting the behaviour of each accessory, and then their
283 contribution to engine power demands and fuel consumption will be reported.

284 **3.1 Alternator**

285 In Fig. 7 – left, the joint probability distribution between I_{alt} and V_{bat} is reported. Indeed, a
286 mild negative correlation can be observed between I_{alt} and V_{bat} (the Pearson's correlation
287 coefficient is -0.52). That is because the alternator is designed to operate as a constant voltage
288 source, so the higher the voltage differential is between the battery and the alternator, the greater
289 is I_{alt} . I_{alt} is strongly dependent on the electrical load which in case of the tractor understudy
290 is mostly caused by the lights, the blower fan of the HVAC, and the radio. The voltage supplied
291 by the alternator is such as to allow V_{bat} to range between 13.5 and 13.8 V. The distribution
292 has a single mode, which is located at 13.6 V and 38.5 A, which occurred 10.5 % of the time.
293 In Fig. 7 – right, the joint probability distribution between P_{alt} and n_{eng} is reported. However,
294 P_{alt} is function of n_{alt} , which in turn depends on n_{eng} , so P_{alt} is poorly correlated with n_{eng}
295 (the Pearson's correlation coefficient is 0.36). This is due by the fact that I_{alt} mostly depends
296 on the electrical load demand and battery-state-of-charge, and both are independent on n_{eng} .
297 The joint probability distribution has a trend that leads back to the alternator characteristic curve
298 (Fig. 4), but much lower values of P_{alt} were recorded due to the greater state of charge of the
299 battery. Indeed, the maximum value of P_{alt} is 1.5 kW, but the alternator could generate up to
300 5 kW (Fig. 4). On average, P_{alt} on idle is lower than P_{alt} on greater n_{eng} ; indeed, the two major

301 peaks of the distribution, located at 875 *rpm* and 2053 *rpm* occur at 0.7 *kW* and 0.9 *kW*,
302 respectively (Fig. 7 – right). This is caused by the fact that the engine mostly operates at the
303 idling speed and at the speed where the maximum power occurs.

304 *Fig. 7*
305

306 3.2 Fan drive

307 In Fig. 8 – left, the joint probability distribution between n_{fan} and n_{eng} is reported. The
308 upper and bottom boundaries of the distribution represent the two extreme operating conditions
309 with minimum and maximum fan drive modulation, respectively. The area between the two
310 boundaries represents the points where intermediate fan drive modulations occurred. One can
311 note that the engine idle condition with minimum modulation is the point where the joint
312 probability is the highest. That is because, differently from the other operating conditions, idling
313 occurs at a specific n_{eng} . All the modes of the distribution are located along the minimum
314 modulation boundary at 800 *rpm*, 1350 *rpm*, and 2050 *rpm*. In Fig. 8 – right, the joint
315 probability distribution between T_{fan} and n_{eng} is reported. The distribution resembles a typical
316 shape of a viscous coupling (Buchholz, 2005). Indeed, when T_{fan} was below 80 °C, the fan
317 drive runs at around 1000 *rpm* (i.e., fan drive idling speed) for reducing the fuel rate. When
318 T_{fan} rose above 80 °C, the fan drive was activated, and n_{fan} was increased. In the vertical band
319 of the distribution, the ECU controls the fan drive behaviour, and the width of this band permits
320 to quantify the system hysteresis, which is about 10 °C (considering only the point where the
321 higher probability occurs), and this amount is aligned to that reported in other studies (Pierce
322 & Shepherd, 1982). Fan drive hysteresis is necessary to prevent high switching frequency which
323 may results in damage of the viscous coupling. T_{fan} ranges from 30 °C (i.e. ambient

324 temperature) to 105 °C (which is close to the engine coolant warning limit). In the distribution,
325 two major modes can be distinguished, located at 1076 rpm, and 1700 rpm of n_{fan} .

326 *Fig. 8*
327

328 The time percentage of β_{fan} , divided into its three main use configurations:

- 329 • maximum modulation when $\beta_{fan} = 0$;
- 330 • intermediate modulation when $0 < \beta_{fan} < 1$;
- 331 • minimum modulation when $\beta_{fan} = 1$.

332 The fan drive worked at minimum modulation condition for 49% of the time and at
333 intermediate modulation condition for 45%. Moreover, the fan drive worked at maximum
334 modulation state only for 9% of the time, and especially only for short transitional periods,
335 during the cold-start and warm-up phases, probably because the measurements were carried out
336 in the summertime, which is the period of the year where this type of tractor is mostly used.

