Biomass and Bioenergy 109 (2018) 166–171

Contents lists available at ScienceDirect

Biomass and Bioenergy

journal homepage: www.elsevier.com/locate/biombioe

Research paper

Effects of three cutting blade designs on energy consumption during mowing-conditioning of Miscanthus Giganteus

Hao Gan[a], Sunil Mathanker[a,b], Md Abdul Momin[a,b], Brendan Kuhns[a], Neal Stoffel[c], Alan Hansen[a,b], Tony Grift[a,b,*]

[a] Department of Agricultural and Biological Engineering, University of Illinois at Urbana-Champaign, 1304 West Pennsylvania Avenue, Urbana, IL 61801, USA
[b] Energy Biosciences Institute, University of Illinois at Urbana-Champaign, 1206 West Gregory Drive, Urbana, IL 61801-3838, USA
[c] Kondex Corporation, 1500 Technology Dr., Lomira, WI 53048, USA

ARTICLE INFO

Keywords:
Biomass
Bioenergy
Disc-cutter
Harvesting
Machine performance
Field capacity

ABSTRACT

This study compared energy consumption during harvest of Miscanthus Giganteus with a New Holland H8080 mower-conditioner among three cutting blade designs being 1) straight, 2) straight, angled at 30° and 3) serrated. Square bales were produced by a New Holland BB9080 large square baler. To calculate energy consumption per unit crop mass in MJ Mg^{-1}, bales of known mass were identified, and the cutting energy to produce this bale was calculated by accumulating the mower-conditioner's energy consumption across the collection area associated with that bale. Energy consumption was also expressed as a Percentage of Inherent Heating Value (PIHV), where energy consumption was divided by the heating value of Miscanthus Giganteus (17.7 GJ Mg^{-1}). Average energy requirement for the whole machine were 12.31 MJ Mg^{-1} (0.070 PIHV), 11.31 MJ Mg^{-1} (0.064 PIHV), and 9.27 MJ Mg^{-1} (0.052 PIHV) for straight, angled and serrated blades respectively. Average energy requirements for the header were 9.50 MJ Mg^{-1} (0.054 PIHV), 8.32 MJ Mg^{-1} (0.047 PIHV), and 7.20 MJ Mg^{-1} (0.041 PIHV) for straight, angled and serrated blades respectively. Average energy requirements for traction were 0.96 MJ Mg^{-1} (0.005 PIHV), 1.21 MJ Mg^{-1} (0.007 PIHV), and 1.04 MJ Mg^{-1} (0.006 PIHV) for straight, angled and serrated blades respectively. The theoretical field capacity increased from straight blades at 1.35 ha h^{-1} to angled blades at 1.52 ha h^{-1} to serrated blades at 2.23 ha h^{-1}. Evidently, the design of cutting blades had a significant effect on energy consumption and field performance of biomass harvesting equipment.

1. Introduction

Biomass has become the largest source of renewable energy in the United States, supplying more than 3% of the nation's total energy consumption. According to the United States Technical Advisory Committee, the goal is to increase this value to 5% by 2030. Accordingly, the U.S. Department of Agriculture (USDA) has planned to increase the use of agricultural crops as biomass for bioenergy [1]. *Miscanthus x giganteus*, hereafter referred to as “Miscanthus”, is a C4 grass which is slated as a source for the production of ethanol. Recently, the interest in Miscanthus in the U.S. has increased because of its high yield and low input demand [2]. Based on a projected average yield of Miscanthus of 30 Mg ha^{-1}, converting 9.3% of U.S. farmland to a Miscanthus crop would offset one-fifth of the current U.S. gasoline consumption [3].

As part of the effort to increase economic sustainability of Miscanthus, reducing harvesting costs can play an important role [4]. Currently, existing harvesters are used for Miscanthus, but they are operated at a relatively low speed because of the density and toughness of the Miscanthus stems compared to other grasses [5]. Therefore, it is imperative to re-evaluate the operation and design of harvesters to minimize the energy consumption as well as optimizing field performance. The most obvious focus is the cutting mechanism itself, which includes the blade design, cutting speed as well as adaptations to improve material throughput.

