

CONCEPTUAL DESIGN, TESTING AND MANUFACTURING OF AN INDUSTRIAL
TYPE ELECTRO-HYDRAULIC VACUUM SWEEPER

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ABSTRACT

CONCEPTUAL DESIGN, TESTING AND MANUFACTURING OF AN INDUSTRIAL TYPE ELECTRO-HYDRAULIC VACUUM SWEEPER

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In this thesis, conceptual design, testing, development and manufacturing processes of the cleaning (elevator and fan system) and electro-hydraulic systems of an industrial type vacuum sweeper are presented. Thesis is financially supported by Ministry of Science, Industry and Technology (Turkey) and Müsan A.Ş. (Makina Üretim Sanayi ve Ticaret A.Ş.) under the SAN-TEZ projects with numbers 00028.STZ.2007-1 and 00623.STZ.2010-1. The main purpose is to make critical design changes on existing fan system, designing a new elevator system and eventually obtaining efficient and powerful cleaning system. For design, Catia and SolidWorks softwares are used. Within the SAN-TEZ project, all CFD solutions were provided by Punto Engineering.

Unlike many industrial type vacuum sweepers, new design will be electrically and electro-hydraulic controlled. All cleaning system of new 'MÜSAN Vacuum Sweeper' will be activated by using hydraulic motors (traction system including hydraulic system is driven by the brushless DC electric motor as well) and the power of all these systems is supplied by batteries which are placed in the middle of the vehicle.

Elevator and fan system can be considered as a group for a street sweeper for cleaning operations. Fan and elevator systems both gain an important place especially in cleaning operations due to lifting heavy and small particles from the ground. Fan system is used for sucking the small materials and dust by vacuum and elevator system is used to elevate heavier materials such stones, bottles, cans. Therefore, it is essential to design an efficient and powerful fan and elevator system for a street sweeper.

The thesis work includes the design, development, supervision of manufacturing, simulation and testing of the cleaning (elevator and fan systems) and electro-hydraulic system of the street cleaners.

Keywords: Electrically controlled street sweeper, cleaning system, CFD (computational fluid dynamics) analysis and experiments of the fan system, manufacturing, electro-hydraulic.

ÖZ

ENDÜSTRİYEL TİP ELEKTRO-HİDROLİK BİR VAKUMLU SÜPÜRÜCÜNÜN KAVRAMSAL TASARIMI, TEST EDİLMESİ VE İMALATI

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Bu tezde, endüstriyel tip vakumlu süpürücüye ait temizleme (elevatör ve fan sistemi) ve elektro-hidrolik sistemlerinin kavramsal tasarımı, test edilmesi, geliştirilmesi ve imalatı anlatılmaktadır. Tez, 00028.STZ.2007-1 ve 00623.STZ.2010-1 proje numaralı SAN-TEZ projeleri kapsamında, T.C. Bilim, Sanayi ve Teknoloji Bakanlığı ve Müsan A.Ş. (Makina Üretim Sanayi ve Ticaret A.Ş.) tarafından desteklenmektedir. Temel amaç, mevcut fan sistemi üzerinde kritik tasarım değişiklikleri yapılması, yeni bir elevatör sistemi tasarlanması ve sonuçta etkin ve güçlü bir temizleme sistemi elde etmektir. Tasarım çalışmalarında Catia ve SolidWorks yazılımlarından yararlanılmıştır. SAN-TEZ projesi kapsamında, bütün CFD çözümlerinde Punto Mühendislik'ten yardım alınmıştır.

Birçok endüstriyel tip vakumlu süpürücülerin aksine, yeni tasarım elektrik ve elektro-hidrolik kontrollü olacaktır. Yeni 'MÜSAN Vakumlu Süpürücünün' tüm temizleme sistemi hidrolik motorlardan (yürüme sistemini oluşturan hidrolik sistem de fırçasız DC elektrik motoru ile tahrik edilecektir) oluşacaktır ve bu sistemlerin gücü aracın ortasına yerleştirilen aküler ile sağlanacaktır.

Elevatör ve fan sistemi, cadde temizleme işlerinde kullanılan araçlar için bir grup olarak düşünülebilir. Fan ve elevatör sisteminin hafif ve ağır partiküllerin yerden kaldırılmasında özellikle temizleme operasyonlarında önemli bir yeri vardır. Fan sistemi küçük materyallerin ve tozun vakum ile çekilmesini sağlarken elevatör sistemi taş ve şişe gibi daha ağır materyallerin kaldırılmasında kullanılır. Bu sebepten dolayı verimli ve güçlü fan ve elevatör sistemi bir cadde süpürücü için olmazsa olmazdır.

Bu tez çalışması cadde temizleyicilerinin temizleme (elevatör ve fan sistemleri) ve elektro-hidrolik sisteminin tasarımı, imalat denetimleri, simülasyon ve deneysel çalışmalarını içermektedir.

Anahtar kelimeler: Elektrik kontrollü cadde süpürücü, temizleme sistemi, fan sisteminin CFD (hesaplamalı akışkan dinamiği) analizi ve deneyleri, imalat, elektro-hidrolik.

To My Parents

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TABLE OF CONTENTS

ABSTRACT	IV
ÖZ	VI
ACKNOWLEDGMENTS.....	IX
TABLE OF CONTENTS	X
LIST OF FIGURES	XIII
LIST OF TABLES.....	XVII
LIST OF SYMBOLS AND ABBREVIATIONS	XIX
CHAPTERS	
1. INTRODUCTION	1
1.1. Mechanical Broom Sweepers.....	2
1.2. Vacuum Sweepers	4
1.3. Regenerative Air Sweepers	5
1.4. Cleaning System for Vacuum Sweepers and Market Demands.....	6
1.5. Organization of the Thesis	7
2. LITERATURE SURVEY	9
2.1. General Properties of Former Müsan VSM 060 Vacuum Sweeper.....	9
2.2. Cleaning Process of Former Müsan VSM 060 Vacuum Sweeper.....	10
2.3. Comparison with the Other Products	12
2.3.1. Echo Sweepers	12
2.3.2. Nilfisk Advance RS 850 and RS851	14
2.3.3. Madvac CN100	19
2.3.4. Tennant Green Machine 636HS	21

