Sustainability analysis of various air-assisted orchard sprayer fan designs: Performance, energy and carbon footprint

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Abstract: The fan unit design of air-assisted orchard sprayers directly affects the amount of deposits on leaf surfaces, the amount of airborne drift and endodrift, as well as the fuel consumption during the operation. Therefore, in addition to reducing fuel consumption by modifying the fan inlet side, re-designing fan blade type, deflector type, and discharge side design of the turbofan unit has been a recent increasing trend in R&D studies to achieve higher airflow and better air-jet uniformity. In this study, to achieve better air penetration into the tree canopy, design and performance comparisons were made of three different turbofan designs, one of which is currently available on the market and popular in sprayers, while the other two are new designs. In accordance with EN ISO 9898, wind tunnel tests were carried out at a PTO rotation speed of 540 r/min with fan blade angles of 15º, 30º, and 45º and fan ratios of 1:3.5 (1890 r/min) and 1:4.5 (2430 r/min) for each turbofan. Simultaneously, air velocity measurements were made from the fan inlet side of the wind tunnel and from the discharge side of the fan. Furthermore, torque (N·m), speed measurement (r/min), and instantaneous fuel consumption (L/h) depending on the operation were also measured. The success of the fan blade type was determined by fan efficiency, and specific fuel consumption was an additional indicator of operating conditions. In experiments, 77.5% fan efficiency and specific fuel consumption 0.44 kg/kW·h was achieved with NVS fan. Regardless of fan type, several operational conditions available to users in fan design were statistically proven to be ineffective in terms of airflow.

Keywords: air-assisted orchard sprayer, turbofan performance evaluation, energy usage in agriculture, carbon footprint

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1 Introduction

The standard air-assisted orchard sprayers are widely preferred in horticulture worldwide for their ability to spray all types of trees effectively. These sprayers can transport sprayed droplets up to the top and inner parts of the tree canopy[1]. The spray droplets are spread throughout the target by the forced air jet, which also moves and lifts the foliage to allow penetration and deposits the droplets on the surface of the plant, including the undersides of leaves[2]. The airflow's kinetic energy decreases due to atmospheric resistance and the resistance created by the leaves and branches[3,4]. The kinetic energy imparted to the droplets by the pump is not sufficient, so an additional driving force-airflow is required to get the droplets to the leaves on the tree[5,6].

In agricultural production, each agricultural process is fulfilled with precision while using less energy. However, when designing high-tech machinery for agriculture, manufacturers often prioritize precision over energy efficiency, resulting in excessive energy consumption at these critical stages of agricultural production[7]. The air-assisted orchard sprayer used in horticulture requires a significant amount of energy. To improve coverage, the fan power and nozzle position should be adjusted according to the type of tree being sprayed[8,9]. On the other hand, because the air jet profile of the turbo unit is uniform and the airflow rate is high, better coverage and deposition on the target surfaces can be achieved. Besides, good distribution uniformity of the pesticide can be achieved at different points on the tree canopy[10]. Recent studies have also shown that the turbofan's airflow needs to be regulated to improve coverage and energy efficiency[11,12]. Therefore, the studies on the energy efficiency of the turbofan unit of orchard sprayers include: optimizing the number of fan blades (5-12), reducing the gap between the rotor and the fan housing, modifying the blade shape, eliminating obstacles at the fan outlet, and narrowing the discharge side of the fan[13].

Nowadays, there are many studies that focus on setting the air jet according to tree canopy with variable air rate application[14-16]. In this way, it is possible to send an airflow of proper magnitude according to the tree canopy[17,18]. However, in order to switch to a variable rate application system, the jet characteristics of the airflow must first be regulated and the airflow must be produced with less energy. Besides, the effectiveness of the system, carbon emissions, and design criteria are not in the sight of the view. In previous studies, the relationship between energy consumption and coverage of the orchard sprayer was determined, and the effects of different diameters, different numbers of fan blades, and different discharge side diameters were revealed. However, the possible effects of a different fan blade design or fan casing have not been examined on both the suction and discharge sides simultaneously. Furthermore,

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torque and fuel consumption have not been measured simultaneously, and the effects of fan efficiency and energy efficiency have not been addressed[7,10].

This study aims to evaluate the operating characteristics and efficiency of three different turbo fans—one available in the market and two with new designs—considering different suction line intake length, blade types, and outlet sections. Measurements of average air velocity in the inlet and outlet sections, fuel power, and torque data of the fans during operation were conducted for various operational situations according to EN ISO 9898 standard. Based on these wind tunnel measurements, calculations were made for fan power, PTO power, fuel power, fan efficiency, system efficiency, carbon footprint, and specific fuel consumption, providing insights for design improvements and energy optimization in orchard sprayer technology.

