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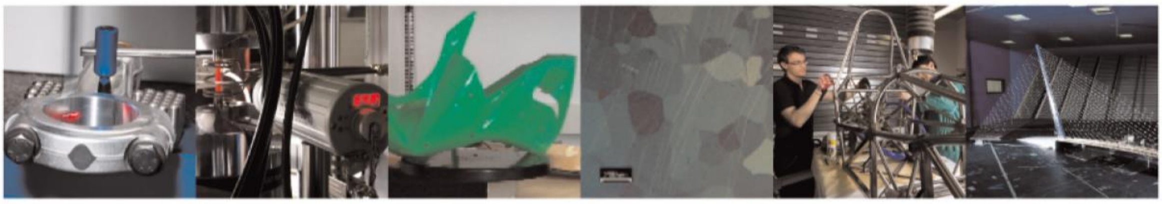

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via G. La Masa, 1 ■ 20156 Milano ■ EMAIL (PEC): pecmecc@cert.polimi.it
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Tube bending machine modelling for assessing the energy savings of electric drives technology

Paolo Albertelli[a]*,Matteo Strano[a]

[a]*Politecnico di Milano, via La Masa 1, 20156 Milan, Italy*
* Corresponding author. Tel.: +390223998557; fax: +390223998585. *E-mail address:* paolo.albertelli@polimi.it

Abstract

The aim of the work is to carry out a comprehensive analysis of the energy saving opportunities offered by the introduction of electric drives in the rotary-draw tube bending technology. Although there is a clear industrial trend towards the replacement, especially for low tonnage forming machines, of the traditionally adopted hydraulic drives, there is a lack of scientific research that has studied its implications on energy consumption. For this purpose, an energy model for tube bending machines was developed. The parameters of the model were identified exploiting experimental power measurements performed on both a hybrid hydraulic-electric and a fully electric machine.

The energy saving analysis was carried out through the updated energy models. For sake of generality, the analysis was extended considering various tube material-diameter combinations and different machine working conditions.

The results showed that relevant energy savings can be obtained and that the improvements are affected by the machine throughput. It was also observed that the minimum achievable energy saving is significantly higher (at least three times) than the energy share for processing the tube.

An efficiency analysis of both types of machines was also reported. The introduction of the electric drives allows increasing the machine efficiency up to 50 percentage points. This achievement slightly decreases with the increment of the rotary-draw bending machine throughput.

Keywords: tube bending machines; energy modelling; eco-solutions; energy efficiency;

1. Introduction

Discrete part manufacturing is an energy demanding industrial sector, that craves for more-efficient machines and production systems. This represents a stimulating challenge for machine tool designers that typically do not consider the energy consumption during the conceptual design and the full development of their machines.

For this purpose, a working group of the International Organization for Standardization (ISO) released a useful technical reference on the methodologies for designing energy-efficient machine tools, ISO 14955 (2014).

ISO 14955 reports that the energy consumed by the machine during its usage phase is typically much higher than the energy spent for the other phases of the machine life (i.e. machine production, transport, set-up and recycling). This is also confirmed by the available scientific literature, Avram and Xirouchakis (2011). The ISO standard also includes a list of suggestions that can be considered for the eco-machine tool design. The standard is mostly focused on machine tools for metal cutting, but it considers also machines for metal forming technology. In the metal forming industry, as well, environmental issues can be addressed by acting either on the manufactured product design, on the process and on the machines, Duflou et al. (2011).

Many studies on energy efficiency in machine tools can be found in the scientific literature. Unfortunately, most of them are focused on metal cutting machines. As reported in Diaz et al. (2010), energy savings can be fulfilled adopting specifically eco-designed subcomponents or implementing strategies oriented to a better machine use. The latter approach can involve both the machining strategy optimization (Yingjie (2014)) and a suitable technological parameters choice, Albertelli et al. (2016).

In metal forming, most of the researches are traditionally focused on process modelling and on the estimation of the required machine load. Very few research works are available in the literature, that deal with the eco-design of metal forming machines (i.e. press brakes, forging presses, powder forming presses, blanking machines, tube bending machines, etc.). As one of the very few published works, Santos et al. (2011) proposed a methodology for the Life Cycle Analysis LCA of hydraulic press-brakes. They demonstrated that the energy spent during the machine building phase is comparable to the energy spent during its use. This is mainly due to the discrete loading character of the sheet bending process. Devoldere et al. (2007) also developed a test case around a press brake, they underlined that a major part of the total energy consumption of the machine does not depend on the production rate, and suggested some initial design improvements.

In tube and sheet metal bending (Pulzer 1998 and Stange 1997), machines are typically equipped with Numerical Controls NC, i.e. they are so-called servo-presses. Three different kinds of drive controls can be found:

- mechanical servo-presses, also called electric machines, Osaka et al (2011) and Du and Guo (2003)
- hydraulic servo-presses, Zhao et al. (2015)
- hybrid machines, which combine hydraulic and mechanical servo-drives, Altan (1998) and Li and Tso (2008)

High tonnage presses are hydraulically driven, because other types of drives would be economically inefficient. In big hydraulic presses, the energy consumption is strongly affected by the mechanical structure design, as demonstrated in Strano et al. (2013). In forming presses with high tonnage, some energy savings can also be expected with better designed hydraulic systems and improved control. To this aim, Zhao et al. (2015) have developed an energy assessment model of large and medium sized hydraulic presses.

In presses and machines built for lower forming loads (e.g. tube bending machines), the efficient use of energy depends mainly on the type of drive mechanisms and on the control strategies. Servo-presses are generally considered less energy demanding than hydraulic machines, but no scientific literature is available with experimental evidence. However, even for hydraulic tube benders, relevant energy savings (up to 40%) can be achieved if developing specific control strategies, as demonstrated by Lin and Renn (2014). Industrially, there is a clear tendency in low tonnage metal bending applications for replacing the hydraulic units with electric modules. This transformation from hydraulic or hybrid to full-electric is mainly pushed by a common belief that fully electric machines are more precisely controlled. At the same time, the full-electric conversion is thought to reduce, as a positive side effect, the energy consumption. While this general opinion is likely correct, there is a lack of clear, quantitative and rigorous assessments (either numerical or based on experiments) of the actual differences, in terms of energy consumption, between the two solutions.

The goal of this research is to bridge the gap of the scientific community on this specific aspect. For this purpose, two NC rotary-draw bending machines were studied: one hybrid machine, equipped with hydraulic units in addition to some electro-mechanical drives; the other one only controlled by electro-mechanical drives. In order to make the energy assessment as much generalized as possible, a combined experimental-analytical approach was adopted. The experimental session was performed first. Power measurements were carried out on both types of machines during the process of bending a representative tube geometry. All phases of the process cycle were characterized registering the power absorptions of the machine and its main sub-modules. In order generalize the analysis, the two machines were experimentally characterized setting different machine velocities, i.e. different machine throughput. This allowed to describe if and how the throughput affects the power consumption.

