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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 CO_2 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.

KEYWORDS: engine accessories; powertrain; real-world data; fuel economy; vehicle 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
$\overline{D}_{OC}$	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

1. Introduction

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

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

On conventional tractors, accessories are belt-driven, which means they are driven by the engine through the serpentine belt, so their speeds are linearly dependent by the engine crankshaft speed. In this solution, accessories are controlled by the engine crankshaft speed and not by the parameter they control (e.g. engine coolant temperature for the fan drive). For this reason, accessories are sized to meet performance requirements at a certain engine speed, and this may lead to two side effects (Campbell & Kittelson, 2009):

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

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

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

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

- The HVAC compressor can be disengaged through a magnetic clutch when the temperature requirement is met or when the evaporator temperature is below a certain value in order to prevent any HVAC damage (Shukla et al., 2018). The angular speed of the compressor determines the flow rate of the refrigerant that circulates in the system, and consequently, the heat transfer rate at the evaporator and condenser. The HVAC compressor demand is about 4 % of the power delivered by the engine (Hahn, 2008). Since the HVAC cooler typically operates in highly transient conditions due to the ON/OFF clutch control strategy, considerable energy saving was obtained by adopting supervisory energy management algorithms (Zhang et al., 2016).
- The brake air compressor is part of an open-loop circuit and it is controlled to maintain the pressure in the brake tank at a certain value. Thus, when no compressed air is needed, the air is released into the atmosphere, named as "idling state" in the following. This leads to an ON/OFF duty cycle, with prolonged circuit idling and significant energy losses (Buning, 2010).

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

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

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

The replacement of belt-driven accessories with electrically-driven ones seems to be the best practical way to reduce tractors' fuel consumption and emissions, but it may lead to more complicated powertrains. To design more efficient powertrains, a greater understanding of where the fuel energy is demanded is of utmost importance. Thus, fuel-saving potentials should be carefully evaluated on real-world data since the power demands of accessories are strongly dependent on tractor operating conditions (Troncon et al., 2019). This data is not reported in any study.

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

2. Materials and methods

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

Table 1 – Main specifications of the tractor engine

Maximum engine power	(kW)	192
Number of cylinders	$(-)$	6
Engine tier	$(-)$	4B
Transmission	$(-)$	Continuously variable transmission
Battery	(V)	14

2.1 Accessories' specification

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

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

The fan drive integrates a modulating viscous coupling which continuously controls the fan angular speed in function of the temperatures of engine coolant, transmission, and lubricating oils. The viscous coupling is controlled in a closed control loop with a PID controller which limits the temperature to set-point values. The PID governor can control the transmission ratio between the engine crankshaft and the fan (τ_{fan}) in the range between 0.75 (maximum modulation) and 1.25 (minimum modulation).

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

Fig. 1:

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

2.2 Acquisition system – CAN-Bus data logger

The test was carried out according to the approach introduced by Molari et al. (2013). Tractor CAN-Bus data were recorded through a stand-alone CAN-Bus data-logger optimized by CNH Industrial. The CAN-Bus data logger automatically recorded all the CAN-Bus messages anytime the tractor engine was turned on; thus, data were collected in totally uncontrolled conditions in order to not interfere with any farming activities.

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

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

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

Fig. 2

Fig. 3

For the purpose of this study, the CAN-Bus signals with the Suspect Parameter Numbers (SPNs) and Parameter Group Numbers (PGNs) (ISO, 2012; SAE, 2013) reported in Table 2 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}	$°C$
Setpoint cabin temperature	1691	57344	Temperature set by the operator in the cab	T_{sp}	$°C$
Engine fuel rate	183	65266	Amount of fuel consumed per unit of time	FR	$l\ h^{-1}$

All the data recorded when the tractor was run for less than $300\ s$ in a day were excluded from the analysis, which could be caused by a non-operational use of the tractor, such as downloading data from the CAN-Bus data-logger. The recorded signals were interpolated at $0.1\ s$ through a spline interpolation algorithm.