2 Materials and methods

2.1 Material

2.1.1 Geometrical differences in fan casing

Three different fan designs were chosen as the subject of the experiment. While one of these is widely used in the pesticide market, the other two are used with different design criteria. The turbofans have a suction diameter of 900 mm and have been shown in long V-shape (LVS) (Figure 1a), normal V-shape (NVS) (Figure 1b), and normal S-shape (NSS) (Figure 1c) designs. Each turbofan had 10 blades on the fan hub.

a. LVS b. NVS c. NSS

Figure 1 Three different fan designs

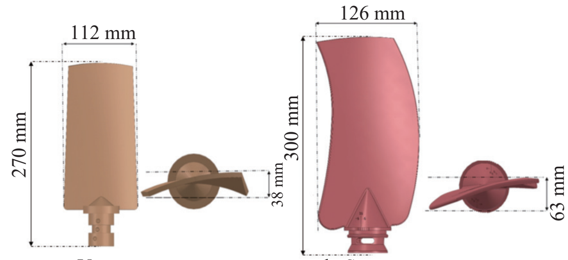
The differences and similarities of the three selected turbofans are described in Table 1. The NVS is available on the market and is considered the reference turbofan. Design parameters such as the effect of fan blade type, suction side dimensional change, and discharge width on operation were evaluated. It was anticipated that the air velocity would increase more quickly at the outlet, particularly as the LVS fan's intake side was long and its outlet side narrow, so this design was implemented. The shape of the fan blades was the only difference between NVS and NSS.

Table 1 Fan designs with different features and geometries

Fan	Number of fan blades and types	Suction side dimension/mm	Discharge side width/mm
LVS	10 (V-shaped)	520	115
NVS	10 (V-shaped)	300	150
NSS	10 (S-shaped)	300	150

2.1.2 Fan blade designs

The V-shaped fan blades used in the LVS and NVS fans are shown in Figure 2a, and the geometric design of the S-shaped fan is shown in Figure 2b. The part of the V-type fan blade that connects to the fan hub is wider and the tip of the blade is narrower. The S-type fan blade is wider than the V-type fan blade. When comparing the top views of the fan blades, the S-blade takes up more space than the V-blade for the same blade angle. The S-fan blade is also longer than the V-fan blade. Since the diameter of the suction section of the turbofan unit is 900 mm, the fan hub diameter of the S-blade is smaller than the V-blade. The top view of the fan blades shows that the distance between the two ends of the fan blades was different. There is a natural angle of attack due to the design of each fan blade. Accordingly, NSS has a notch and the angle can be changed in this way. LVS and NVS blade angle shifting system can be replaced with a bolt. The blade angles of the fans, which are the angles between the fan normal and the suction side, can be set to 15°, 30°, and 45°. In addition, each fan blade is made of polyethylene (PE).



a. V-type b. S-type

Figure 2 Side and top view of V-type and S-type fan blades

2.1.3 Wind tunnel

The wind tunnel was built according to the EN ISO 9898 standard (Figure 3). Since the diameter of the fan suction side is 900 mm according to the standard, the diameter of the tunnel built was 1350 mm (1.5 times larger than the fan suction diameter). The tunnel was made of S235JR steel, and to get a smooth surface on the tunnel, the surface was painted with an electrostatic sprayer. In this way, the sections that had the potential to disrupt the airflow in the tunnel, obstructing the flow and causing turbulence, were eliminated. In addition, the tunnel was 2000 mm long according to the standard. Three holes were drilled at intervals (the position of each hole was determined at an angular interval of 120º of the tunnel section) of 3/4 (1500 mm) of the total length of the tunnel, and air velocities were measured from the suction line using a hot-wire anemometer at predetermined locations in the tunnel section. The chassis of the portable turbofan was attached directly to the tractor. The suction line of the fan was placed completely inside the tunnel, and the fan was designed to be completely in the center of the tunnel. The gap between the fan and the tunnel was covered with linoleum to prevent air leaks.