An energy model suitable for both tube bending machines was proposed. The model parameters were identified exploiting the experimental measurements. The identified models were used for performing a general energy assessment of the machines. The performed analysis points out the achievable energy savings and their relevance in comparison with the energy required for forming the tube. For sake of generality, a sensitive analysis considering different test cases for the processed tube was also accomplished.

The paper is structured as follows. In Section 2 the methodology for performing the energy assessment and the studied rotary draw tube bending machines are thoroughly described. In Section 3 the tube bending model development is described. In Section 4 the main achieved results in terms of energy savings and machine efficiency (considering also the energy share of different meaningful test cases) are reported. Conclusions and future works are described in section 5.

2. Materials and methods

Two comparable tube-bending machines were critically analyzed from the energy consumption perspective. The energy savings potentials linked to the rotary draw tube bending machine electrification are mainly connected with replacing the hydraulic system (including the pump), that typically consumes energy even when the machine is not performing active phases of the bending process. Due to their high level of programmability, controllability, speed, accuracy and repeatability, electrical drives are a preferred drive technology when high production and quality performances is required. Moreover, these performances were easily assessed in the production environment by checking the quality of the bent tube and measuring the production cycle. On the contrary, since energy consumption can depend on the way the bending machine is used, a comprehensive and exhaustive energy assessment needs a more structured approach.

Before proceeding with a detailed description of the studied machines, the conceived methodology for performing the energy assessment was explained. One of the most important issues is the possibility to generalize the results of the energy analysis. For this purpose, it was decided to separate the energy expenditure of the working phase into two different components or shares: one which is tube dependent and one which is machine dependent. This approach was inherited by Diaz et al. 2009 that dealt with machine tools for metal cutting. The tube dependent share is made of the internal deformation energy of the tube itself (which obviously depends on the tube material and wall thickness). It can be easily estimated by means of empirical models or by means of relatively simple FEM (Finite Element Method) simulations, Kobayashi et al. 1989. The machine dependent components are the energy required for winning the inertial and frictional resistance of the machine itself and for keeping the clamps closed. The machine-dependent shares can hardly be predicted by a simple and reliable model. So far, only a specific research on hydraulic tube bending machines modelling has been found in the scientific literature, Lin and Renn (2014). The model was developed for evaluating innovative control strategies. Only numerical simulations are reported. Much attention was put in modelling the hydraulic cylinder, but the capabilities of predicting the energy absorbed by the whole machine was not experimentally verified. For this reason, in the present research, an empirical-analytical energy model for rotary draw bending machines was developed. This choice was also supported by a detailed literature review. Indeed, it was found that the empirical-analytical modelling methodology assures, at the same time, a high level of accuracy in terms of consumed energy estimation and the possibility of analyzing different scenarios of use for the considered machine, Balogun et al. 2013. Moreover, in the present paper this modelling approach is developed for the first time for a tube bending machine.

The model parameters were identified through a regression approach (Montgomery, 2001) carried out on power measurements. As will be adequately explained in section 3, the power model was developed as a function of the process rate (machine throughput). The experimental energy assessment was performed following the guidelines provided by the ISO 14955 (2014). Power measurements were performed under different working conditions. The following states were considered: “machine off”, “machine warm-up”, “ready for operations” and “working”. In addition, a characteristic combination of the four states was also defined, in order to model the “shift regime”. In this analysis, we assumed the “shift regime” as a typical 1-shift, 8-hours, working day, as defined in Tab. 1.

Tab. 1: Shift regime definition – working day – 1 shift

Time interval	hours
machine off	16
machine warm-up	0.5
working and ready for operations	7.5

During the active (working) phase, the tube is processed: the tooling and the tube are automatically positioned and the bends are executed. During the passive (ready for operations) phase, the tube is manually unloaded and a new tube is loaded onto the rotary-draw bending machine. In order to focus only on the machine dependent energy contribution, the experiments were conducted on both machines without the presence of the tube. Once developed, the model was used for comparing the energy share associated to the machine under different working conditions. In Section 4 the analysis is extended considering the energy share linked to the forming process itself. Achievable energy savings were compared to the energy share associated to the bending process and the machine efficiency increment due to the electrification process was evaluated. Different test cases that involve various materials and wall thicknesses were also considered to extend the results of the analysis. The described approach is schematized in Fig. 1.

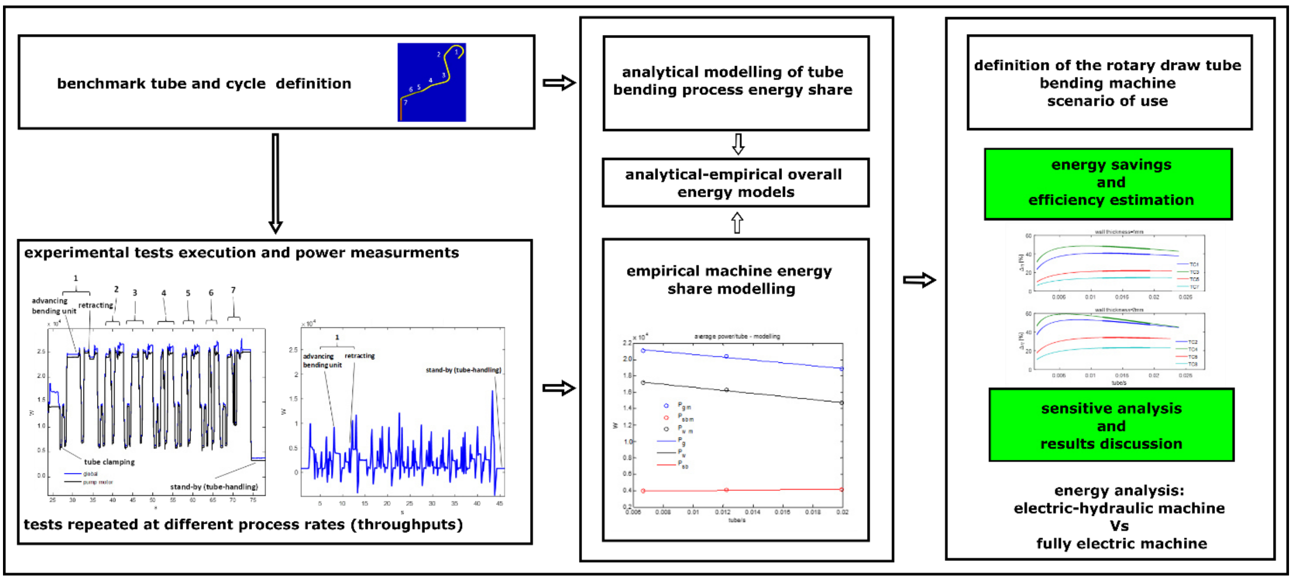

Fig. 1: Approach description

The hybrid electro-hydraulic rotary draw tube bending machine along with its main units and axes is shown in Fig.1. In this machine, only X, Z and W4 axes are driven by electric drives. On the contrary, in the fully electric machine, all the axes are piloted by servomotors. As far as the bending performance is concerned, the machines are very similar. Both are equipped with a tooling setup able to bend tubes with the following dimensions: 80 mm in terms of outer diameter (OD) and 1÷2 mm in terms of initial tube wall thickness (t_0). Different metal materials can also be processed, ranging from copper to aluminum alloys, from structural to stainless steels.