3. OBJECT OF THE PRESENT INVESTIGATION	25
4. DESIGN AND MANUFACTURING OF THE CLEANING SYSTEM	29
4.1. Elevator System	29
4.1.1. Applications of Elevator Systems	29
4.1.2. Theory and Calculations.....	31
4.1.3. Elevator System Design	38
4.1.4. Manufacturing of the Elevator System.....	43
4.2. Vacuum System Development and Design	45
4.2.1. Fan Selection and Centrifugal Fans.....	45
4.2.2. Characteristics of Former Müsan Fans.....	46
4.2.3. Velocity Vector Diagrams and Triangles	48
4.2.4. CFD Analyses	54
4.2.5. Fan Sizing and Development.....	62
4.2.6. An Investigation on Number of Blades	67
4.2.7. Manufacturing of the Rotors and Vacuum Fans.....	68
4.3. Manufacturing of the Other Sub-Systems before Experiments	73
4.3.1. Trash Container and Filter.....	73
4.3.2. Battery Cassette.....	76
4.3.3. Cabin	77
5. EXPERIMENTS	79
5.1. AMCA Standard to be Used.....	79
5.2. Manufacturing of Test Set-up.....	80
5.3. Properties of Test Devices	86
5.4. Testing and Calculation Methods	90
5.5. Experiments of Former Müsan Fans.....	93
5.5.1. Testing of Curved and Straight Blade Fans	94
5.5.2. Comparison of Curved and Straight Blade Fans	95
5.6. Experiments of New Fans.....	98
5.6.1. Experiments of Straight Blade Fans	99
5.6.2. Performance and Efficiency Curves.....	102
5.7. System Tests and Operation Points	104
5.7.1. System Test Procedure and Test Devices.....	104
5.7.2. System Experiment 1: With ‘Fan Assembly Box’	109
5.7.2.1. Calculations of System Experiment (1)	110
5.7.2.2. Determining the System Characteristic and Operation Point.....	112
5.7.2.3. Speeds and Pressure in Fan Assembly Box	113
5.7.2.4. Calculation of the Filter Loss Coefficient (FLC).....	116

5.7.3.	System Experiment 2: Directly Connected	121
5.7.3.1.	Calculations of System Experiment (2)	123
5.7.3.2.	Determining the System Characteristic and Operation Point.....	125
5.7.3.3.	Calculation of the Filter Lost Coefficient (FLC).....	126
6.	ELECTRO-HYDRAULIC TRACTION SYSTEM DESIGN	129
6.1.	Determining the Torque Requirement	129
6.2.	Hydraulic Equipments	132
7.	ACCESSORIES.....	135
7.1.	Vacuum Hose	135
7.2.	Power LED	138
8.	DISCUSSION OF RESULTS	141
9.	CONCLUSION.....	144
	REFERENCES	145
	APPENDICES	
A.	OVERALL DIMENSIONS AND TECHNICAL DRAWINGS	148
B.	HYDRAULIC SYSTEM SCHEMATIC	151
C.	HYDRAULIC SYSTEM COMPONENTS	152
D.	ELECTRIC SYSTEM (PLC) SCHEMATICS.....	153
E.	MANUFACTURING PHOTOS OF THE NEW SWEEPER.....	155

LIST OF FIGURES

FIGURES

Figure 1 Elgin Road Wizard Mechanical (Broom) Sweeper [5].....	3
Figure 2 Elgin's Crosswind Regenerative Air Sweeper [5].....	6
Figure 3 Müsan VSM 060 Sweeper [6]	10
Figure 4 Cleaning Process of VSM 060 Vacuum Sweepers [7].....	11
Figure 5 Echo 40 Series [9]	12
Figure 6 Nilfisk Advance RS 850 [9].....	14
Figure 7 Nilfisk Advance RS 851 [11].....	14
Figure 8 General Properties of Nilfisk Advance Sweeper [10].....	15
Figure 9 Placement of Some Componenets of Nilfisk Sweeper RS 850 [10].....	17
Figure 10 Centrifugal Fan of Nilfisk Sweeper RS 850 [10]	18
Figure 11 Internal Fan & Airflow System of Madvac CN100 [12].....	20
Figure 12 Madvac CN100 [12]	20
Figure 13 Centrifugal Fan of 636HS [13].....	21
Figure 14 Green Machine 636HS during discharging [13].....	22
Figure 15 Tennant Green Machines 636HS [13].....	22
Figure 16 Vacuum System of Tennant Green Machines 636HS [13]	23
Figure 17 An Illustration of Elevator System	28
Figure 18 Inclined Elevator Conveyor of Centurion Sweeper [14]	29
Figure 19 Elevator Conveyor Parts [14]	30
Figure 20 Determining the Polar Distance (I) [16]	32
Figure 21 Locations of Sub-Assemblies on Sweeper	38
Figure 22 Initial Design of Elevator System.....	39
Figure 23 Second Design of Elevator System with Electric Motors	40
Figure 24 Operation principle of the Elevator System	41
Figure 25 Previous Electric Motor for Elevator System (440W DC).....	42
Figure 26 Hydraulic Motor (BR 160) for the Elevator System.....	42
Figure 27 Assembly of the Protective Side Sheets to the Chassis	43
Figure 28 Elevator System and Band & Blades	44

Figure 29 Assembly of Hydraulic Motor to System.....	44
Figure 30 Centrifugal Fan Blades [18], [19].....	46
Figure 31 Centrifugal Fan of Former Müsan Sweeper	47
Figure 32 2D View of the Former Fan	47
Figure 33 Former Rotor (to be drawn velocity triangles).....	48
Figure 34 Velocity Triangles on Blade Tips	49
Figure 35 Pressure&Velocity Distributions of Former Fan (D=250 mm)	55
Figure 36 Velocity Distribution of Trash Container of the Current Sweeper.....	56
Figure 37 Arrangements of the Vacuum System Components.....	58
Figure 38 CFD Analysis of System (Determining approximate requirements)	60
Figure 39 Streamlines of New Design.....	61
Figure 40 Former Müsan Fan (D= 250 mm).....	68
Figure 41 Former Müsan Rotors (Curved & Straight).....	69
Figure 42 Welding of Blades around the Fan Rotor (D=400 mm & 8 Blades).....	69
Figure 43 Welding of Blades around the Fan Rotor (D=400 mm & 10 Blades).....	70
Figure 44 Mounting of the Rotor to the Housing.....	70
Figure 45 Assembly Box for the Fans (Initial Design).....	71
Figure 46 Hydraulic Motor for Fans.....	71
Figure 47 Arrangement of the fans directly (final design)	72
Figure 48 Mounting of the Fan group to Sweeper	72
Figure 49 Trash Container and Thruster Slot (Initial Design)	73
Figure 50 Electro mechanic Thruster (Initial Design).....	74
Figure 51 Inside of the Trash Container and Filters	74
Figure 52 Trash Container and Hydraulic Cylinder Slot	75
Figure 53 Battery Cassette (with slots)	76
Figure 54 Placement of the Battery Cassette on the Sweeper	77
Figure 55 Cabin Framework (Initial Design)	78
Figure 56 The Sweeper before Experiments.....	78
Figure 57 Schematic of the Test Set-up.....	80
Figure 58 3D View of Test Set-up.....	82
Figure 59 Probe Fixer mounted on the Pipe.....	82
Figure 60 Test Set-Up for the Experiments of Current Fans	83
Figure 61 Throttling Valve (Conical).....	84