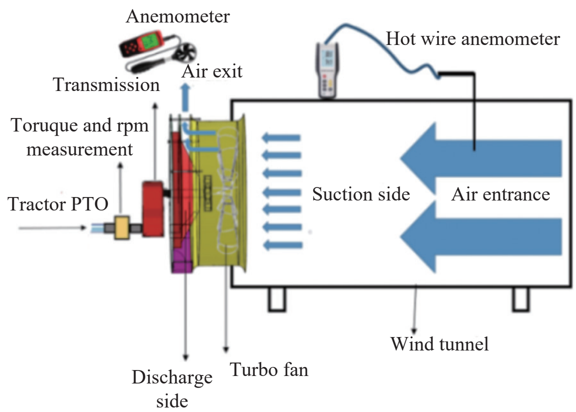


Figure 3 Schematic view of instrumentation and experimental positioning

2.1.4 Measuring equipment and vehicles

A stationary New Holland TD 95-tractor (four-cylinder diesel engine with 62.5 kW) was used to drive the PTO. Torque and angular velocity were measured between PTO and fan transmission (Grant 2020 series). The air velocity measurements from the suction side were carried out with Hotwire Anemometer (ThiesClima, Germany) from the wind tunnel. Discharge side air velocity measurements were taken from the fan discharge side using Digital Vane Type Anemometer (Tasi TA8165, Turkey). In the experiments, two flowmeters (AICHI OF05ZAT, China) were used to measure the fuel consumption values in real time. One of the flowmeters measured the amount of fuel passing through the fuel supply line between the fuel tank and the injection pump. The other measured the amount of fuel returned to the fuel tank from the injectors and injection pump. The difference between the two measures was the real-time diesel consumption. The two flowmeters were calibrated using diesel fuel prior to the studies (sensivity of flowmeter ±3%).

2.2 Method

2.2.1 Fan test procedure and test bench

The tests of the air-assisted orchard sprayer were carried out in accordance with EN ISO 9898. As required by the test standard, air velocity measurements were taken from the suction side using a hot-wire anemometer. For this measurement, the tunnel diameter was divided into 3 sections, and measurements were taken (18 data in each trial) from 6 regions along the diameter. Each measurement from the suction side was assumed as a replicate. In accordance with the relevant standard, the air velocity was measured with a vane type anemometer from 54 different points on the fan outlet section, and the fan outlet air flow rate was determined by taking the average velocities of these measured airflows. In the study, the turbofans were operated at 540 r/min PTO with 3 different blade angles (15°, 30°, and 45°) and 2 different transmission ratios, 1:3.5 (1890 r/min) and 1:4.5 (2430 r/min). Due to the uncertainty of diesel cylinder fuel explosion, variations of ±2% occurred in instantaneous torque and speed measurements.

2.2.2 Mathematical calculations

The coefficient of variation (CV) approach was used to assess the uniformity of the air jet velocity distribution.

$$CV = \frac{\sigma}{\mu} \tag{1}$$

The CV is calculated in %, σ is the standard deviation, and μ is the arithmetic mean.

The mass flow rate of the axial flow was also estimated:

$$\dot{m} = \rho \cdot A \cdot V_{\mathrm{ave}} \tag{2}$$

where, $\dot{m}$ is mass flow rate, kg/s; ρ represents air density, kg/m^3; A is the discharge area of the axial fan, m^2; and V_{ave} is the discharge side's average velocity, m/s[19].

The Bernoulli equation was used to determine the fan power (N_{fan}) by averaging the individual air discharge velocities observed at the discharge end of the axial fan:

$$N_{\mathrm{fan}} = \dot{m}\left(k\frac{V^2}{2}\right) \tag{3}$$

where, k is the kinetic energy correction factor (k=1.1); and V is the discharge velocity[13].

The PTO power (N_{PTO}) was computed as:

$$N_{\mathrm{PTO}} = \frac{M_d x n}{9550} \tag{4}$$

where, M_d is the torque, N·m; and n is the angular velocity, r/min. Then, fan efficiency $N_{\mathrm{fan,eff}}$ of the experiment was calculated as:

$$N_{\mathrm{fan,eff}} = \frac{N_{\mathrm{fan}}}{N_{\mathrm{PTO}}} \tag{5}$$

The fuel power (N_{F}) was theoretically calculated according to the fuel consumption data of each experiment:

$$N_{\mathrm{F}} = \frac{BxH}{3600} \tag{6}$$

Since the diesel engine runs at about 28% efficiency, the multiplication factor B represents the fuel consumption per hour (kg/h) and the energy value of the fuel (kg/kJ), which can be 42.0 kJ/kg[20]. The efficiency of the whole system $N_{\mathrm{sys,eff}}$, which means ratio of the fan power to the fuel power, can be calculated as:

$$N_{\mathrm{sys,eff}} = \frac{N_{\mathrm{fan}}}{N_{\mathrm{F}}} \tag{7}$$

One of the most important indicators in the experiments is special fuel consumption (SFC), which is described as the fuel consumption to complete the work.

$$\mathrm{SFC} = \frac{B}{N_{\mathrm{fan}}} \tag{8}$$

where, SFC is special fuel consumption, kg/kW·h; B is fuel consumption, kg/h; and N_{fan} is calculated fan power, kW[21].