According to a standard published by the American Society of Agricultural & Biological Engineers, the power requirement of a counter shear cutting sickle-bar mechanism is 4.5 kW m^{-1} as opposed to 8.0 kW m^{-1} for an impact cutting rotary disc [6], and the cutting energy increases markedly when blades become dull [7]. In addition, the blade oblique angle was found inversely proportional to cutting energy requirement [8]. These results were confirmed in a laboratory study which indicated that a straight cut required an average of 8.4 MJ ha^{-1} as compared to 6.7 MJ ha^{-1} for a 30° oblique cut and 5.6 MJ ha^{-1} for a

* Corresponding author. Department of Agricultural and Biological Engineering, University of Illinois at Urbana-Champaign, 1304 West Pennsylvania Avenue, Urbana, IL 61801, USA.
E-mail address: grift@illinois.edu (T. Grift).

https://doi.org/10.1016/j.biombioe.2017.12.033
Received 29 March 2017; Received in revised form 29 December 2017; Accepted 31 December 2017

60° oblique cut [9]. Another field study was conducted in which a single disc-cutter was used to test the effect of blades with oblique angles of 0°, 30° and 40° angles on a mower-conditioner [10]. The results showed that the highest energy consumption of 46.1 MJ ha^{-1} occurred when using blades of 0° oblique angle and the lowest energy consumption of 9.1 MJ ha^{-1} occurred when using blades of 40° oblique angle. In a later field study where blades with 0°, 20° and 30° oblique angles were used on all discs on a mower-conditioner, the average machine energy consumed was 18.5 MJ Mg^{-1}, 15.9 MJ Mg^{-1}, and 13.5 MJ Mg^{-1} respectively [11].

In this study, a straight blade, an angled blade as well as a serrated blade were evaluated in terms of their effect on energy consumption on a mower-conditioner that was adapted for harvesting Miscanthus Giganteus biomass. The objective of this study was to compare energy consumption for the whole machine, and separately for the cutting head and for traction as well as determination of the theoretical field capacity among the three types of cutting blade.

2. Materials and methods

The equipment used to harvest Miscanthus comprised a 168 kW New Holland H8080 hydrostatic drive mower fitted with a 750 HD specialty rotary disc head (Fig. 1) and a New Holland BB9080 large square baler powered by a John Deere 7930 tractor (Fig. 2). Both machines were fitted with RTK-GPS position monitoring and a Controller Area Network (CAN) bus. To isolate the header's energy consumption from the machine energy consumption, the mower was fitted with pressure transducers in the feed and return hydraulic hoses of one of two hydraulic motors that drove the header (Honeywell, Model Z), with a range of 68.95 MPa (10,000 PSI). The same type of pressure transducer was used to measure the pressure differential across one of two hydraulic drive motors, allowing calculation of instantaneous traction power.

The three cutting blade types that were tested on the mower-conditioner are shown in Fig. 3 with the straight blade with 0° angle on the left, the 30° angled blade in the center, and the serrated blade on the right. All blades had an overall length of 110 mm, an overall width of 50 mm and a thickness of 4 mm.

The mowing header had a width of 4.7 m featuring 12 discs, spinning at a maximum rotational speed of 3000 RPM. Each disc had two slots for mounting blades, located 180° relative to each other, for a total of 24 blades. To accommodate cutting Miscanthus, some modifications were made to the header, including a smaller auger to convey materials, a crop lifter to remove material from the cutting disc, and fingers at the center of the auger to propel the crop into steel conditioning rolls that replaced the original rubber rolls.

Fig. 1. New Holland H8080 self-propelled mower-conditioner with 750 HD specialty rotary disc head.

Fig. 2. New Holland BB9080 large square baler powered by a John Deere 7930 tractor.

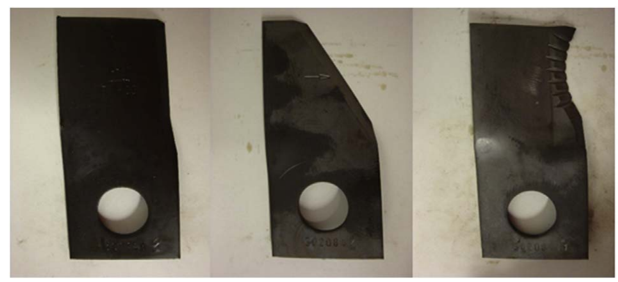

Fig. 3. Three blade types were compared in this study, from left to right: 0° straight blade, 30° angled blade, and a serrated blade. All blades had an overall length of 110 mm, overall width of 50 mm and a thickness of 4 mm.