337 In Fig. 9, the joint probability distribution between P_{fan} and P_{eng} is reported. Even if the
338 heat generated by a combustion engine is proportional to the engine power (Ferrari, 2016), P_{fan}
339 is not correlated with P_{eng} (Pearson's correlation coefficient is 0.29). This is due to the fact that
340 viscous coupling is handled by few temperature signals and not by the engine operating point.
341 In certain conditions (i.e., engine idling), this may lead to an unwanted early cooling with a
342 decrease of engine efficiency due to large heat exchange (Haghighat et al., 2018). The
343 distribution shows an increasing trend of the P_{fan} with respect to engine load, up to 11.5 kW
344 when P_{eng} is greater than 160.0 kW. The lowest value of P_{fan} is 0.2 kW, during the ignition
345 phases.

346

Fig. 9

347 **3.3 HVAC compressor**

348 In Fig. 10, portions of S_{HVAC} , γ_{HVAC} , t_{cab} , and P_{HVAC} signals are reported. From the analysis
 349 of the signals' behaviours, it can be noted that when the switch is on "Auto" and the difference
 350 between T_{cab} and T_{sp} (named ΔT hereafter) is lower than 5°C , it leads to a cycling clutch
 351 operation, which causes oscillations in P_{HVAC} . Cyclic fluctuations are quite frequent even during
 352 HVAC manual states (i.e. "Speed 1", "Speed 2", and "Speed 3" settings), where the blower
 353 speed is fixed and the HVAC compressor does not vary the cooling capacity.

354

Fig. 10

355

356 During the test period, the HVAC compressor was engaged for 82.5 % of the time, of which:
 357 61 % on "Auto" state, 4 % on "Speed 1" state, 17 % on "Speed 2", and 0.5 % on "Speed 3"
 358 state. The prolonged use of the HVAC compressor is caused by the high temperature recorded
 359 in the period of the test. Indeed, T_{cab} ranged from 18.6°C to 43.3°C ; in particular, for 25 %
 360 of the time, Δt was lower than 5°C .

361 In Table 3, the average values of tp_c and D_{OC} are reported. For both automatic and manual
 362 states, tp_c is approximately the same, at around 120 s. However, on manual states, the HVAC
 363 was engaged longer than the automatic state, since D_{OC} on manual states is higher than the
 364 automatic state: this is probably since on manual states the ΔT was on average 10.1°C , and on
 365 automatic states Δt was on average 7.7°C .

Table 3 – Average parameters of HVAC compressor duty cycle for the operating conditions
 * Automatic state; ** Manual states.

<i>OC</i>	<i>tp_i</i> [s]	<i>D_i</i> [%]
Idle	120*/118**	53*/70**
Moving	136*/135**	54*/65**
Field work	137*/114**	63*/71**

366

367 **3.4 Brake air compressor**

368 In Fig. 11, a portion of the p_{tb} , p_{tk} , and P_{BAC} signals are reported. For the compressor, an
369 ON/OFF duty cycle can be observed. When p_{tk} drops below 7 bar, the compressor is in the
370 load state and p_{tb} rises from 0.2 bar to 10.8 bar. Similarly, P_{BAC} varies in function of the
371 circuit load state and n_{BAC} (see formula (8)). Indeed, the peak values of P_{BAC} in idling and
372 loading states are 0.1 kW and 2.9 kW, respectively. When the brake air compressor is on idling,
373 it does not produce any useful work. The higher n_{BAC} is and the higher the air flow rate provided
374 by the brake compressor is, the lower the time required for refilling the tank is.

375

376 *Fig. 11*

377

378 The brake air compressor operated in the idle state for 98 % of the time, and in Fig. 11, its
379 intermittent behaviour is highlighted. In Table 4, the average values of tp_c and D_{OC} are
380 reported. Every few minutes, the brake air compressor switches into a load state and is in this
381 state for around a minute. The frequency of the switching state of the brake air compressor
382 should be dependent on the air losses inside the tank, and on frequency of actuation of trailer
383 brake. However, this effect was not observed during the test.

Table 4 – Average parameters of brake air compressor duty cycle for the operating conditions

<i>OC</i>	<i>tp_i</i> [s]	<i>D_i</i> [%]
Idle	130	38
Moving	187	29
Field work	236	38

384 **3.5 Comparative analysis of accessories power demand**

385 In Table 5, for each tractor operating condition, the time contribution and the average values of
386 n_{eng} and P_{eng} are reported. The tractor operated most of the time (i.e., 66 %) for field work

387 operating conditions. Thanks to the data coming to beacon scanner, the tractor operated for 49%
 388 of time with a plough, for 33% of time with a cultivator and for 11% of time with a trailer. In
 389 the rest of time the tractor operated without any implement, mostly for road transportation.