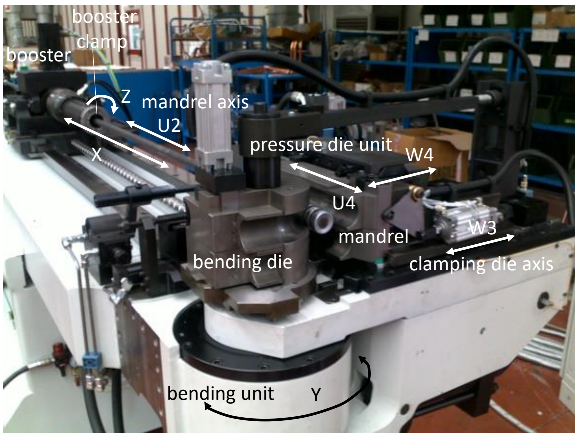

Fig. 2 Electro-hydraulic tube-bending machine with the main units and axes

The tube pictured in Fig. 3 was taken as a meaningful benchmark for all performed analysis. The tube has 7 bends along its length (numbered in Fig. 2), in different planes, with different bending angles α (from a minimum of α=10° for bend #5 to a maximum of α=180° for bend #1), Tab. 2. All curves are bent to a constant mean bending radius R_m=100 mm. The Degree of Bend (DoB) is an empirical measure that indicates the difficulty of the process in terms of risk of defects on the bent tube (thinning, wrinkling, fracture, etc.), as further detailed in Mentella and Strano (2012). The DoB is calculated as the ratio between the bending radius R_m and the outer diameter OD (DoB=R_m/OD). The DoB ratio is equal to 1.25 for the nominal analysed benchmark tube (the minimum possible value being 0.5). The tube wall factor WF, defined as the ratio between the outer diameter OD and the wall thickness, ranges from 40 to 80. The smaller the ratio DoB/WF, the easier the bending process: when this number exceeds 25, the bending process is very difficult; when this number approaches the value of 70, the bending process is virtually impossible. In this tests case, for a tube with 1.0 mm wall thickness, the DoB/WF is equal to 64; for a tube with 2 mm wall thickness, DoB/WF is equal to 32. In any case, the test case can be considered significant, since it represents a severe bending process.

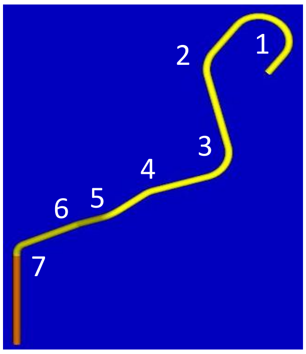

Fig. 3: Reference tube used for the experimental assessment

Tab. 2: Details of the bending operations performed on the benchmark tube

	α [deg]	Length of straight portion [mm]	Rotation between planes [deg]	Mean radius [mm]
1	180	100	30	100
2	60	150	-30	100
3	90	300	200	100
4	70	300	-60	100
5	10	150	-180	100
6	45	300	90	100
7	120	100	-90	100

Not only the process is severe in terms of wall factor WF and degree of bend DoB, but it is also demanding with respect to the kinematics of the machine. In fact, it needs all the most typical movements generally required by this kind of rotary draw bending machines and all the driving units are involved in this cycle. For each bend, a sequence of steps needs to be carried out: the booster (X axis, hydraulically actuated, shown in Fig. 1) feeds the tube to the die, the tube is clamped by the clamping die unit and the bending unit performs the bend with the support of the pressure die unit that moves both along W4 and U4. The bending unit is retracted as well as the pressure die unit. The tube is then rotated (Z) and axially fed along X in order to be prepare for the subsequent bend.

For each tube-bending machine, several absorbed power measurements were performed. For characterizing the power of the main energy demanding units, specific measurements were carried out focusing on some subsystems of the bending machine. This was done measuring directly the electric power absorbed by the analyzed subsystem or, if not possible, sequentially switching on the main machine modules and estimating it through differential power measurements.

The active power P(t) was calculated through Eq.(1). The three-phases voltages (v_1, v_2, v_3) and the corresponding currents (i_1, i_2, i_3) were acquired with a properly designed power meter at very high sampling frequency (f_s=30kHz). Specifically, each current ($i_j(t)$) was measured using a LEM ring that exploits the Hall's principle.

Such a high sampling rate was selected in order to be able to correctly measure the power absorbed by bending machine units piloted with drives that exploit the pulse-width modulation PWM, typically characterized by a switching frequency of 4-6 kHz.

$$P(t) = v_1(t) \cdot i_1(t) + v_2(t) \cdot i_2(t) + v_3(t) \cdot i_3(t) \tag{1}$$

The high frequency power P(t) was further processed. Indeed, a moving average elaboration was performed for filtering the high frequency noise that generally affects the measurements. A time interval $\overline{\Delta t} = 0.1s$ was used for the moving average processing. Fig. 4 and Fig. 5 show some of the global power measurements carried out during the tube bending sequence. Fig. 4 refers to the electro-hydraulic machine while Fig. 5 shows the power absorbed by the fully-electric machine. Different steps (7 bends) of the working cycle are easily noticeable in the power measurement profile associated to the hydraulic bender, Fig. 4. Moreover, for each single bend the advancing and the retracting of the bending unit can be seen. The smaller peaks are associated to repositioning movements in order to let the tube advance between bends.

For what concerns the hydraulic machine, both the global power and the power absorbed by the main pump motor were simultaneously acquired, Fig. 4. Even before executing a detailed analysis, it is worth noting that the energy absorbed by the pump of the motor in the whole cycle represents the most relevant share to the global energy. The fully electric draw-bending machine is characterized by a completely different power measurement profile that makes much harder discriminating the different phases of the analyzed process, Fig. 5. This is due to a shorter cycle time (45 s vs. 75 s) and to a smaller value of both average power and power peaks. In other words, the power peaks due to the bending and the retraction movements are comparable to the peaks due to the axial repositioning movements.