Figure 62 Flow Straightener.....	84
Figure 63 Outer Dimensions of the Test Setup	85
Figure 64 Test Setup for the New Fans	86
Figure 65 Testo-435 Measurement Device [22]	88
Figure 66 Mounting of Electric Motor and Invertor.....	88
Figure 67 Energy Analyzer for Power Measurement.....	89
Figure 68 Area Sectors and Velocity Distribution through the Pipe Section.....	90
Figure 69 Former Fans and Initial System Curve (CFD)	93
Figure 70 Set-up and Experiments of the Former Fans	94
Figure 71 Performance Curves of the Former Fans	96
Figure 72 Efficiency Curves of the Former Fans	97
Figure 73 Comparison of the CFD and Experiment Results (Former fans).....	98
Figure 74 Test Setup for New Fans	99
Figure 75 Comparison of the Different Number of Blades (3000 rpm).....	102
Figure 76 Comparison of the Different Number of Blades & Rotations.....	102
Figure 77 Efficiency Curves of Different Blades of the VSM 08.....	103
Figure 78 Comparison of VSM 08 (NEW) and VSM 06 (OLD) Fans.....	103
Figure 79 Measuring Air Speed at Fan Outlet with Pitot Tube.....	104
Figure 80 An Illustration of the Measurement Style.....	105
Figure 81 Inclined Manometer and Pitot Tube	105
Figure 82 Measuring with 'U' Type Manometer	106
Figure 83 Drilled Holes at Vehicle.....	107
Figure 84 Measuring Air Speeds inside of the Container (Filter Inlet).....	108
Figure 85 Measurement Points (Fan Outlet)	109
Figure 86 Characteristics Curve and Operation Point	113
Figure 87 Mounting of the Fans	114
Figure 88 Mounting of the Electric Motors.....	114
Figure 89 Assembly Box is on the Vehicle	115
Figure 90 Measurement Points on Fan Assembly Box.....	115
Figure 91 Filters and Brushless Electric Motor (250W)	117
Figure 92 Assembly of the Filters to the Vehicle (Top View)	117
Figure 93 Inside of the Container (Filter Bottom View).....	118
Figure 94 'U' Manometer and Drilled Measurement Holes	119

Figure 95 Illustration of Measurement Points and Container.....	119
Figure 96 Views of Connection of the Fans Directly	121
Figure 97 Electric Motors and Fans	122
Figure 98 Taking measurements during Experiment-2.....	122
Figure 99 Characteristics Curve and Operation Point	126
Figure 100 Assembly of the Filters to the Vehicle (Top View)	127
Figure 101 Torque-Rotation Graphic for the Position-1.....	130
Figure 102 Torque-Rotation Graphic for the Position-2.....	131
Figure 103 MD10V-28 Hydrostatic Pump (Samhydraulik)	133
Figure 104 Wheel Type Hydromotor	134
Figure 105 Characteristics of the Hydromotor.....	134
Figure 106 Portable Vacuum Hose (MADVAC CN100).....	135
Figure 107 Collecting the Trashes with Vacuum Hose (MADVAC) [12].....	136
Figure 108 Elastic Vacuum Hose (MADVAC 101).....	136
Figure 109 Pressure Distribution of the Section	137
Figure 110 LED daytime running lights of Audi A4 [23]	139
Figure 111 LED Aircraft Landing Light [24]	139
Figure 112 LED Aircraft Sidewall Light [24].....	140
Figure 113 Overall Dimensions of the New Müsan Sweeper (VSM 08).....	148
Figure 114 Elevator System of the New Müsan Sweeper (VSM 08).....	149
Figure 115 Fan System of the New Müsan Sweeper (VSM 08)	150
Figure 116 Hydraulic System of the New Müsan Sweeper (VSM 80).....	151
Figure 117 Electric System of the New Müsan Sweeper (VSM 08).....	153
Figure 118 Electric System Inputs/Outputs and LCD Screen	154
Figure 119 Prototype Vehicle (VSM 080).....	155
Figure 120 VSM 080 Sweeper before the External Cladding and Accessories	156

LIST OF TABLES

TABLES

Table 1 Technical Specifications of VSM 060 vacuum sweeper [6].....	9
Table 2 Technical Specifications of ECHO 40 SERIES Sweeper [8].....	13
Table 3 Technical Specifications of Nilfisk Sweepers [9], [10], [11]	16
Table 4 Part List of given pictures in Figure 9 and Figure 10	18
Table 5 Technical Specifications of Madvac CN100 Sweepers [12].....	19
Table 6 Technical Specifications of Green Machine 636HS [9]	21
Table 7 Resistance and Manufacturing Coefficients [16].....	35
Table 8 Overall Results of Elevator System	37
Table 9 Properties of Former Fan Blade	49
Table 10 Overall Results of Former Fans	53
Table 11 Properties of the Current Müsan Fan	54
Table 12 Typical solid particle sizes [2]	58
Table 13 Pneumatic solid transportation velocities [2].....	59
Table 14 Properties of the Former Fan (Backward-Curved Type Blade)	62
Table 15 Properties of the Former Fan (Backward-Straight Type Blade)	62
Table 16 Comparison of Results (Rotor Diameter is 300 mm)	64
Table 17 Blade Speeds (Rotor Diameter is 300 mm)	65
Table 18 Comparison of Results (Rotor Diameter is 400 mm)	66
Table 19 Blade Speeds (Rotor Diameter is 400 mm)	66
Table 20 General Properties of New Fan Blade	67
Table 21 Diameter & Area Values of the Initial Test-Setup	91
Table 22 A sample Measurement Table belonging to Former Müsan Fans.....	92
Table 23 Experiment Results (Former Fans) and Comparison Table.....	95
Table 24 Test Results (New Fans) and Comparison Table	100
Table 25 Experiment Data Table at Outlet	110
Table 26 Experiment Results Table	112
Table 27 Experiment Result Table for FLC	120
Table 28 Experiment-2 Data Table at Outlet	123