According to the manufacturer's specifications, the New Holland BB9080 large square baler required a minimum nominal tractor PTO power of 91 kW, and produced bales with a width of 1.2 m, a height 0.9 m and a maximum length of 2.6 m. It was equipped with a yield monitoring system which provided the mass and volume of collected material, as well its real time RTK-GPS location and time stamp at 0.33 Hz.

The mower-conditioner was fitted with a data acquisition system comprising a HBM Somat eDAQ data logger which collected CAN bus data, as well as an HBM EGPS-200 PLUS unit which collected RTK-GPS data and provided an interface to the hydraulic pressure transducers. Data were recorded at a rate of 1 Hz including engine rotational speed, engine torque, hydraulic pressure in the feed and return lines of the hydraulic motor on the cutting head and drive motors, latitude and longitude from the RTK-GPS, true ground speed and fuel consumption.

2.1. Field experiments

In this study, a 3.64 ha field was split into three approximately equal-sized subplots, in which the three cutting blades were tested (Fig. 4). The south-west corner of the field was located at lat, lon: 40.062, −88.19953. The Miscanthus crop was harvested in March, 2014, as the moisture content reached a level of approximately 15%. During the tests, the mower's speed was determined empirically by a single operator, whose aim was to drive the machine as fast as possible while cutting the crop completely and preventing material clogging. Fig. 4 shows the passes of the baler in the field, where the numbers indicate the bale drop order. Notice that the baler did not always follows straight paths because the operator sometimes swerved to pick up material in low-yielding areas such as the sixth pass from the top of the map. It also swerved around an antenna tower located in the center of the plot.

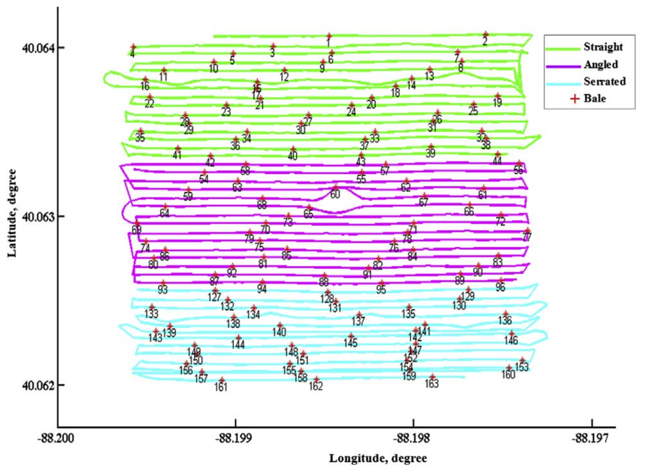

Fig. 4. A 3.64 ha plot was divided into three parts of approximately equal size to compare energy consumption of straight, angled and serrated cutting blades.

2.2. Calculation of energy consumption

Data from the eDAQ system as well as from the yield monitor were imported into MATLAB® R2013b for processing and analysis. To calculate the energy consumption of the mower-conditioner, its header and hydrostatic transmission, data segments were selected starting just after dropping a bale, and ending at the dropping of the subsequent bale. Only uninterrupted straight segments were used where the machine did not turn, back up or swerve around obstacles, as shown in Fig. 5. The mass of the bale dropped at the end of the segment was then used to calculate the accumulated energy input for that bale in MJ Mg^{-1}.