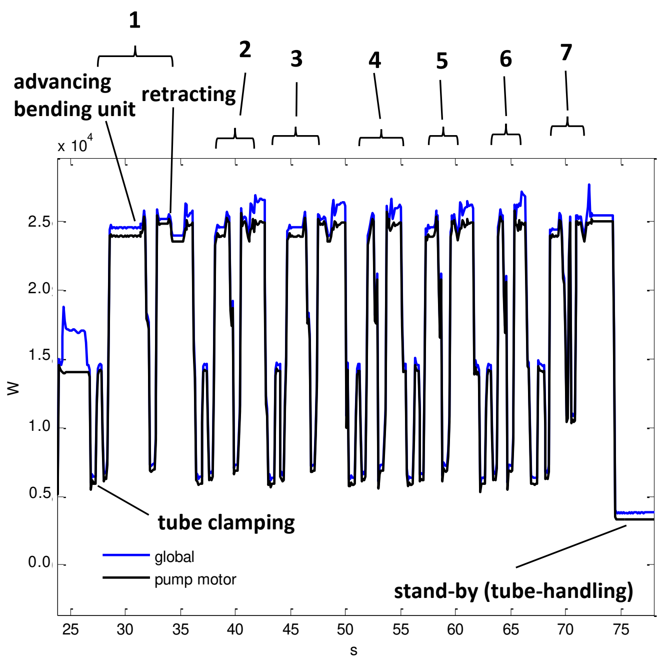

Fig. 4: Powers revealed on the electro-hydraulic machine – tube processing

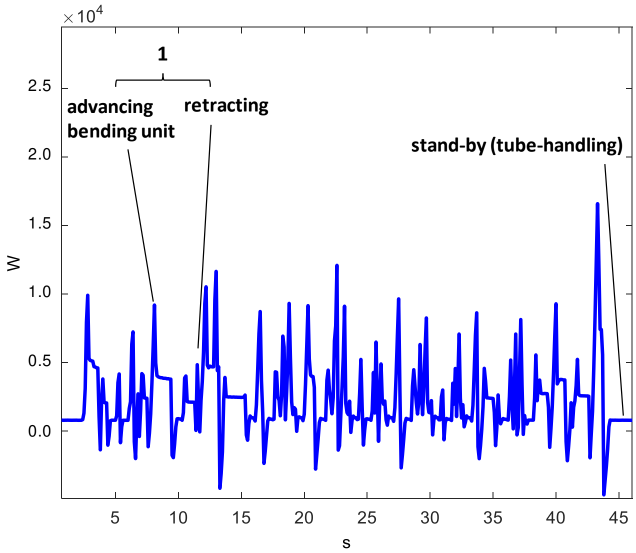

Fig. 5: Power measurements- fully electric machine – tube processing

The preliminary energy analysis was carried out considering the "ready for operations" state. Indeed, as reported in Gutowski et al. (2006) and in Santos et al. (2011) the machines can absorb considerable amount of energy due to the high stand-by power even if they are not processing the workpiece.

The experimentally measured stand-by power linked to the "ready for operation" mode was split into its main contributions and the most relevant machine functions were considered, Tab. 3. Just focusing on the stand-by power, it is worth of noting that the hydraulic draw bending machine is much more energy demanding compared to the fully electric version. Although the analysis of the "ready for operation" state gives a rough indication of the energy, it cannot be used for a comprehensive assessment because the power absorbed by the machine typically depends on the specific working conditions and on the processed part, Albertelli et al. (2016). For this reason, an energy model for draw bending machines was developed and used for performing a meaningful machine energy consumption comparison, section 4.

Tab. 3: Stand-by power for both machines

Electro-Hydraulic Bending Machine			Fully Electric Bending Machine		
component	stand-by power [kW]	function	component	stand-by power [kW]	function
PLC +24V units	0.26	base power supply +diagnostic	PLC +24V units +fan	0.363	base power supply +diagnostic
Drives	0.077	control	Drives	0.222	control
Pump Motor	3.89	main unit	Drives (IGBT)	0.215	drives power
SUM	4.22		SUM	0.8	

3. Energy modelling development

In order to generalize the energy assessment, it is necessary to extend the analysis to the shift regime that involves also the rotary draw-bending machine active motions. In this Section, the proposed model accounts only for the machine-dependent energy, hence separating it from the contribution of the tube deformation. Section 4.1 will include the analysis of the energy share required for forming the tube considering, for sake of generality, different reasonable scenarios (tube materials and thickness).

The model takes into account the machine power consumption as a function of the working conditions. This feature allows making the energy comparison more robust, general and exhaustive.

In order to reach this goal, a suitable model is developed starting from an energy model already used for metal cutting machine tools, Gutowski et al. (2006). This model is described by Eq. (2).

$$P(MRR) = P_0 + \mathrm{K} \cdot MRR \qquad (2)$$

This modelling approach was preferred to the one described in Zhao et al. (2015) that focused mainly on machine subcomponents losses modelling. The model approach considers that the absorbed power P depends linearly, through the coefficient K, on the process rate. The contribution to the power that is not related to the process rate is P_{i0}. In machining, the material removal rate MRR can be considered a meaningful indicator of the productivity. For instance, in machine tools, if MRR is increased the absorbed power increases too. This behavior can be mainly due to the increment of the rate for processing the material but also to other aspects (i.e. friction of the axes that increases with the feed velocity). In order to adapt the model to tube forming machines (rotary draw-bending machines), in this paper we assume as process rate quantity the tube-bending machine throughput TBMT (TBMT=1/Δt), where Δt is the time for processing each single tube.

$$P_i = f(TBMT) = P_{i0} + K_{i0} \cdot TBMT \tag{3}$$

P_i is the power absorbed by the machine as a function of its throughput TBMT. K_{i0} is the coefficient, measured in Joule, that describes this linear dependence. From the interpretation perspective, K_{i0} represents the energy difference for the production of each single tube, mapped on different machine functions (stand-by, working and for the whole machine), ascribable to a unit increment of the machine throughput TBMT.

Such a model was proposed for both the analyzed bending machines. The model parameters identification was done using the experimental power measurements performed while bending the reference tube in Fig. 3. For both machines the cycle was repeated at different throughput levels.