Table 29 Experiment-2 Results Table	125
Table 30 Experiment-2 Result Table for FLC	127
Table 31 Position-1: ($a_{\max}=0$), ($V_{\max}=15$), Gradient ($\Theta=15^0$).....	130
Table 32 Position-2: ($a_{\max}=2.5$), ($V_{\max}=15$), Gradient ($\Theta=0^0$).....	131
Table 33 Properties of the MD10V-28 Hydrostatic Pump (Samhydraulik)	132
Table 34 Properties of the Wheel Type Hydromotor (HR L130)	133
Table 35 Equipment List to be used in the Hydraulic System of VSM 08	152

LIST OF SYMBOLS AND ABBREVIATIONS

AMCA	: Air Movement and Control Association
CFD	: Computational Fluid Dynamics
ΔP	: pressure difference
Q	: flow rate
V_{m2}	: meridional velocity
U_2	: peripheral velocity
W_2	: relative velocity
V_{inlet}	: inlet velocity
g	: grams
h	: hour
s	: second
hp	: horsepower
kg	: kilogram
Pa	: pascal
kPa	: kilopascal
MPa	: mega pascal
kW	: kilowatt
l	: liter
mbar	: millibar
m	: meter
cm	: centimeter
mm	: millimeter
T	: torque
N	: Newton
rpm	: revolutions per minute
m^3/s	: cubic meter per second
DC	: direct current
V	: volt

CHAPTER 1

INTRODUCTION

Mechanical street sweepers aren't as old as the streets they clean, but it might surprise you to learn that they do predate the automobile. Like many inventions devised during the Industrial Revolution, the street sweeper's actual parentage is a bit unclear. Some sources claim C.S. Bishop invented a mechanized street sweeper in 1849, while others say C.B. Brooks, inventor of the paper hole puncher, came up with his street sweeper truck in 1896.

In any case, by the time the automobile became widely available in the U.S. in the early 20th century, street sweepers were already on the scene and keeping things clean. Officials for the city of Boise, Idaho, bought the first Elgin street sweeper in 1914, and the Illinois-based company has been in business ever since.

The first mechanical street sweepers may have predated the automobile, but they would have relied on horse-drawn carts to move about the city. The street sweeper truck ushered in the use of horseless carriages (you might know them as cars) to move the street sweeper around town. As technology has improved and environmental concerns have become more important, street sweepers have become quieter, cleaner and more efficient. Some are even using alternative fuels as we continue on into the 21st century. [1] Industrial type sweepers can be classified in two ways according to their structure and their cleaning method.

There are two types of sweepers according to structure: [2]

- Walk-behind sweepers
- Ride-on sweepers

And three types of sweepers according to the way of cleaning:

(February, The Stormwater Manager's Resource Center Web Site, 2009)

- Mechanical broom sweepers
- Vacuum sweepers
- Regenerative air sweepers

1.1. Mechanical Broom Sweepers

Mechanical broom sweeping technology may be explained as cleaning with a broom and a dustpan [2]. For years, mechanical broom sweepers were the only machines that were used for road sweeping by municipalities and departments of transportation. Mechanical broom sweepers are still the primary machines in use.

Typically, these machines have a 'main broom' that runs transversely from one side of the sweeper to the other such that the broom bristles contact the paved surface the full width of the sweeper unit. The broom rotates in a clockwise fashion when viewed from the left side of the vehicle and collected debris is swept onto some type of a conveyor belt for transfer to a containment hopper.

Mechanical broom machines may or may not be equipped with a 'gutter broom' on one or both sides of the sweeper. Gutter brooms are relatively small (typically 300 to 600 mm in width), are located to the left, right, or both sides of the sweeper, and are primarily used to transfer debris into the path of the main broom. Even though mechanical sweepers are usually outfitted with a series of water spray nozzles, because they have no vacuum component, they still tend to create a substantial amount of dust in dry weather.

From an environmental point of view, mechanical broom sweepers may have a negative effect on the amount of micron size particle pollution. This is because the action of the broom tends to break larger particles down into smaller ones, creating more small-micron particles than there were to start with. Since, debris pick-up is via an elevator, rather than involving any type of air/suction action, large amount of these small particles are left on the pavement surface. [2]



Figure 1 Elgin Road Wizard Mechanical (Broom) Sweeper [5]

Advantages: Mechanical broom sweepers remain the standard for sweeping extremely heavy or packed-down material.

Disadvantages: A mechanical sweeper is a poor choice where environmental concerns exist about small particle pollution or air quality. Also, mechanical sweepers are significantly more expensive to maintain than vacuum or regenerative air sweepers, due to having so many moving parts including continuous ground contact by main broom and mechanical movement by the elevator system.

1.2. Vacuum Sweepers

Vacuum sweepers may be compared to a household vacuum system. An engine powers a fan, which creates vacuum/suction. Typically, there is a suction inlet on one side of the sweeping head, and then the 'used' air is constantly exhausted during the sweeping process. Most vacuum sweepers do not have an air blast that transfers debris to the vacuum opening. Instead; they employ some type of broom system to brush debris toward the vacuum opening in the head. Water may or may not be used in vacuum sweeper cleaning system.

Nowadays, vacuum sweepers are generally equipped with a sophisticated filtration system for dust containment. With this system, sweeper can provide cleaning to a small-micron level and become preferable for any application where dangerous or toxic materials are present. [2]

Advantages: Vacuum sweepers provide thorough cleaning near the curb line. The sweeper can clean air PM10 (aerodynamic diameters are smaller than 10 μm , provided by the European Certification System named EUnited for road sweepers) if a sophisticated filtration system is used and therefore can be a good choice for industrial cleanup situations. Beside an efficient vacuum (or fan) system, unlike current design of MÜSAN V06 Sweeper Vehicle (which works with hydraulic system), new design of MÜSAN Sweeper will generate less dust particles by using an elevator system for bigger garbage such stones, cans, bottles.

Disadvantages: Vacuum sweepers working on areas such roads and dirtier places where not only dust particles to be vacuumed vehicle cannot pick up large garbage. Since suction tubes are smaller than regenerative air sweepers, on uneven surfaces required vacuum may not be enough to remove garbage.