In the published reports, the $CO_{2\mathrm{eq}}$ value was calculated by multiplying by 3728 kg for each kilogram of diesel fuel consumption, thus the carbon footprint value was calculated for each operating condition using the following equation:

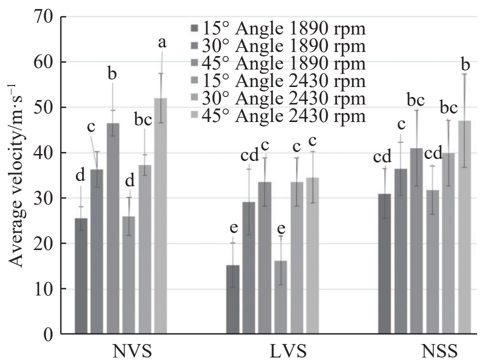
$$CO_{2\mathrm{eq}} = B \times 3728 \tag{9}$$

In this study, the air velocity data were analyzed using one-way analysis of variance (ANOVA) with a post-hoc Tukey's HSD test in SPSS 20.

3 Results and discussion

3.1 Air jet velocity comparison

According to the results of the experiments, the data of LVS and NSS fans were compared separately with the NVS (reference turbofan) to emphasize the results of design diversity. Accordingly, when comparing the air velocities of LVS and NVS fans, increasing the length of the fan suction side and narrowing the fan discharge section reduced the average air velocities, and this was true for all 6 experimental conditions (Figure 4).



Note: Different letters in the same columns indicate significant differences at p<0.05.

Figure 4 Evaluation of each fan under different experimental conditions from the discharge side

In the method devised by [22], a system was designed to adjust the discharge side of the fan, and the 70% increase in airflow was used efficiently in both stationary and mobile experiments. In theory, a narrower outlet means that the air velocity exits at a higher air velocity. Because the NVS deflector was also used in the LVS, it was clear that there was no increase in air velocity. In addition, when NVS and NSS were compared, it was found that the maximum air velocity produced by NVS was higher than that of NSS. On the other hand, the average air velocities produced by NSS were closer to each other than NVS. The NVS fan was found to have the highest average air velocity of 52 m/s at 45° blade angle and 2430 r/min. Duga et al.[12] used an 800 mm diameter axial fan with a flow regulator on the deflector unit to prevent turbulence on the discharge side and measured the maximum velocity at 30 m/s. Among these fans, it was found that the manufacturer offered 6 different options for the LVS, but in the experiments some options made no statistical sense. In fact, there were 3 different options statistically for the LVS fan. Although the manufacturer offered 6 different options for the NSS, when looking at the average air velocities, it was found that there were actually 4 different options. In addition, 6 different options were offered for NVS, but it was determined that these options were statistically 5 different possibilities. According to the results, when the blade angle of 15° was set for all three fans, whether the transmission ratio was 1:3.5 (1890 r/min) or 1:4.5 (2430 r/min) had no statistically significant impact. As a result, the stationary blades also need to be adjusted in accordance with the angles of the fan blades. In a study, Liu et al.[23] found that a 10º variation between the flow regulator and fan blade angle is required for optimum performance. When the blade angle of the fan decreases, the gap between the blades decreases, so the hub part of the fan acts as if it were wider. Because of this, and as seen in the results of the trials with 15° blade angle, the produced airflow decreased. As is well-known, the S-shaped fan blade has a narrower fan hub than the V-shaped fan hub. However, when the V-shaped blades were set to 15°, the fan hub operated as if wider.