390 In this operating condition, much greater values of the average value of $P_{eng,OC}$ than those
 391 of the other operating conditions were observed. Even if the average value of $P_{eng,OC}$ during
 392 moving condition is 57% of that of the field work, the average value of $P_{aux,OC}$ during moving
 393 conditions is 88% of that of the field work operating condition. This is mostly because the two
 394 operating conditions lead to a similar average value of $n_{eng,OC}$. However, in terms of
 395 percentage, the average value of $P_{aux,OC}$ is the greatest during the idling operating condition,
 396 since the engine does not produce any useful work and most of the power from the engine is
 397 used for overcoming the engine losses (i.e., friction, thermodynamic, pumping losses, etc.) and
 398 accessory loads. During field work operating conditions, the greatest $P_{eng,OC}$ was observed, and
 399 the contribution of $P_{aux,OC}$ with respect to $P_{eng,OC}$ is only 14%.

Table 5 – Average values for each defined operating condition

* percentages of $P_{aux,OC}$ were calculated with respect to P_{eng}

<i>OC</i>	<i>Time_{OC}</i> [%]	<i>n_{e,OC}</i> [rpm]	<i>P_{eng,OC}</i> [kW]	<i>P_{aux,OC}</i> [kW/%*]
Idle	16	853	10.4	4.0/38.5
Moving	18	1444	65.0	14.5/22.3
Field work	66	1573	114.8	15.7/13.7

400

401 The histogram of Fig. 12– left reports the average power demand of each accessory during
 402 each operating condition. Each operating condition leads to a different engine operating point
 403 and consequently to a different accessory power demand. The HVAC compressor and the fan
 404 drive are the accessories with the greater power demands in any operating condition, while the
 405 alternator and the brake compressor are by far less demanding. During the engine idling, the
 406 HVAC compressor is the most demanding accessory, while in all the others, the fan drive is the
 407 most demanding. Indeed, when n_{eng} is lower than 1300 rpm, P_{HVAC} is greater than P_{fan} (Fig.

408 4). The power demand of the brake air compressor is in any condition the lowest, even if it
409 could reach significant peak values (Fig. 4). That is because the brake air compressor was in
410 idle over prolonged periods, and this reduces the average value of its power demand.

411 The results reported in this paper can be compared with other types of heavy-duty vehicles,
412 long haulage trucks and city buses, where their accessory power demands are around 5 – 7 %,
413 respectively (Sjostedt et al., 2014). This great difference is probably caused by the fact that the
414 cooling fan provides the major contribution to engine heat controls due to the lower ground
415 speed of agricultural tractors than on-road vehicles. Considering that tractors of this class are
416 used up to 850 hours per year (Mattetti, Maraldi, et al., 2019), a total of 18613 l of fuel were
417 consumed. Of that, 15.4 % were consumed for accessories. In Fig 12 – right. the yearly fuel
418 consumption caused by each accessory and each operating condition are reported. Most of the
419 fuel is consumed during the field work operating state by the fan drive and the HVAC
420 compressor, while the alternator and the brake air compressor are responsible for only 2.1 % of
421 the total fuel consumption.

422

423

Fig. 12

424

425

4. Conclusions

426 The current agricultural market trend is pushing tractor manufactures and research centres
427 to investigate hybrid configurations in order to increase tractor fuel efficiency. In the last
428 decades, research and manufacturers mostly focused on engine combustion efficiency, but more
429 recently, efforts have been made to investigating on the efficiency of all power demands in
430 powertrains. On tractors, a non-negligible part of the power is absorbed by accessories due to
431 their size. However, their operational requirements are different than the nominal and no studies