Since the selected model deals with the absorbed power, the average powers were computed from the executed experimental tests. In particular, the average global power $P_{g\,m}$, the average stand-by power $P_{sb\,m}$ and the average working power $P_{w\,m}$ can be determined as follows:

$$\begin{aligned}
&E_{cum} = \int_{t_1}^{t_2} P_{global}(t)dt;\\
&E_{\text{stand-by cum}} = \int_{t_1}^{t_2} P_{\text{stand-by}}(t)dt;\\
&E_{\text{working cum}} = E_{\text{movement}} + E_{\text{control}} =\\
&= \int_{t_1}^{t_2} \left(P_{global}(t) - P_{\text{stand-by}}(t)\right)dt;
\end{aligned} \tag{4}$$

E_{cum} is the cumulated energy linked to the global power $P_{global}(t)$, $E_{\text{stand-by}}$ is the cumulated energy associated to the stand-by power $P_{\text{stand-by}}(t)$ and E_{working} is the cumulated energy of the power used for the machine control and motion. The underlined area represents the energy required for processing each single tube, E_{tube}.

ΔT is the duration of the tube processing

$$\Delta T = t_2 - t_1 \tag{5}$$

Finally, the average power values are:

$$\begin{aligned}
&P_{g\,m} = E_{cum}/\Delta T;\ \mathrm{P}_{\text{sb m}} = E_{\text{stand-by cum}}/\Delta T;\\
&\mathrm{P}_{\text{w m}} = E_{\text{working cum}}/\Delta T
\end{aligned} \tag{6}$$

An example of the cumulated energy computation, in this case for the hydraulic machine, is graphically shown in Fig. 6. Once computed the average powers, Eq. (3) was used for fitting (LSR (Least Square Regression) approach) the experimental data where “i” stands for “global”, “sb” or “w”. In Fig. 7 both the experimental data and the fitting linear models are reported. It can be observed that the model fitting exhibits quite high coefficient of determinations ($R_g^2 = 0.988;\ R_{sb}^2 = 0.994; R_{sb}^2 = 0.94$ for the hydraulic machine). The identified parameters, for both the machines, are reported in Tab. 4.

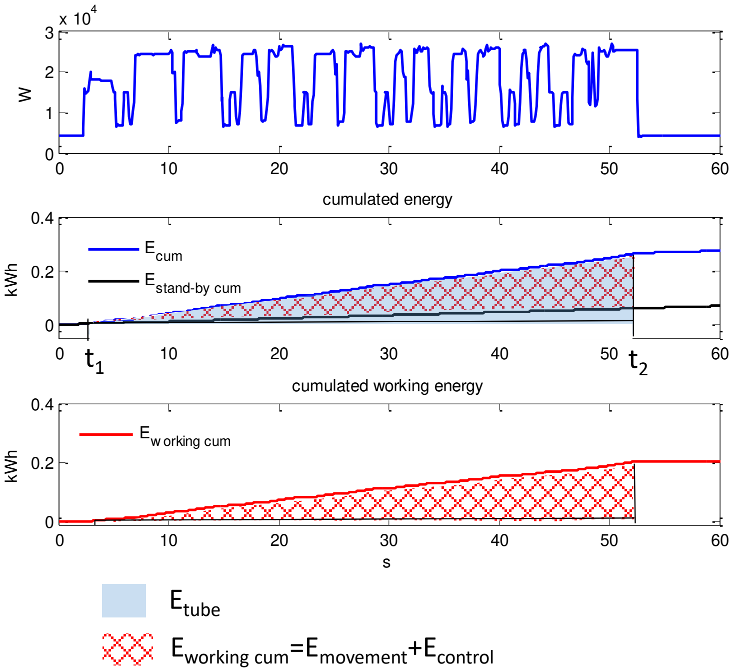

Fig. 6: Cumulated energy (stand-by and process energy)

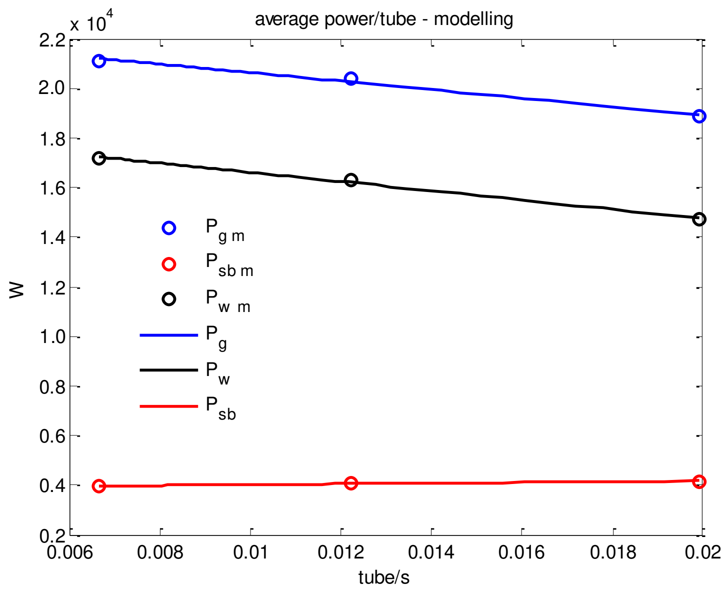

Fig. 7: Effect on tube processing time on average powers (hydraulic) – identified models

Tab. 4: Model parameters identification

Parameter [unit]	Hybrid hydraulic-electric machine	fully electric machine
P_{g0} [W]	22330	341.56
P_{sb0} [W]	3863.3	793.37
P_{w0} [W]	18467	-451.81
K_g [W/(tube/s)]	-170472	87245
K_{sb} [W/(tube/s)]	15504	-70.353
K_w [W/(tube/s)]	-185976	87315

Focusing first on the hydraulic machine, a negative K_{i0} can be observed for both the working average power and for the global average power. It means that increasing the throughput, the average power used for each single tube decreases. This peculiar behavior was not typically observed in metal cutting machine tools where the power increases with the process rate, Gutowski et al. (2006). In that case, the typical trade-off is observed when energy reduction is sought: the process rate increment makes the machine power bigger but the task is executed in shorter time. The overall effect on the energy depends on P_{i0} and K_{i0}.

For the hydraulic rotary draw tube bending machine, the advantages of the machine throughput maximization for what concern the energy saving are unequivocal.

It was also observed that the stand-by average power shows a positive slope. It means that by increasing the bending feed rate, a moderate increment of the stand-by power can be observed. This could be linked to the pump motor. Indeed, pump efficiency of the pump tends decreasing with the oil flow increment and, as a consequence of the losses increment, a higher power for cooling the oil could be requested.

The same modelling approach was also used for the fully electric machine. For such a bending machine, it can be noted that the coefficient K_g is positive. This means that average global power increases with the increment of the process rate (throughput). This is the typical behavior observed also in machine tools for metal cutting. It was moreover observed that the identified K_{sb} is negative but with a negligible value that means that the absorbed average stand-by power is practically insensitive to the throughput.