To calculate the energy consumption in each segmented area, the bale mass, collection area, machine energy consumption, header and traction energy consumption, and theoretical field capacity were calculated as follows:

1. The mass of the bale was calculated by accumulating instantaneous mass flow data from the baler's yield monitoring system, which provided RTK-GPS referenced mass flow information every 3 s.
2. To calculate the collection areas for each bale segment, the GPS data of the segment's start and end points were used to calculate the length of the segment using conversion equations [12], after which the length was multiplied by the header width of 4.7 m. To relate the segment data from the baler to the mower-conditioner measurements, the chosen start and end locations of the segment were matched with the closest locations in the mower-conditioner data file. For instance, bale No.4 (see Fig. 5) was formed between baler GPS locations A (lon_4^{start}) and B (lat_4^{end}, lon_4^{end}). The mower GPS locations closest to A and B were found as locations C and D. Subsequently, the mowing data between location C and D were extracted and assigned to bale No.4. This procedure was repeated for all selected segments and associated bales.
3. The CAN Electronic Engine Control 1 message (ID 217056256) contains both the Engine Speed RPM, and the Percent Torque parameters. Engine Speed RPM was converted to the rotational velocity in rad s^{-1}, and the instantaneous engine torque in Nm, was calculated from the Percent Torque value multiplied by the engine's rated torque of 729.2 Nm. Instantaneous engine power was calculated as the product of the latter two parameters. To calculate machine energy consumption, instantaneous engine power was integrated over time. A filter was applied that removed data points where the vehicle speed was close to zero, as well as data points which' values were out of range.
4. The hydraulic power to the header of the machine was calculated by the measured pressure differential across the feed and return lines of one of two hydraulic motors, multiplied by the flow rate through both motors combined. The flowrate from the New Holland H8080 series specifications sheet indicates that the rated pump flow to the header is 302.8 L/min (80 GPM), at a rated engine speed of 2200 RPM (the CAN data showed that during the tests, the engine ran consistently at rated engine speed).

The hydrostatic transmission of the H8080 machine consist of two variable displacement pumps driven in tandem with a total maximum flow output of 310.4 L/min (82 GPM) at a rated engine speed of 2300 RPM. This flowrate propels the machine at a maximum in-field speed of 20 km h^{-1}. Since the engine during experiments ran at 2200 RPM, the hydrostatic transmission pump's output flow during experiments was calculated as 2200/2300*310.4 = 296.9 L/min (78.4 GPM). Furthermore, since the flowrate through a motor is proportional to the rotational speed of the drive motor, the flowrate was multiplied by the actual machine speed divided by the maximum in-field speed. Header and traction energy were then calculated by integrating instantaneous hydraulic power over time, in the same manner as for machine energy consumption. Note that in the pump flow calculations the volumetric efficiency of the pumps was assumed 95% per recommendation of CNH Industrial (personal communication).

Energy consumption in MJ Mg^{-1} was calculated by dividing the accumulated energy per segment by the associated bale mass, and in MJ ha^{-1} by dividing the accumulated energy per segment by the associated collection area. The energy consumption in Percentage of Inherent Heating Value (PIHV) was calculated by dividing the energy consumption in MJ Mg^{-1} by the heating value of Miscanthus Giganteus, which is 17.7 GJ Mg^{-1}.

5. Theoretical field capacity was calculated by multiplication of the machine's average true ground speed (from RTK-GPS) in a segment leading to a bale by the header width of 4.7 m.

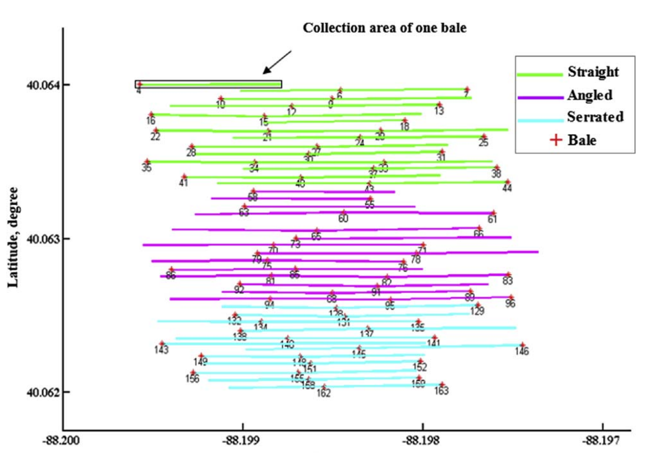

Fig. 5. Straight segments of the baler path were used to calculate energy input to the mower-conditioner among three cutting blade types.