432 had quantified the amount of fuel consumed by the accessories and their environmental impact
433 on the real operating condition. In the period under study, the tractor consumed 462 l of fuel
434 only for the accessories. Tractors of this class are used up to 850 hours per year, so accessories
435 are responsible for yearly fuel consumption of 2866 l . This consumption can be reduced by
436 equipping tractors with electrically-driven accessories, which can be turned off when they are
437 not needed. For example, an electrical brake air compressor can be turned off when the brake
438 air tank is full, and in that case, a yearly fuel savings of 34.3 l could be achieved. Electrically-
439 driven accessories can also be run with the engine off. This is especially useful for reducing the
440 need for unnecessary idling, which is accounted for 67% of the entire idling (Molari et al.,
441 2019). In that case, it is expected to save 216 l of fuel every year. The total benefit will be a
442 fuel savings of 232 l, which is 8.1% of the entire fuel consumed for accessories. Even though
443 this amount seems to be negligible, one should consider that 1.2 million of tractors of this class,
444 in terms of engine power, are in use in the U.S. (Perdue & Hamer, 2019), and thus a fuel savings
445 of 278 million litres could be obtained per year, which would lead to a reduction of CO₂
446 emissions of about 747 thousand tons per year (2.69 kg of CO₂ per litre of diesel fuel was used
447 of conversion factor). This can lead to a significant improvement in tackling climate change.
448 The estimated savings were calculated assuming that all tractors are used with the same duty
449 cycle, but this may not occur. So, the actual savings may be significantly different than those
450 reported in this study. However, this study is the first comprehensive study on tractor
451 accessories and the first one that attempted to quantify the potential fuel savings caused by the
452 electrification of accessories. This figure will permit to evaluate if farming may benefit from
453 tractors with electrically driven accessories.

454 In the coming years, researchers and tractor manufacturers should think of novel solutions
455 for accessories which permit to increase the tractor fuel efficiency and to add new
456 functionalities, otherwise farmers could perceive electrically-driven accessories as solutions

457 which lead to the higher purchasing price. Future research will be required to investigate the
458 power losses and fuel-saving, with related emissions, in tractors with hybrid powertrains, to
459 compare their efficiency. In addition, the use of dedicated equipment to accurately assess the
460 energy demand of accessories: for example, heat meter can be installed to measure the heat
461 exchange and the real demand of HVAC compressor.

462

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585 Captions

586 *Fig. 1: HVAC compressor, situated on the right side of the tractor, behind the fan drive*
587

588 *Fig. 2: current sensor installed at the alternator output. Alternator current sensor is in the*
589 *yellow circle.*
590

591 *Fig. 3: the pressure sensor used for monitoring the pressure at the outlet in the yellow*
592 *circle.*
593

594 *Fig. 4: accessories' characteristic curves of the accessories in the tractor used for the test.*
595 *alternator (on top left), fan drive (on top right), heating, ventilation and air-conditioning*
596 *compressor (HVAC) compressor (on bottom left) and of brake air compressor (on bottom*
597 *right).*
598

599 *Fig. 5: example of signal classifications in function of the tractor operating conditions. A*
600 *portion of engine power (P_{eng}) (on the left), and a portion of tractor trajectory (on the right)*
601 *are reported.*
602

603 *Fig. 6: schematisation of the ON/OFF duty cycle used for the characterisation of the HVAC*
604 *and brake air compressors. t_{pc} and t_{wc} are the pulse period and the pulse width of each*
605 *cycle, respectively.*
606

607 *Fig. 7: joint probability distribution between battery potential (V_{bat}) and alternator output*
608 *current (I_{alt}) (on left), and joint probability distribution between the engine speed (n_{eng}) and*
609 *the alternator power demand (P_{alt}) (on right). The bins where the probability is lower than*
610 *0.05% are not displayed.*
611

612 *Fig. 8: joint probability distribution between engine speed (n_{eng}) and fan speed (n_{fan}) (on*
613 *left), and engine coolant temperature (T_{fan}) and n_{fan} (on right). The bins where the*
614 *probability is lower than 0.05% are not displayed.*

615

616

617 *Fig.9: joint probability distribution between engine speed (P_{eng}) and fan drive power demand*
618 *(P_{fan}). The bins where the probability is lower than 0.05% are not displayed.*

619

620 *Fig. 10: portion of the heating, ventilation and air-conditioning compressor (HVAC) signals*
621 *used for the calculation of HVAC power demand (P_{HVAC}): HVAC switch status (S_{HVAC}) (first*
622 *from the top), HVAC compressor engagement (γ_{HVAC}) (second from the top), temperature*
623 *gradient between actual and setpoint cabin temperatures (ΔT) (third from the top), and P_{HVAC}*
624 *(forth from the top).*

625

626 *Fig. 11: a portion of the brake air compressor signals: trailer brake compressor pressure*
627 *(p_{tb}), trailer brake tank pressure (p_{tk}), and trailer brake power demand (P_{BAC}). In*
628 *particular, two engagements of the compressors can be seen.*

629 *Fig. 12: relative power demands (P_{aux}) (on left) and yearly fuel consumption (FC_{aux}) (on*
630 *right) of alternator (alt), fan drive, heating, ventilation and air-conditioning (HVAC)*
631 *compressor, and brake air compressor (BAC) according to the operating conditions.*

632