1.3. Regenerative Air Sweepers

Regenerative air sweepers, have hydraulic systems that use air jets to churn up the dirt on the road, then swirl it toward the center of the truck. Negative pressure under the truck creates a vacuum that carries the dirt up into the receptacle at the truck's rear. Filters clean the air in the receptacle and reuse that air to loosen debris on the street, while water is sprayed to keep down the dust.

Some sweepers use spinning brushes instead of regenerative air or water to get the debris underneath the truck, where a vacuum moves it into the storage hopper. These models are usually smaller than the big street or highway sweepers, and they're used at industrial sites where cleaning up particulates is paramount. [1]

Advantages: Regenerative air sweepers can clean a wide range of debris in a large variety of situations. The amount of exhausted, pollutant-laden air is lower than the other sweeper types. There are fewer moving and wear parts than mechanical sweepers. They can be used to clean trapped basins by adding hand hose.

Disadvantages: During regeneration process some air is lost. Regenerative air sweepers can't handle millings and other extremely heavy-duty applications as well as can mechanical sweepers. They use water for dust suppression, which leaves some dissolved small-micron debris in pavement cracks and on the surface.



Figure 2 Elgin's Crosswind Regenerative Air Sweeper [5]

1.4. Cleaning System for Vacuum Sweepers and Market Demands

Nowadays, 'Vacuum Sweepers' are used widely in cleaning operations on roads or any other places like airports, harbors, train stations, ports and parking areas. Since using a cleaning machine is the fastest and cheapest method to acquire clean environment, official and private companies or municipalities are using these vehicles.

As given in the previous captions, there are three types of sweepers according to the way of cleaning. Combining all of these systems together effectively, efficient and powerful cleaning system of a vacuum sweeper is acquired.

Today, since customer demands are rising on sweeper vehicles generating less dust particles, less air pollution (health), less sound and optimum costs basically, Sweeper manufacturers are working on the 'Cleaning System' especially to gain active role on market and also to supply customer demands.

This thesis consist of 'Cleaning System' and 'Electro-Hydraulic System' of new 'Sweeper Design' under the SAN-TEZ Project continuing with the corporation of METU, MÜSAN A.Ş. (Sincan-Ankara) and Punto Engineering (CFD solutions). As you can see in next chapters, cleaning system of new design includes the elevator and vacuum system mainly.

1.5. Organization of the Thesis

The outline of the thesis can be summarized as follows:

In Chapter 2, technical specifications of the former Müsan VSM 060 sweeper are presented, cleaning system of former and new design is compared, from the cleaning system point of view; sweepers in market and former design are compared.

In Chapter 3, the object of the SAN-TEZ research and development, work share of the project members and general design properties of the new design are presented.

In Chapter 4, calculations and design process of elevator system, development and design of fan system, manufacturing of the whole cleaning system (fan and elevator systems), and manufacturing of the other individual sub-systems are presented.

In Chapter 5, information about the test process, manufacturing test set-ups, performance and system experiments and discussion of the results with the graphs are presented.

In Chapter 6, determining the torque requirements, torque-rotation graphs, transition to the electro-hydraulic system are presented.

In Chapter 7, Illumination and other accessories such vacuum hose, LED lighting are presented.

In Chapter 8, discussion of results is presented.

In Chapter 9, conclusion is presented.

CHAPTER 2

LITERATURE SURVEY

In study [2], many sweeper products in the market were investigated and technical specifications of them were compared. Comparison of products generally and comparison of cleaning systems (fans) of products in the market and differences from the MÜSAN design are given in this chapter.

2.1. General Properties of Former Müsan VSM 060 Vacuum Sweeper

Table 1 Technical Specifications of VSM 060 vacuum sweeper [6]

	One Side Brush	Two Side Brush
Sweeping Area Max. (m ² /h)	12.500	15.000
Speed(Traveling) (km/h)	0-10	0-10
Sweeping Range (m)	1,35	1,55
(With Central Brush) (m)	1,05	1,05
Engine Power	20 hp	20 hp
Engine Starting	Electric Battery	Electric Battery
Waste Container (l)	400	400
Suction System	Hydraulic Motor With Vacuum Aspirator	Hydraulic Motor With Vacuum Aspirator
Filter Area (m ²)	6	6
Filtering System	Electro-mechanic Vibrator	Electro-mechanic Vibrator
Minimum Turning Radius (m)	1,55	1,55
Brakes	Service & Parking Front	Service & Parking Front
Max. Grade ability	% 26/15°	% 26/15°
Weight (kg) With Water Tank (100l)	1.000	1.000
Discharging Height (m)	1,2	1,2
Width (m)	1,35	1,55
Length (m)	2,15	2,15
Height (m)	2,3	2,3
Fuel Consumption (l/h)	3,5	3,5



Figure 3 Müsan VSM 060 Sweeper [6]

2.2. Cleaning Process of Former Müsan VSM 060 Vacuum Sweeper

As shown in Figure 4, cleaning system of former Müsan VSM 060 Sweeper consists of brooms (main and side brushes) and vacuum system including 250 mm diameter curved blade fan. Detailed information and calculations about the current fan will be given in next Chapters.

Cleaning process is as follows: Debris on the ground is collected to the hopper with the aid of main and side brooms. The garbage and dusts on the ground are vacuumed by the centrifugal fan through the filter and are collected at the upper part of the hopper.

The main difference between the old and new MÜSAN Sweeper 'Cleaning System' is 'Elevator System' and surely efficient 'Fan (Vacuum) System'.