4. Results and discussion

The identified models can be used to perform an energy assessment considering the defined shift regime. The energy consumed for processing a single tube ($E_{1\text{-tube}}$) is used as the key performance indicator. The indicator computation was done dividing the global energy absorbed in a full shift (8 working hours) for the number of processed tubes N. The following relationships can be used:

$$T_{working} = 7.5h;\;\; T_{warm-up} = 0.5h;\;\; \Delta t_{\text{t-handling}} = 20s \tag{7}$$

where $T_{working}$ is the available time for the production, $T_{warm\text{-}up}$ is duration of the machine heating cycle and $\Delta t_{t\text{-handling}}$ is the time required for the tube handling (load-unload).

More specifically, $T_{working}$ can be expressed as a function of the number of processed tube.

$$T_{working} = T_{working\ \text{active}} + T_{handling} = \Delta T \cdot N + \Delta t_{t-handling} \cdot N \tag{8}$$

Thus, the energy computation E_{total} that refers to the shift can be described by the following equation:

$$\begin{aligned} E_{total} &= f(TBMT) = E_{tubes} + E_{t-handling} + E_{warm-up} = \\ &= N \cdot \Delta T \cdot P_g(TBMT) + T_{handling} \cdot P_{sb}(TBMT) + P_{warm-up} \cdot T_{warm-up} \end{aligned} \tag{9}$$

Where E_{tubes} is the energy used for processing the N tubes in the whole shift, $E_{t\text{-handling}}$ is the total energy used during the shift to load/unload the tube and $E_{warm\text{-}up}$ is the energy used for heating up the machine at the beginning of the shift.

The specific energy for processing a single tube $E_{1\text{-tube}}$ can be computed as:

$$E_{1-tube}(TBMT) = E_{total} / N \tag{10}$$

In Fig. 8 and Fig. 9 both the energy used for processing a single tube $E_{1\text{-tube}}$ and the global energy E_{total} as a function of the machine throughput were reported. All the energy shares that compose the total energy E_{total} are also reported.

It is worth of noting that in both the cases $E_{1\text{-tube}}$ is decreasing with the throughput increment while E_{total} shows a different behavior. If E_{total} is increasing with the throughput for the electric machine this is not true for the hydraulic machine.

This means that the hydraulic technology is not efficient at all. This is due to the need of powering the hydraulic unit even if the machine is not performing an active part of the cycle.

This behavior (in electric-hydraulic machine) should push the user, in order to minimize the energy consumption, to use the machine always at the maximum possible rate and keeping it off at the longest possible time, i.e. to anticipate the production. Unfortunately, this is not always possible because the maximum achievable feed rate is generally limited by the properties of the hydraulic components (i.e. pump and cylinder). Moreover, the process power requirements might hit the power limits of the machine. In fact, in these models the additional power required for the tube deformation is not yet included; for thick tubes with high strength, the tube deformation energy might be very large (as shown in the following sub-section 4.1).

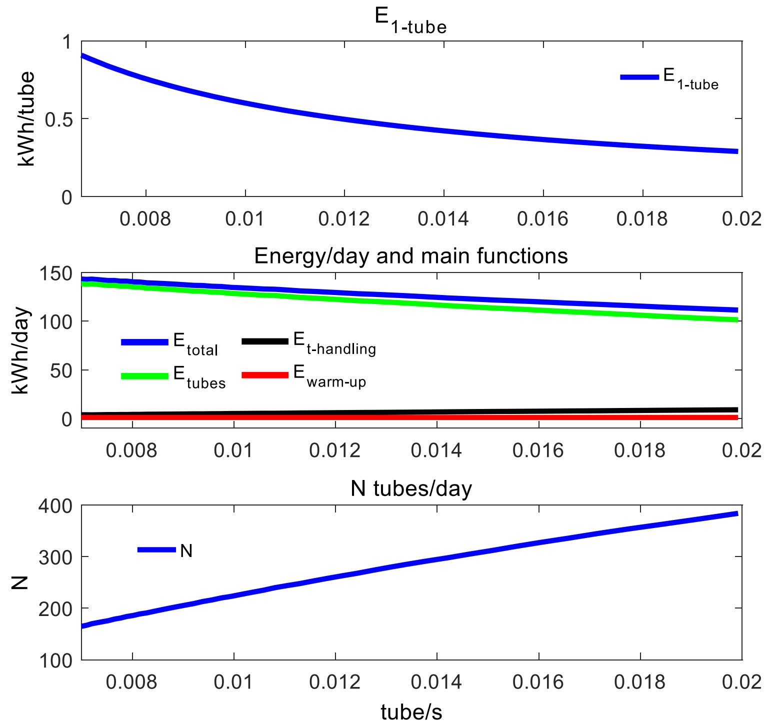

Fig. 8: Shift regime analysis – working day production – Electro-hydraulic

On the contrary, in the fully electric machine, not only the energy consumed per each shift is one order of magnitude less, but it is also less sensitive to the variation of throughput. Since the total energy (mildly) increases with the production rate, the user is more free to use the machine at a rate suggested by a “just in time” operating approach.