3. Results and discussion

The mower-conditioner's machine energy consumption as a function of yield for the three blade types is shown in Fig. 6. Although the measurements exhibit ample variability, it is clear that serrated blades consumed less energy than the 30° angled blades, which, in turn, consumed less energy than 0° straight blades. A comparison between the 0° straight and 30° angled blade shows that an energy reduction of 8.2% is possible, corroborating findings in previous research [8], [9], [10], [11]. The straight line fit curves in Fig. 6 indicate a weak inversely proportional relationship between the mowing energy in MJ Mg^{-1} and the yield in Mg ha^{-1}.

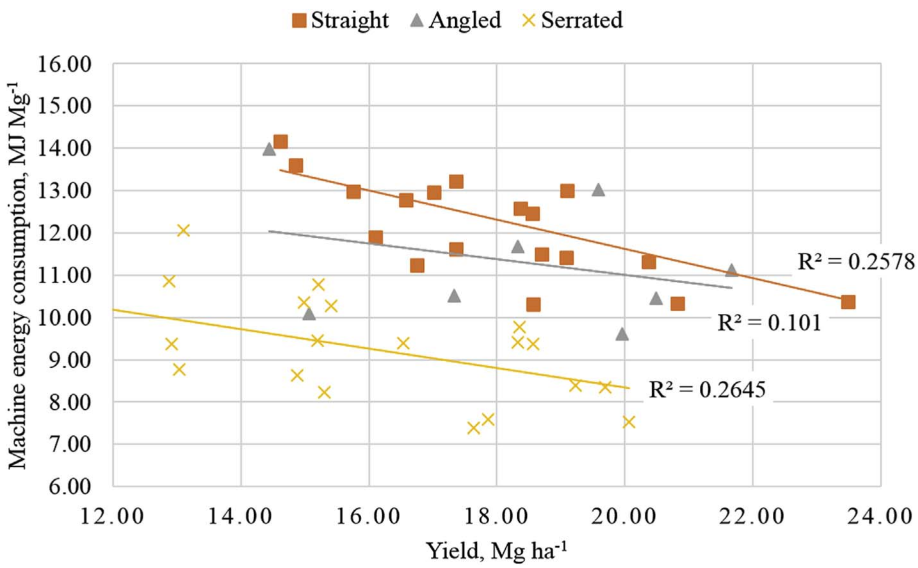

Fig. 6. Machine energy consumption versus yield during mowing-conditioning of Miscanthus for three cutting blade designs.

Table 1
Student t-test comparisons among straight, angled and serrated blades in terms of a) machine energy consumption, b) header energy consumption, and c) traction energy consumption. Note that non-significant differences are denoted in bold.

a) Machine energy, MJ Mg^{-1}

	Straight	Angled	Serrated
Straight			
Angled	**0.1190**		
Serrated	2.2929E-08	0.0009	

b) Header energy, MJ Mg^{-1}

	Straight	Angled	Serrated
Straight			
Angled	**0.0927**		
Serrated	1.0001E-05	0.0031	

c) Traction energy, MJ Mg^{-1}

	Straight	Angled	Serrated
Straight			
Angled	**0.4139**		
Serrated	**0.4960**	**0.1950**	

Table 1 shows a 2-tailed *t*-test to detect differences in machine, header and traction energy consumption among the three blade types at a 95% confidence level. In the case of machine energy consumption, there were significant differences between the serrated and angled blade ($p = .0009$), as well as between the straight and serrated blade ($p = 2.2929E\text{-}08$). However, the difference between the angled and the straight blade was not significant ($p = .1190$). Similarly, in the case of header energy consumption, there were significant differences between the serrated and angled blade ($p = .0031$), as well as between the straight and serrated blade ($p = 1.0001E\text{-}05$). As was the case for machine energy consumption, the difference between the angled and the straight blade was not significant ($p = .0927$). In the case of traction energy consumption no significant differences were found among the blade types (all $p > .05$).