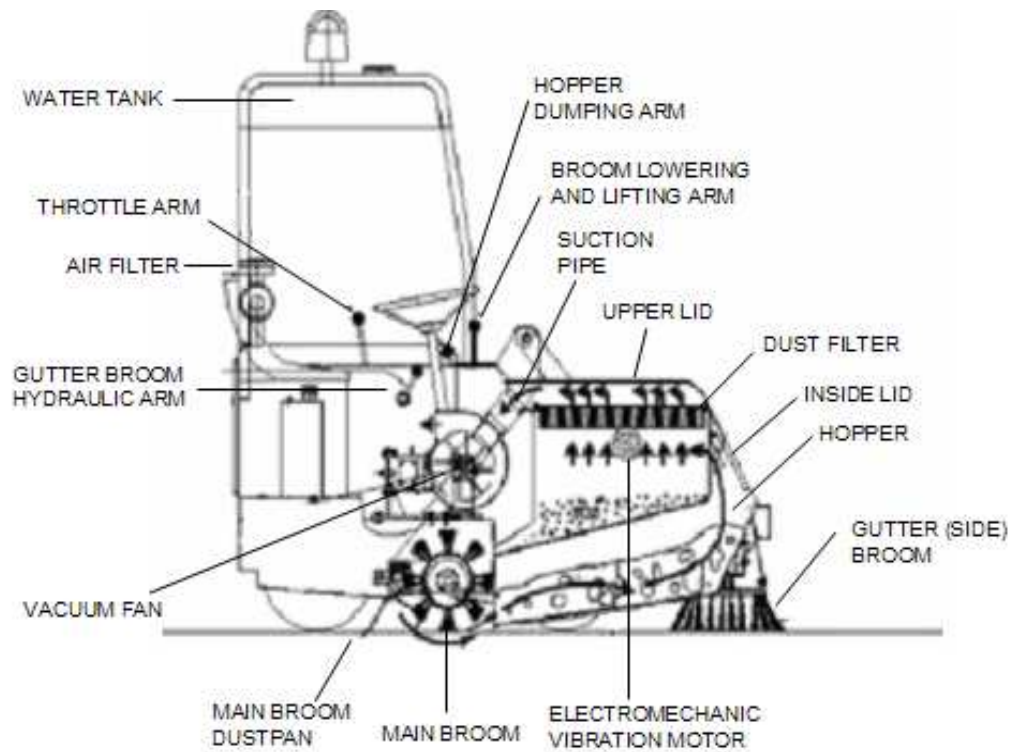


Figure 4 Cleaning Process of VSM 060 Vacuum Sweepers [7]

Vacuum fan is driven by a high speed hydraulic motor and starts to work automatically when the main broom is activated. An electro-mechanic vibrator is utilized for filter cleaning. Filtered air is used as oil cooler in the system. Total hopper volume is 400 l. Operator manually checks out the debris level by the help of the cover at the front side.

2.3. Comparison with the Other Products

New Müsan VSM 080 sweeper is compared with the former one and with the other sweepers in the market. The vehicles that will be compared are listed as follows:

- Echo 40 Series
- Nilfisk Advance RS 850
- Nilfisk Advance RS 851
- Madvac CN100
- Tennant Green Machine 636HS

2.3.1. Echo Sweepers



Figure 5 Echo 40 Series [9]

Table 2 Technical Specifications of ECHO 40 SERIES Sweeper [8]

Standard Sweeper	ECHO 40 (US - Metric)	ECHO 44 (US - Metric)
Width	53 in (1350 mm)	53 in (1350 mm)
Length	98.5 in. (2500 mm)	98.5 in. (2500 mm)
Height	62 in. (1570 mm)	62 in. (1570 mm)
Weight	-	-
Model D	2756 lbs. (1250 kg)	2756 lbs. (1250 kg)
Model E	-	5535 lbs. (2510 kg)
Model G/LP	2712 lbs. (1230 kg)	2712 lbs. (1230 kg)
Power Source	-	-
Model D	Diesel	Diesel
Model E	-	Electric
Model G/LP	Gas or LP Gas	Gas or LP Gas
Theoretical sweeping capacity per hour	-	-
Model D	226000 sq. ft. (21000 m ²)	226000 sq. ft. (21000 m ²)
Model E	-	135600 sq. ft. (12600 m ²)
Model G/LP	226000 sq. ft. (21000 m ²).	226000 sq. ft. (21000 m ²)
Cleaning path with main broom	39 in. (1000 mm)	39 in. (1000 mm)
Cleaning path with main broom and right side broom	55 in. (1400 mm)	55 in. (1400 mm)
Cleaning path with main broom, right and left side broom	67 in. (1700 mm)	67 in. (1700 mm)
Hopper Capacity	12.7 cu. ft. (360 liters)	12.7 cu. ft. (360 liters)
Discharge Height	0 to 60 in. (0 to 1510 mm)	0 to 60 in. (0 to 1510 mm)
Suction system	Turbine	Turbine
Filtering system	Polyester multi-bag 91.5 sq. ft. (8.5 m ²)	Polyester multi-bag 91.5 sq. ft. (8.5 m ²)
Steering	Hydraulic on rear wheel	Hydraulic on rear wheel
Drive	Hydrostatic on rear wheel	Hydrostatic on rear wheel
Number of wheels	3	4

2.3.2. Nilfisk Advance RS 850 and RS851



Figure 6 Nilfisk Advance RS 850 [9]



Figure 7 Nilfisk Advance RS 851 [11]