2.3 Signal post-processing

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

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

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

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

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

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

a_{alt}, b_{alt}, c_{alt} and d_{alt} being regression coefficients calculated by fitting the efficiency curve of the alternator (provided by the manufacturer). n_{alt} is the alternator angular speed and it was calculated with equation (4):

$$n_{alt} = n_{eng}\ \tau_{alt} \tag{4}$$

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

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

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

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

The HVAC compressor power demand (P_{HVAC}) is dependent on refrigerant charge level, blower speed, actual ambient temperature, and compressor speed; however, the compressor speed is much more influential than the others (Joudi et al., 2003; Macagnan et al., 2013). To limit the number of sensors to install in the tractor, a pragmatic approach was adopted in this study, where P_{HVAC} was calculated with the equation (6), considering only the HVAC compressor speed (n_{HVAC}) (Zhong, 2018), and γ_{HVAC}:

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

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

$$n_{HVAC} = n_{eng}\, \tau_{HVAC} \tag{7}$$

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

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

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

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

$$n_{BAC} = n_{eng}\, \tau_{BAC} \tag{9}$$

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

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

Fig. 4

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

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

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

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

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

Fig. 5

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

$$D_{OC} = \frac{t_{wc}}{t_{pc}} \tag{11}$$

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

Fig. 6

The average duty cycle ($\overline{D}_{OC}$) was calculated for each operating condition with the equation (12):

$$\overline{D}_{OC} = \frac{1}{N_{OC}} \sum_{i=1}^{N_{OC}} D_{OC,i} \tag{12}$$

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

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

$$FC_{*,OC} = \frac{P_{*,OC}}{P_{eng}} \int_0^{t_{OC}} FR_{OC} \cdot dt \tag{13}$$

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

3. Results and discussion

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

3.1 Alternator

In Fig. 7 – left, the joint probability distribution between I_{alt} and V_{bat} is reported. Indeed, a mild negative correlation can be observed between I_{alt} and V_{bat} (the Pearson's correlation coefficient is -0.52). That is because the alternator is designed to operate as a constant voltage source, so the higher the voltage differential is between the battery and the alternator, the greater is I_{alt}. I_{alt} is strongly dependent on the electrical load which in case of the tractor understudy is mostly caused by the lights, the blower fan of the HVAC, and the radio. The voltage supplied by the alternator is such as to allow V_{bat} to range between 13.5 and $13.8\ V$. The distribution has a single mode, which is located at $13.6\ V$ and $38.5\ A$, which occurred $10.5\ \%$ of the time. In Fig. 7 – right, the joint probability distribution between P_{alt} and n_{eng} is reported. However, P_{alt} is function of n_{alt}, which in turn depends on n_{eng}, so P_{alt} is poorly correlated with n_{eng} (the Pearson's correlation coefficient is 0.36). This is due by the fact that I_{alt} mostly depends on the electrical load demand and battery-state-of-charge, and both are independent on n_{eng}. The joint probability distribution has a trend that leads back to the alternator characteristic curve (Fig. 4), but much lower values of P_{alt} were recorded due to the greater state of charge of the battery. Indeed, the maximum value of P_{alt} is $1.5\ kW$, but the alternator could generate up to $5\ kW$ (Fig. 4). On average, P_{alt} on idle is lower than P_{alt} on greater n_{eng}; indeed, the two major peaks of the distribution, located at $875\ rpm$ and $2053\ rpm$ occur at $0.7\ kW$ and $0.9\ kW$, respectively (Fig. 7 – right). This is caused by the fact that the engine mostly operates at the idling speed and at the speed where the maximum power occurs.

Fig. 7

3.2 Fan drive

In Fig. 8 – left, the joint probability distribution between n_{fan} and n_{eng} is reported. The upper and bottom boundaries of the distribution represent the two extreme operating conditions with minimum and maximum fan drive modulation, respectively. The area between the two boundaries represents the points where intermediate fan drive modulations occurred. One can note that the engine idle condition with minimum modulation is the point where the joint probability is the highest. That is because, differently from the other operating conditions, idling occurs at a specific n_{eng}. All the modes of the distribution are located along the minimum modulation boundary at $800\ rpm$, $1350\ rpm$, and $2050\ rpm$. In Fig. 8 – right, the joint probability distribution between T_{fan} and n_{eng} is reported. The distribution resembles a typical shape of a viscous coupling (Buchholz, 2005). Indeed, when T_{fan} was below $80\ °C$, the fan drive runs at around $1000\ rpm$ (i.e., fan drive idling speed) for reducing the fuel rate. When T_{fan} rose above $80\ °C$, the fan drive was activated, and n_{fan} was increased. In the vertical band of the distribution, the ECU controls the fan drive behaviour, and the width of this band permits to quantify the system hysteresis, which is about $10\ °C$ (considering only the point where the higher probability occurs), and this amount is aligned to that reported in other studies (Pierce & Shepherd, 1982). Fan drive hysteresis is necessary to prevent high switching frequency which may results in damage of the viscous coupling. T_{fan} ranges from $30\ °C$ (i.e. ambient temperature) to $105\ °C$ (which is close to the engine coolant warning limit). In the distribution, two major modes can be distinguished, located at $1076\ rpm$, and $1700\ rpm$ of n_{fan}.