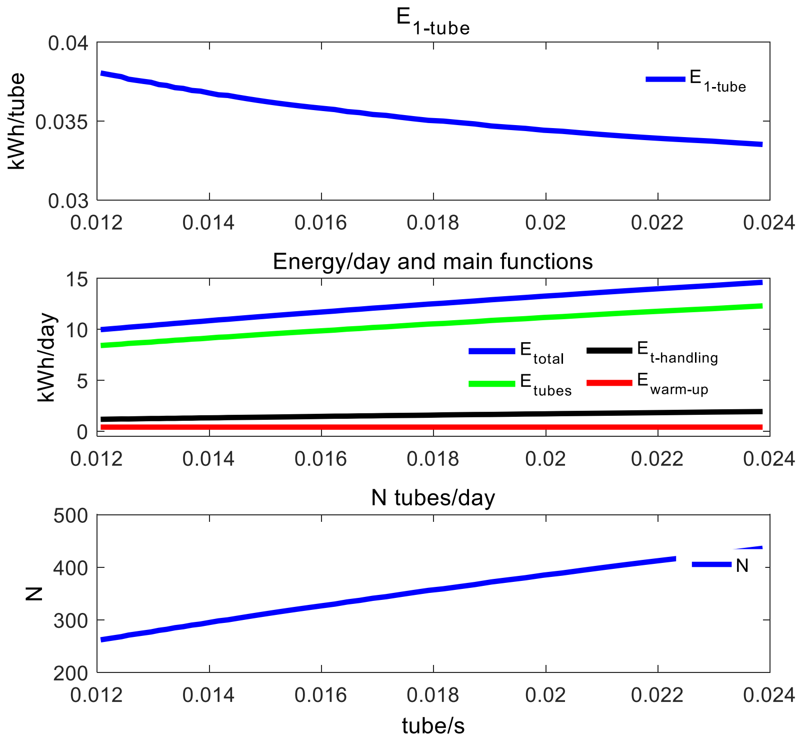

Fig. 9: Shift regime analysis – working day production – Fully Electric

4.1. Tube dependent energy share

In order to perform a more comprehensive evaluation of the energy savings connected to the electrification of the tube bending machine, the energy share due to the plastic deformation of tubes must be added. To give the analysis a general validity, it has been performed considering different realistic production test cases (TC). In fact, the deformation energy share depends on the tube material and thickness. Obviously, the deformation energy also depends on the tube outer diameter OD and on the mean bending radius R_m, but these two variables were determined by the tube reference geometry, Fig. 3. Typical materials used with the investigated machines are copper, steel, stainless steel, aluminium alloys of the 2000 or 6000 series. In Tab. 5, the tube-dependent energies required for bending several material/diameter combinations are reported. In the fourth column, the average bending moment required for each bend is given.

The bending moment M_b is calculated under the assumption of a fully plasticized cross section with exponential strain hardening (n), in plane strain, according to Eq. (11)

$$M_b = \gamma \cdot 4K \int_{\frac{OD}{2}-t_0}^{\frac{OD}{2}} \int_{0}^{\frac{\pi}{2}} \rho^2 \sin\alpha \cdot \left[\varepsilon_0 + \frac{2}{\sqrt{3}} \ln\left(1+\rho\frac{\sin\alpha}{R_m}\right)\right]^n d\alpha \cdot d\rho \tag{11}$$

where the integration variables ρ and α represent the radial position along the tube cross section and the angular position along the tube circumference respectively. K, ε_0 and n are the strain hardening parameters of the tube materials according to the Swift-Kuprkowsky hardening law, Daly et al. (2007). OD, t_0 and R_m are the tube outer diameter, initial thickness and mean bending radius; γ is an empirical parameter, greater than one, that depends on the specific selection of the rotary draw bending tooling setup and that must be determined experimentally.

In the fifth column, the tube-dependent energy required for each benchmark tube $E_{\text{forming 1-tube}}$ is reported. It can be easily calculated once the bending moment M_b and the bending angle are known for each bend of the sample part, Eq. 12.

$$E_{\text{forming 1-tube}} = \sum_{j=1}^{n_{curves}} M_b \cdot \phi_j \tag{12}$$

The total energy requirement for each benchmark tube ranges from 0.008 to 0.113 kWh, depending on the material and thickness combinations. This total value will contribute to the energy requirement, regardless of the production lot size and the type of machine used.

Tab. 5: Tube-dependent moment and energy shares

Test Case	tube material	Wall thickness	Bending moment	tube-dependent energy $E_{\text{forming 1-tube}}$ (advancing)
		[mm]	[N*m]	[kWh]
TC1	Docol 600	1	7347	0.034
TC2	Docol 600	2	14289	0.068
TC3	Aisi 304	1	11733	0.052
TC4	Aisi 304	2	22790	0.103
TC5	DHP copper annealed	1	2557	0.012
TC6	DHP copper annealed	2	4972	0.024
TC7	Al 6061 annealed	1	1373	0.007
TC8	Al 6061 annealed	2	2673	0.013

Fig. 10 reports the energy savings that can be accomplished ($\Delta E_{1\text{tube}}$) substituting the hydraulic technology with all electric drives, as a function of throughput. The expected savings are divided by the energy required for forming each single tube $E_{\text{forming 1 tube}}$. Energy savings are larger for lower production rates. The plotted ratio is always greater than 3, i.e. the energy savings allowed by the electric conversion are at least 3 times larger than the forming energy. In some of the analysed cases, the energy saving can be up to 65 times of the energy used for bending the tube. This happens for tubes with low tensile strength (copper and aluminium tubes), i.e. when the required bending moment is low.

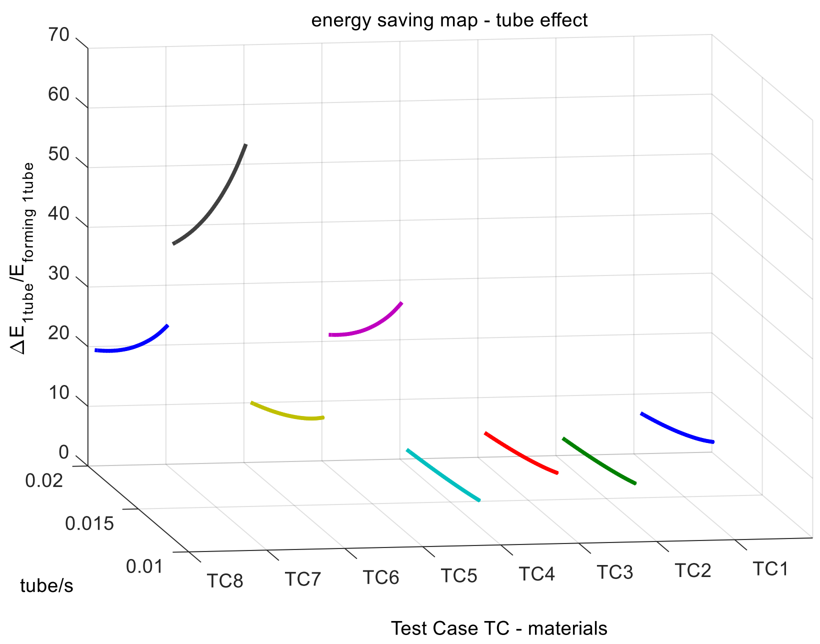

Fig. 10: Energy saving map

Most of the papers that deal with the machine energy assessment and modelling generally use the percentage ratio between the energy required for the process and the energy consumed by the machine (η) as an efficiency indicator, Albertelli et al. (2016).

This approach can be used also for the tube bending machines.

$$\eta = f(TBMT) = \frac{E_{\text{forming 1-tube}}}{E_{\text{forming 1-tube}} + E_{\text{total}}(TBMT)} \tag{13}$$

The efficiency of the electric machine $\eta_{electric}$ as a function of the test cases and the process throughput is shown in Fig. 11. The efficiency was plotted for a fictitiously extended production ratio range. For very low throughput the efficiency tends, as expected, toward zero. It can be observed that for the analysed machine throughput range, the efficiency of the electric machine appears to be constant (very low curve slope) with the production rate (thicker segments on curves in Fig. 11). On the contrary, the efficiency is significantly dependent on the considered test case. Although fully electric machines show a considerably higher efficiency in comparison to hydraulic machines, in some of the studied cases the efficiency is very small, i.e. the energy consumption of the deformation process is negligible with respect to the energy consumed by the machine.