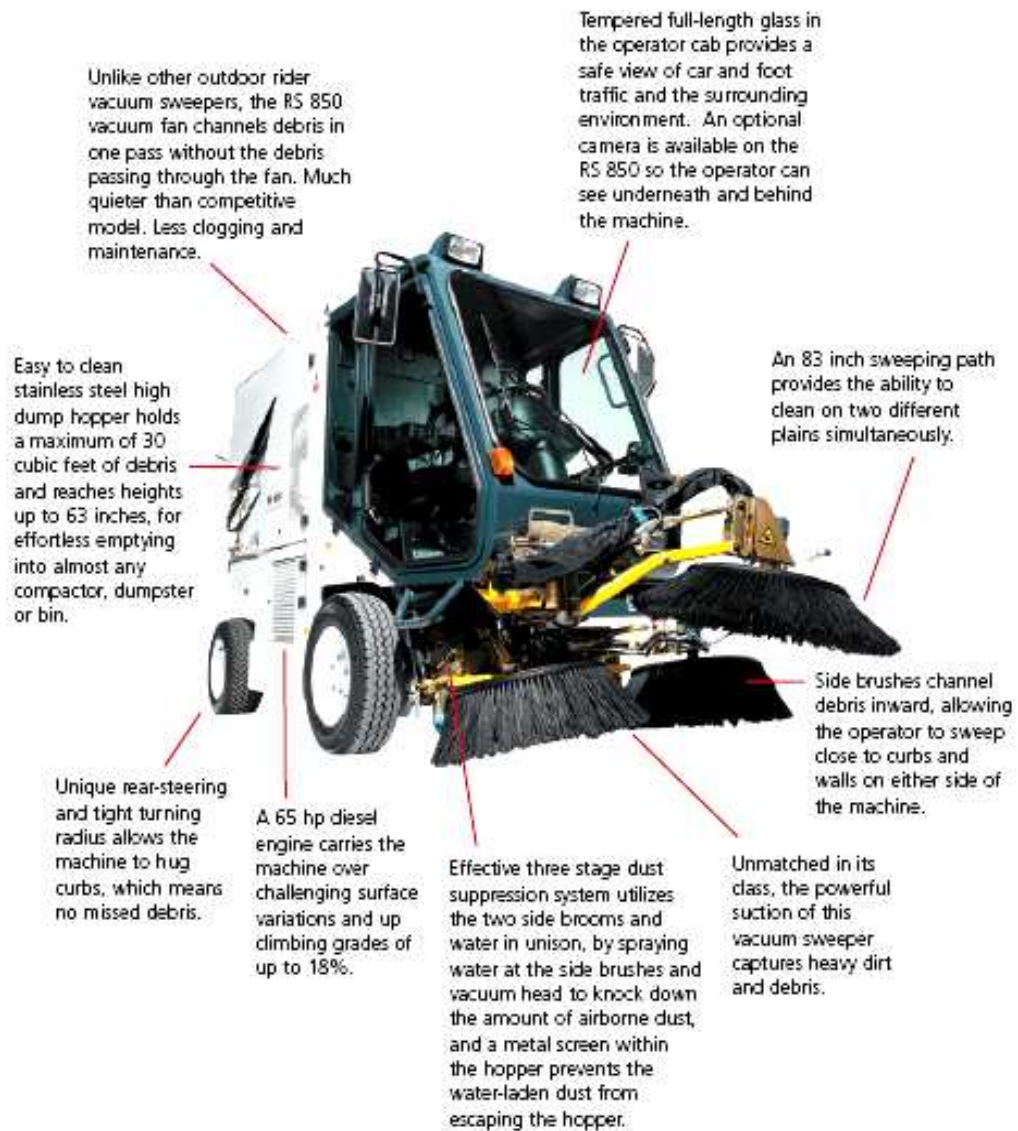


Figure 8 General Properties of Nilfisk Advance Sweeper [10]

Table 3 Technical Specifications of Nilfisk Sweepers [9], [10], [11]

Standard Sweeper	Nilfisk Advance RS 850	Nilfisk Advance RS 851
Width	1447.8 mm	1350 mm
Length	3324.8 mm	2850 mm
Height	2291 mm	2360 mm
Weight	2600 kg	2500 kg
Power Source	Diesel, 64.4 hp @2600 rpm, 3 cyl	Diesel, 48.6kW, 3cyl
Cleaning path with main broom	1600.2 mm	-
Cleaning path with 3rd Brush	2108.2 mm	1600 mm (2nd Side Broom)
Hopper Capacity	842 liters	850
Dust Suppression	Three Stage: water curtain at brushes and in pipe plus screens in the hopper	-
Suction system	Hydraulically driven Vacuum Fan located at the top of hopper	Hydraulically driven Vacuum Fan located at the top of hopper
Airflow	700 ft ³ /sec (19.8 m ³ /s)	2 m ³ /s
Suction Tube Diameter	7 in (177.8 mm)	
Steering	Rear-Power	-
Number of wheels	4	4
Side Broom Diameter	25.6 in (650.24 mm)	700 mm

Detailed information about the cleaning system (elevator and fan system), which is the main investigation topic of this study, is not available (such as dimension of fans or arrangement of components in the sweeper) due to the privacy policy of companies.

Throughout the survey it is seen that the outdoor sweeping technology, using vacuum sweeping, use centrifugal fans for vacuum. On the other hand, there is not enough information about the diameter and blade types of fans but the flow rate is changing in the range of (0.5-1.5 m³/s). An important fact is that, these values are for the sweepers less than 3500 mm length.

It is also seen that the elevator system is generally used for the longer and heavier outdoor sweeping vehicles. So, designing of an elevator system for new Müsan Sweeper (3540x1310x2295 mm and approximately 2000 kg) is a forerunner and unusual application in outdoor sweeping industry for the given dimensions of new Müsan Sweeper.

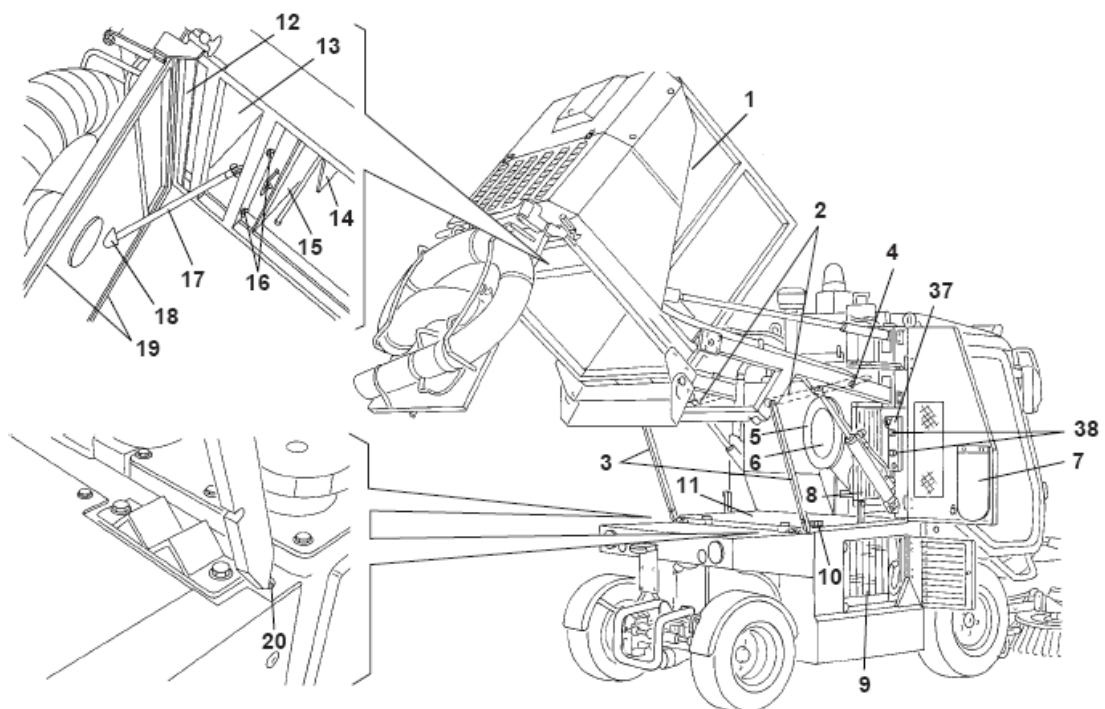


Figure 9 Placement of Some Componenets of Nilfisk Sweeper RS 850 [10]