Energy consumption of the mowing machine, its header as well as traction are shown in Fig. 7. The machine energy was measured as average (standard deviation) being 12.31 (1.47) MJ Mg^{-1} (0.070 PIHV) for straight blades, 11.31 (1.51) MJ Mg^{-1} (0.064 PIHV) for angled blades and as 9.27 (1.21) MJ Mg^{-1} (0.053 PIHV) for serrated blades. Average energy consumption of the header was 9.50 (1.82) MJ Mg^{-1} (0.054 PIHV), 8.32 (0.78) MJ Mg^{-1} (0.047 PIHV), 7.20 (0.83) MJ Mg^{-1} (0.041 PIHV), for straight, angled and serrated blades respectively, whereas the average energy consumption for traction was 0.96 (0.19) MJ Mg^{-1} (0.005 PIHV), 1.21 (0.17) MJ Mg^{-1} (0.007 PIHV), and 1.04 (0.22) MJ Mg^{-1} (0.006 PIHV), for straight, angled and serrated blades respectively.

Since the mower engine output power was measured from the CAN bus, and since the header and traction power were calculated individually from hydraulic pressure differentials across and flow rates through motors, the average overall efficiency of the hydraulic drive system was also calculated (see Fig. 7) as (9.50 + 0.96)/12.31 = 85.0%, (8.32 + 1.21)/11.31 = 84.3% and (7.2 + 1.04)/9.27 = 88.9% for straight, angled and serrated blades respectively. These numbers should be interpreted with caution, since they are affected by the temperature-related viscosity of the hydraulic fluid, which was unmonitored.

Table 2 shows the average energy consumption data for the machine, header and traction, in MJ h^{-1}, as well as average power in kW. In addition, the average machine speed and the calculated Theoretical Field Capacity values are shown as: 1.35 ha h^{-1} for the straight blade, 1.52 ha h^{-1} for the angled blade, and 2.23 ha h^{-1} for the serrated blade. In other words, the theoretical field capacity using serrated blades was 1.65 times higher than the theoretical field capacity using straight blades and 1.46 times higher than the theoretical field capacity using angled blades. This shows that serrated blades can yield a significant increase in theoretical field capacity because the operator can maintain a higher machine speed while maintaining a good quality of cut.

The observed differences in energy consumption of the header are obviously caused by different cutting processes among the three blades. The specifications of the 750 HD specialty head of the H8080 mower state that the tip speed of the blade reaches 301 km h^{-1} (83.6 m s^{-1}). Unfortunately, scant literature is available that explains the underlying physical processes at these cutting speeds. Earlier work used a universal testing machine to cut stems of Miscanthus at very low speeds [13], where the authors recognized three cutting processes being compression, local failure in tension and shearing. A comparison between straight and serrated blades showed that the serrated blade compressed the stem less than a straight blade. The serrated blade also broke the cortex into many small parts and caused easier failure due to cross-sectional tension. Finally, the stem was cut in the process of shear for both blades. The serrated blade needed less force in the first two processes which causes less energy consumption. When comparing the

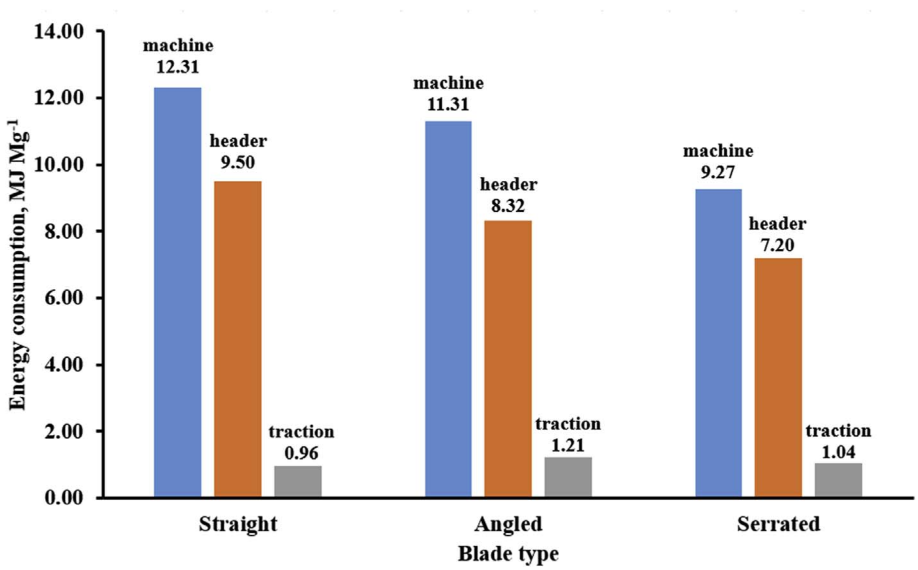

Fig. 7. Comparison of machine energy, header energy, and traction energy among three types of cutting blade.