Fig. 8

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

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

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

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

Fig. 9

3.3 HVAC compressor

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

Fig. 10

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

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

OC	tp_i [s]	D_i [%]
Idle	$120^{*}/118^{**}$	$53^{*}/70^{**}$
Moving	$136^{*}/135^{**}$	$54^{*}/65^{**}$
Field work	$137^{*}/114^{**}$	$63^{*}/71^{**}$

3.4 Brake air compressor

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

Fig. 11

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

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

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

3.5 Comparative analysis of accessories power demand

In Table 5, for each tractor operating condition, the time contribution and the average values of n_{eng} and P_{eng} are reported. The tractor operated most of the time (i.e., 66 %) for field work operating conditions. Thanks to the data coming to beacon scanner, the tractor operated for 49% of time with a plough, for 33% of time with a cultivator and for 11% of time with a trailer. In the rest of time the tractor operated without any implement, mostly for road transportation.

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

OC	$Time_{OC}$ [%]	$n_{e,OC}$ [rpm]	$P_{eng,OC}$ [kW]	$P_{aux,OC}$ [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

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

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

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

Fig. 12

4. Conclusions

The current agricultural market trend is pushing tractor manufactures and research centres to investigate hybrid configurations in order to increase tractor fuel efficiency. In the last decades, research and manufacturers mostly focused on engine combustion efficiency, but more recently, efforts have been made to investigating on the efficiency of all power demands in powertrains. On tractors, a non-negligible part of the power is absorbed by accessories due to their size. However, their operational requirements are different than the nominal and no studies had quantified the amount of fuel consumed by the accessories and their environmental impact on the real operating condition. In the period under study, the tractor consumed $462\ l$ of fuel only for the accessories. Tractors of this class are used up to 850 hours per year, so accessories are responsible for yearly fuel consumption of $2866\ l$. This consumption can be reduced by equipping tractors with electrically-driven accessories, which can be turned off when they are not needed. For example, an electrical brake air compressor can be turned off when the brake air tank is full, and in that case, a yearly fuel savings of $34.3\ l$ could be achieved. Electrically-driven accessories can also be run with the engine off. This is especially useful for reducing the need for unnecessary idling, which is accounted for 67% of the entire idling (Molari et al., 2019). In that case, it is expected to save $216\ l$ of fuel every year. The total benefit will be a fuel savings of $232\ l$, which is 8.1% of the entire fuel consumed for accessories. Even though this amount seems to be negligible, one should consider that 1.2 million of tractors of this class, in terms of engine power, are in use in the U.S. (Perdue & Hamer, 2019), and thus a fuel savings of 278 million litres could be obtained per year, which would lead to a reduction of CO_2 emissions of about 747 thousand tons per year ($2.69\ kg$ of CO_2 per litre of diesel fuel was used of conversion factor). This can lead to a significant improvement in tackling climate change. The estimated savings were calculated assuming that all tractors are used with the same duty cycle, but this may not occur. So, the actual savings may be significantly different than those reported in this study. However, this study is the first comprehensive study on tractor accessories and the first one that attempted to quantify the potential fuel savings caused by the electrification of accessories. This figure will permit to evaluate if farming may benefit from tractors with electrically driven accessories.

In the coming years, researchers and tractor manufacturers should think of novel solutions for accessories which permit to increase the tractor fuel efficiency and to add new functionalities, otherwise farmers could perceive electrically-driven accessories as solutions which lead to the higher purchasing price. Future research will be required to investigate the power losses and fuel-saving, with related emissions, in tractors with hybrid powertrains, to compare their efficiency. In addition, the use of dedicated equipment to accurately assess the energy demand of accessories: for example, heat meter can be installed to measure the heat exchange and the real demand of HVAC compressor.

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Captions

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

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

Fig. 3: the pressure sensor used for monitoring the pressure at the outlet in the yellow circle.

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

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

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

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

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

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

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

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

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