In the experiments, it was observed that increasing the blade angle and fan rotation speed directly increased the flow at the inlet and discharge sides of the fan (Table 2). The maximum flow rate of 53 929 m^3/h was achieved with NVS. García-Ramos et al.[6] measured air velocity and flow at different blade angles and observed approximately 42 000 m^3/h with an 800 mm fan using a similar methodology. Considering the direction of rotation of the fan, the counter-clockwise rotation of the fan (viewed from the back of the fan) resulted in higher velocities on the right side of the fan, similar to the previous study[24]. The NVS demonstrated a more uniform jet profile compared to the other fans, attributed to its lower coefficient of variation (CV%). The coefficient of variation CV is expected to be <30% for the threshold of a stable jet, according to [18]. Increasing the air velocity and blade angle for each fan also contributed to making the jet smoother. It was also found that the jet was interrupted due to the narrow discharge side of the LVS. Due to the velocity difference, pressure difference, and fan blade design, the flow rate difference may occur in the suction and discharge directions of the fan[13]. The highest fuel consumption was observed in NVS. As shown in the results of the LVS fan, the long suction side of the fan reduced fuel consumption by facilitating fan suction, but the narrow discharge section reduced the air jet and airflow. Similarly, Failla et al.[7] found in their study that fuel consumption decreased when the discharge section was widened, and fuel consumption increased when the fan blade angle was increased because of air resistance.

a) Power consumption and efficiency

The fan power, PTO power, fuel power, fan efficiency, and system efficiency were calculated from the data obtained from the experiments as shown in Table 3. It was found that the efficiency of

Table 2 Torque, fuel consumption data, suction and discharge flow rates measured in the tests

Fan	Transmission/ r·min^{-1}	Angle/ (°)	Torque/ N·m	Fuel consumption/ L·h^{-1}	CV/ %	Discharge Q/m^3·h^{-1}	Suction Q/m^3·h^{-1}
NSS	1890	15	460	11.56	18	38 127	32 550
	2430	15	507	12.11	17	39 016	37 735
	1890	30	572	12.88	16	44 874	39 177
	2430	30	588	13.48	18	49 040	41 649
	1890	45	594	13.15	20	50 382	43 778
	2430	45	696	14.36	22	51 834	48 044
NVS	1890	15	207	8.56	21	28 918	26 177
	2430	15	253	9.11	21	31 894	28 155
	1890	30	487	11.88	20	37 607	34 061
	2430	30	546	12.58	14	45 812	37 872
	1890	45	606	13.29	11	47 129	44 602
	2430	45	704	14.45	11	53 929	47 864
LVS	1890	15	126	6.56	17	18 793	13 700
	2430	15	180	7.47	24	20 017	15 554
	1890	30	266	9.26	10	35 859	28 876
	2430	30	417	11.05	13	41 337	31 829
	1890	45	477	11.76	7	41 288	33 787
	2430	45	491	12.22	16	42 511	33 958

the fan in the NVS was 77.5% and the efficiency of the system in the same operating condition was 69.7%. İtmec et al.[25] found a system efficiency of 52% for a 900 mm diameter fan in their study. It was observed that the efficiency of the fan and the system increased when the air velocity and the angle of the fan blades were increased on all the fans. The importance of fan efficiency was to compare the efficiency values at the same rotation speed for the same fan casing and identify the best fan blade design. Especially for the LVS, the fan in which the discharge side was narrowed, the fan and system efficiencies were very low. In the experiments, the lowest specific fuel consumption value was obtained with the NVS fan. Specific fuel consumption is an important parameter that shows how efficient a system is. In addition, low specific fuel consumption means the engine is more efficient and consumes less fuel. The low specific fuel consumption was also important here as a design criterion, as the aim was to achieve the best operating conditions with low fuel consumption and high airflow performance. Significantly, the lowest specific fuel consumption of 0.44 kg/kW·h was achieved with NVS at 2430 r/min and 45° blade angle.

b) Carbon footprint

A well-designed fan could not only reduce the amount of fuel needed to produce the same amount of air volume, but could also reduce the amount of carbon emissions into the environment. In the study, regardless of fan design, as fan revolution speed increased, carbon emissions also increased due to the increased fuel consumption (Figure 5). On the other hand, when the blade angle was increased, the fuel consumption increased as more resistance was created to produce air, and therefore carbon emissions increased. As expected, the LVS fan had very low fuel consumption and efficiency, resulting in very low carbon emissions. However, the high air velocity and fuel consumption of the NSS fan resulted in significant carbon emissions. The NVS fan achieved the highest average air velocity with high fan efficiency and lower carbon emissions than the NSS. Lal[26] found a carbon emission of 51.4 kg CO_2eq/hm^2 for PTO-driven small tillage equipment. In addition, Failla et al.[7] found that a 700 mm diameter axial fan could generate carbon emissions of 20.45 kg CO_2eq/hm^2. As shown in Figure 4,