Table 2
Average energy and power values among three blade types for a) machine energy consumption, b) header energy consumption, and c) traction energy consumption. Standard deviations are shown in parentheses.

a)

	Machine energy		Machine power		Machine speed	TFC
	$MJ\ Mg^{-1}$	PIHV	$MJ\ h^{-1}$	kW	$km\ h^{-1}$	$ha\ h^{-1}$
Straight	12.31 (1.47)	0.070	293.8 (11.84)	81.6 (3.29)	2.87 (0.33)	1.35 (0.16)
Angled	11.31 (1.51)	0.064	307.2 (10.08)	85.3 (2.80)	3.24 (0.58)	1.52 (0.27)
Serrated	9.27 (1.27)	0.052	316.9 (10.84)	88.0 (3.01)	4.74 (0.92)	2.23 (0.43)

b)

	Header energy		Header power	
	$MJ\ Mg^{-1}$	PIHV	$MJ\ h^{-1}$	kW
Straight	9.50 (1.82)	0.054	227.6 (38.35)	63.2 (10.65)
Angled	8.32 (0.78)	0.047	227.6 (19.75)	63.2 (5.49)
Serrated	7.20 (0.83)	0.041	247.2 (20.01)	68.7 (5.56)

c)

	Traction energy		Traction power	
	$MJ\ Mg^{-1}$	PIHV	$MJ\ h^{-1}$	kW
Straight	0.96 (0.19)	0.005	22.93 (4.04)	6.4 (1.12)
Angled	1.21 (0.17)	0.007	33.01 (3.39)	9.2 (0.94)
Serrated	1.04 (0.22)	0.006	35.85 (7.17)	10.0 (1.99)

straight and angled blades, the main difference was the relative movement between the blades' edges and the Miscanthus stem; a straight blade only applies a force in the direction perpendicular to the stem, whereas the angled blade applies perpendicular and tangential forces to the stem, yielding a slicing action. This is why the straight blade mainly shatters the stem upon impact, whereas the angled blade cuts it. This may explain why the average energy consumption using the angled blade was lower than that of the straight blade. While there was not a clear explanation for the difference between angled and serrated blades, a reasonable conjecture is that the serrations cause points of very high local pressure onto the stem, which initiate the failure of the stem while the blade itself cuts it through completely.

As shown in Table 2, the differences in traction energy among all cutting blades were not significant at the 95% confidence level. This shows that the cutting process is virtually independent of the traction needed to move the machine forward at a constant speed. The slight increase in traction energy for the angled blade type could be explained by the fact that the angled blade did not produce a clean cut, as observed during the experiments in the field. This meant that ample partially cut stems remained the field, causing a drag force onto the header.

4. Conclusions

This study comprised a field comparison of the effect of 1) a 0° straight cutting blade, 2) a 30° angled cutting blade and 3) a serrated cutting blade on the energy consumption and theoretical field capacity of a mower-conditioner machine while harvesting Miscanthus Giganteus.

The results showed that while mowing Miscanthus using serrated blades, the mower-conditioner saved 18.0% of machine energy compared to angled blades and 24.7% compared to straight blades. However, to put these energy savings into perspective, the machine energy consumption for all blades is very low relative to the inherent heating value of Miscanthus, being 0.070 PIHV (Percentage of Inherent Heating Value) for straight blades, 0.064 PIHV for 30° angled blades and 0.052 PIHV for serrated blades.

Weak inversely proportional relationships were found between machine energy consumption in $Mg\ ha^{-1}$ and yield in $Mg\ ha^{-1}$ for straight, angled and serrated blades. For machine energy consumption, significant differences were found for straight versus serrated blades and angled versus serrated blades, but not for straight versus angled blades. The same results were found for header energy consumption. For traction energy consumption, no significant differences were found among the three blade types.