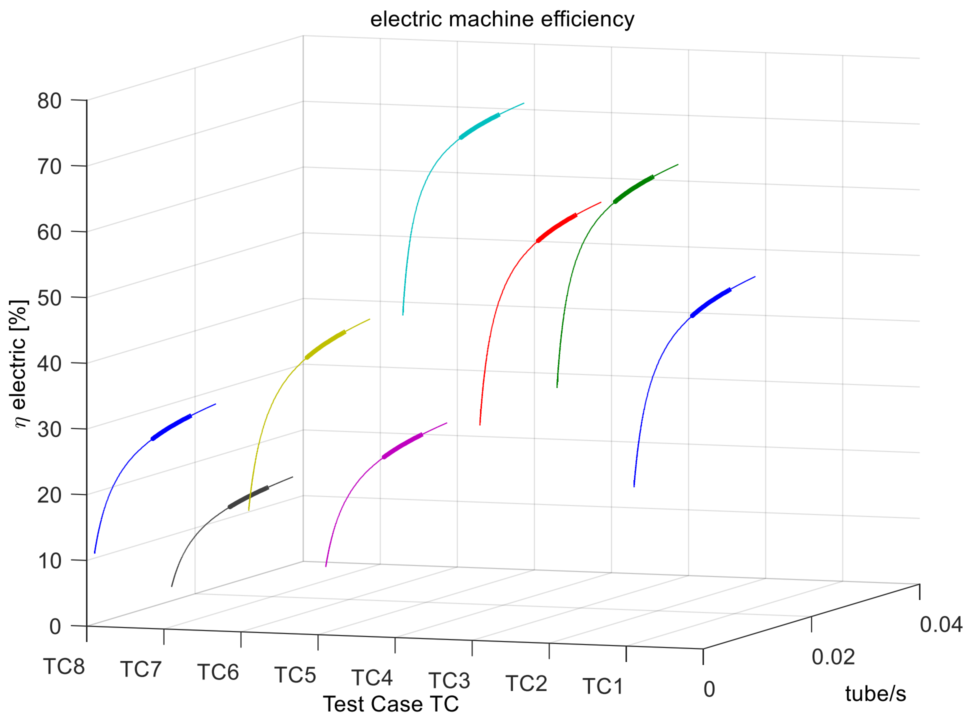

Fig. 11: Electric tube bender efficiency [%]

The same analysis (machine efficiency) was also performed for the hybrid electro-hydraulic tube bender. The advantage yielded by the full electric machine over the hybrid machine can be estimated using Eq. (14) which calculates the efficiency increment:

$$\Delta\eta = \eta_e - \eta_h \qquad (14)$$

The resulting efficiency increment map is reported in Fig. 12. Higher efficiency increments are observable for the test cases in which a thicker tube is processed.

It is worth of noting that in some of the analyzed cases (TC1, TC2, TC3, TC4) the efficiency increment decreases when the machine throughput increases. This happens for the steel tubes, when the hydraulic and electric machine exhibit a higher efficiency. Indeed, for these cases the electric machine has an efficiency that is close to be constant while the hydraulic machine increases significantly its efficiency with the increment of the throughput. For the other cases, i.e. the light alloys bending cases, the efficiency of the hydraulic machine is very low (less than 5%) and its increment as a function of the throughput is practically comparable with the one observed in the electric machine. This results in curves (Fig. 12) that are nearly flat.

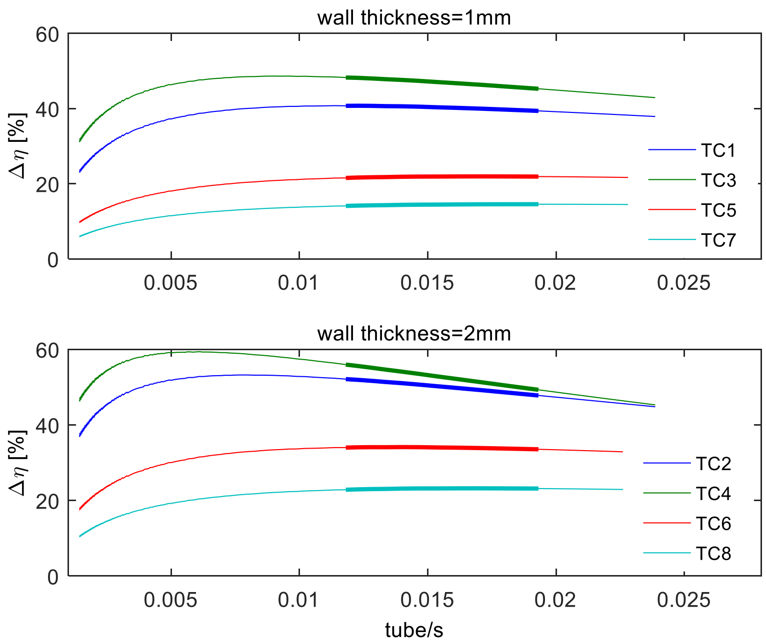

Fig. 12: Tube bending machine efficiency increment (η_e- η_h)

5. Conclusions

In this paper, a comprehensive study of the achievable energy consumption improvements granted by the adoption of all electric drives, in substitution of the traditional machines (which are hybrid hydraulic-electric), in the rotary-draw tube bending process was carried out.

The performed analysis was done using a combined experimental-analytical approach. An energy model for the machine dependent share was first developed exploiting experimental power measurements. The energy model can estimate the energy consumption in the whole machine working range. The model shows that while the total energy consumption of the hydraulic machine decreases with the throughput (production rate), the full electric machine consumption increases with the production rate. However, the machine energy share of the full electric unit is about 10 times smaller than the hybrid machine.

The analysis was extended considering the tube dependent energy share. This was done using a suitable analytical model. In order to generalize the results of the studies, various meaningful test cases were considered.

Finally, energy savings were evaluated as a function of both the specific processed tube and considering the machine throughput dependence. It was observed that even in the worst cases the minimum achievable saving are close to 3 times the energy required for forming the tube. This happens when the tubes with a higher tensile strength are bent.

Moreover, a comprehensive analysis of the efficiency of both machines was proposed.

Results revealed that considerable efficiency increments in the draw-bending machines can be accomplished. Better results can be achieved when high tensile strength materials are processed (Δη close to 50%). It was observed that in these cases the efficiency increments decrease with the throughput increment.

Future research efforts will be focused on the experimental validation of the estimated energy savings in a real production environment, incorporating cost modeling and overall environmental impact. Smaller operating costs and environmental impact are expected for hybrid/electric drives due to their shorter cycle times and simplified maintenance. Conversely, larger investment costs are generally required.

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