Table 3 Power and efficiencies of the operational conditions

Fan types	Fan rotation speed/$r\cdot min^{-1}$	Fan blade angle/(°)	Air flow rate/$m^3\cdot h^{-1}$	Fan power/kW	PTO power/kW	Fuel power/kW	Fan efficiency/%	System efficiency/%	Specific fuel consumption/ ($kg\cdot kW^{-1}\cdot h^{-1}$)
NSS	1890	15	38 127	6.9	24.5	32.5	28.0	21.1	1.44
	2430	15	39 016	7.4	26.1	34.0	28.2	21.6	1.41
	1890	30	44 874	11.2	29.0	36.2	38.7	30.9	0.99
	2430	30	49 040	14.6	30.0	36.7	48.6	39.7	0.75
	1890	45	50 382	15.8	31.0	36.9	51.0	42.9	0.72
	2430	45	52 834	24.0	34.4	40.3	69.7	59.4	0.51
NVS	1890	15	28 918	3.0	11.7	24.0	25.6	12.5	2.45
	2430	15	31 894	4.0	14.3	25.6	28.0	15.7	1.96
	1890	30	44 607	11.0	27.5	33.4	39.9	32.9	0.93
	2430	30	45 812	11.9	29.0	35.3	41.1	33.7	0.91
	1890	45	47 129	23.1	30.9	37.3	69.8	61.8	0.49
	2430	45	53 929	28.4	32.5	40.6	77.5	69.7	0.44
LVS	1890	15	18 793	0.8	7.1	18.4	11.5	4.5	7.05
	2430	15	20 017	1.0	10.2	21.0	9.8	4.7	6.42
	1890	30	35 859	5.7	15.0	26.0	38.0	22.0	1.40
	2430	30	41 337	8.7	23.6	31.0	37.1	28.2	1.09
	1890	45	41 288	8.7	27.0	33.0	32.3	26.4	1.16
	2430	45	42 511	9.5	27.7	33.5	34.3	28.4	1.05

there was no statistical difference in some airflow operating parameters between fan types. Choosing a low carbon emission application with equivalent airflow values can lead to a more environmentally beneficial outcome.

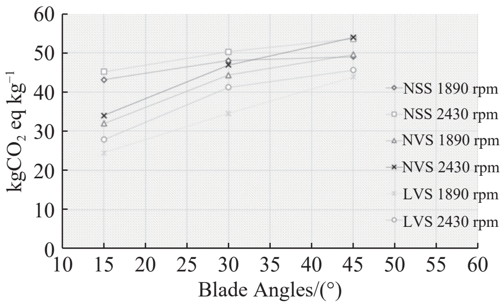


Figure 5 Carbon emissions (kgCO_2eq/kg) at different blade angles and fan speeds

4 Conclusions

In this study, the efficiency of turbofans available on the market was investigated, looking at two different design criteria to improve fan performance. Following the methodology outlined in EN ISO 9898, these variations were tested in a laboratory environment. Several key findings emerged from our research:

When using an S-shaped blade design, a decrease in average air velocity was observed compared to the V-shaped blade design. V-shape blade configuration demonstrated superior volume flow rate. Lengthening and widening the fan blades while narrowing the fan hub did not increase air velocity. In fact, this modification reduced both system and fan efficiency, resulting in higher fuel consumption and carbon emissions.

Reducing the blade angle resulted in the hub section of the fan acting as a wider surface, which reduced the amount of airflow produced. Adjusting the blade angle was critical to optimizing performance, but attention must also be paid to adjusting the stationary blades on the suction side to match this adjustment. Although the manufacturer offers different application options to the farmer, it was found that the average air velocity data were statistically similar.

Contrary to popular belief, the narrowing of the fan's discharge section did not have an effect on air velocity. Instead, it caused a decrease due to a blockage in the discharge section. Furthermore, the deflector should be re-designed when the fan casing design was changed. On the other hand, extending the fan suction side reduced the fuel consumption by facilitating suction.

For same fan casing, fan efficiency was an indicator to see the effect of blade design. With specific fuel consumption, it was revealed which fan could produce more airflow with less energy. It was found that simply extending the suction side of the NVS fan, without changing the blade type or discharge side, could further improve fan efficiency and reduce specific fuel consumption.

It was statistically proved that several of the operational conditions available to users in the design of fans made no sense in airflow. It also demonstrated that the turbofan unit was far from suitable for every orchard.

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