Arguably, the most useful gain from the use of angled or serrated blades may lie in the fact that the operator can maintain a higher machine speed, yielding a significant increase in theoretical field capacity; the serrated blade's theoretical field capacity was 1.46 times higher than that of the angled blade and 1.65 times higher than that of the straight blade.

Although these are encouraging numbers, it must be noted that these results were obtained in tests in a single crop, and that no data were collected regarding the durability of the blades. In addition, no attempt was made to determine the optimal cutting speed of the angled blades. Future studies should address these factors.

Acknowledgements

This project was funded by the Energy Biosciences Institute through a program entitled Engineering Solutions for Biomass Feedstock Production. The authors are grateful to Kondex Corporation for providing the cutting blades and to Tim Mies and Collin Reeser of the EBI Energy Farm for managing and conducting the harvest operations. Thanks also to Ed Roy from CNH Industrial for providing detailed design parameters of the harvester.

References

[1] R.D. Perlack, Biomass as Feedstock for a Bioenergy and Bioproducts Industry: the Technical Feasibility of a Billion-ton Annual Supply, Oak Ridge Natl. Lab, TN, 2005.

[2] U. Jorgensen, Benefits versus risks of growing biofuel crops: the case of Miscanthus, Curr. Opin. Environ. Sustain. 3 (2011) 24–30.

[3] E.A. Heaton, F.G. Dohleman, S.P. Long, Meeting US biofuel goals with less land: the potential of Miscanthus, Global Change Biol. 14 (2008) 2000–2014, http://dx.doi.org/10.1111/j.1365-2486.2008.01662.x.

[4] E.A. Heaton, F.G. Dohleman, A.F. Miguez, J.A. Juvik, V. Lozovaya, J. Widholm, O.A. Zabotina, G.F. McIsaac, M.B. David, T.B. Voigt, N.N. Boersma, S.P. Long, Miscanthus. A promising biomass crop, Adv. Bot. Res. 56 (2010) 76–137, http://dx.doi.org/10.1016/B978-0-12-381518-7.00003-0.

[5] E. Anderson, R. Arundale, M. Maughan, A. Oladeinde, A. Wycislo, T. Voigt, Growth and agronomy of *Miscanthus* x *giganteus* for biomass production, Biofuels 2 (2011) 71–87, http://dx.doi.org/10.4155/bfs.10.80.

[6] ASABE, Agricultural machinery management data, Am. Soc. Agric. Biol. Eng. 2011 (2011) 1–14.

[7] C.R. Tuck, M.J. O'Dogherty, D.E. Baker, G.E. Gale, Laboratory studies of the performance characteristics of mowing mechanisms, J. Agric. Eng. Res. 50 (1991) 61–80.

[8] S. Persson, Mechanics of Cutting Plant Material, American Society of Agricultural Engineers, St. Joseph, Michigan, USA, 1987.

[9] P.C. Johnson, C.L. Clementson, S.K. Mathanker, T.E. Grift, A.C. Hansen, Cutting energy characteristics of Miscanthus x giganteus stems with varying oblique angle and cutting speed, Biosyst. Eng. 112 (2012) 42–48, http://dx.doi.org/10.1016/j.biosystemseng.2012.02.003.

[10] J.D. Maughan, S.K. Mathanker, B.M. Fehrenbacher, A.C. Hansen, Impact of cutting speed and blade configuration on energy requirement for Miscanthus harvesting, Appl. Eng. Agric. 30 (2014) 137–142.

[11] J.D. Maughan, S.K. Mathanker, T.E. Grift, A.C. Hansen, K.C. Ting, Impact of blade angle on miscanthus harvesting energy requirement, Trans. ASABE 57 (2014), http://dx.doi.org/10.13031/trans.57.10373.

[12] V.I. Adamchuk, {EC}01-157 Precision Agriculture: Untangling the {GPS} Data String, Hist. Mater, Univ. Nebraska- Lincoln Ext, 2001.

[13] Q. Liu, S. Mathanker, Q. Zhang, A.C. Hansen, Biomechanical properties of miscanthus stems, Trans. ASABE (2012) 1125–1131.