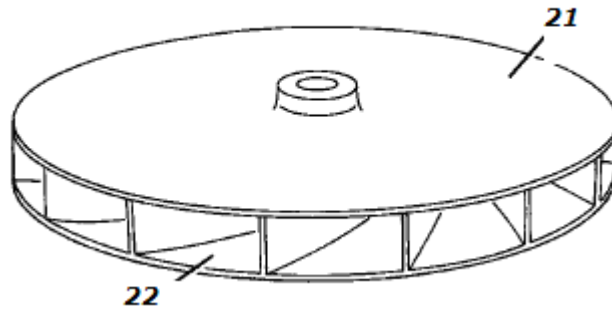


Figure 10 Centrifugal Fan of Nilfisk Sweeper RS 850 [10]

Table 4 Part List of given pictures in Figure 9 and Figure 10

No.	Part Name
1	Lifted and dumped hopper
2	Lifted hopper support rods (not applied)
3	Lifted hopper support rods (applied)
4	Lifted hopper support rod fasteners (not applied)
5	Suction hose gasket
6	Suction hose (from suction inlet to hopper)
7	Hopper front suction hole cover
8	Cab climate control condenser
9	Hydraulic system oil cooler
10	Fuel tank filler neck
11	Engine compartment panel
12	Inlet air breather filter
13	Suction fan compartment
14	Debris deflector
15	Dust and debris suction filter
16	Suction filter fasteners
17	Hopper lid (open) support rod
18	Hopper lid support rod housing
19	Suction sealing gasket
20	Hopper support rod housing
21	Centrifugal fan
22	Side sheet and blade type (Curved)
37	Front wheel lifting bracket
38	Bracket mounting knobs

2.3.3. Madvac CN100

Table 5 Technical Specifications of Madvac CN100 Sweepers [12]

Standard Sweeper	Madvac CN100
Height	1981 mm
Power Source	Kubota D1105T turbo, 3 cylinders, 1123 cc, Water cooled diesel. 31,5 HP (23,5 kW) @ 3000RPM
Sweeping Width	Variable from 44" (1,118 mm) to 89" (2,261 mm) with independent hydraulic right/left adjustable from cab. Adjustable down pressure and brush speed directly from cab
Hopper Capacity	1.15 cu. yd. (0.85 m ³) overall capacity Constructed of stainless steel, 60" (1,524 mm) dumping height
Dust Control	Three stage filtration with pleated panel filters for fine dust
Fan System	Patented flow through technology, vacuum compacts debris as it loads and permits vacuum without material passing through impeller fan.
Suction Tube Diameter	8 in (203 mm)
Drive	Front wheel drive with two wheel motors
Number of wheels	4

- 8-micron dust filtration system captures the finest particles to provide ultimate cleanliness for operator and pedestrians.
- Powerful variable vacuum cleans rough and delicate surfaces such as brick, cobblestones, bike paths and others.

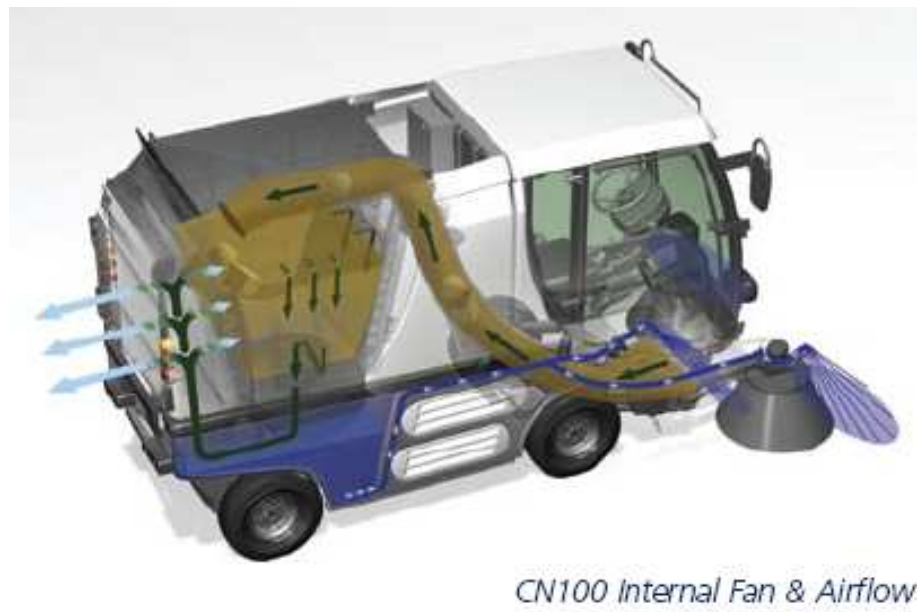


Figure 11 Internal Fan & Airflow System of Madvac CN100 [12]



Figure 12 Madvac CN100 [12]

2.3.4. Tennant Green Machine 636HS

Table 6 Technical Specifications of Green Machine 636HS [9]

Standard Sweeper	Tennant Gren Machine 636HS
Width	1140 mm
Length	3450 mm
Height	1980 mm
Weight	2800 kg
Power Source	Kubota 1505T Diesel 42 hp (31.4 kW)
Cleaning path (minimum)	1200 mm
Cleaning path (maximum) fully extended	2050 mm
Hopper Capacity	1000 liters
Dust Suppression	Side entry cyclone swirl plate, Stainless steel filter screen, Twin cyclones
Suction system	Vacuum impeller 12 blades, alloy steel
Drive	Infinitely variable hydrostatic 4-wheel
Number of wheels	4
Broom Diameters	750 mm

The 636 differs from smaller Green Machine models in that its suction performance comes from a large diameter remotely-mounted clean fan. This "pure vacuum" system is better suited to coarser debris such as construction dirt or heavier accumulations of grit or sand along curb-lines.

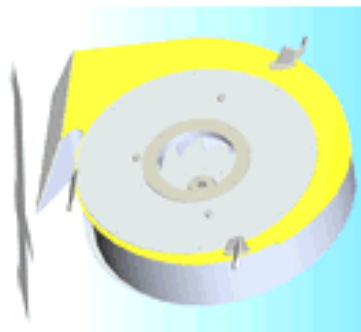


Figure 13 Centrifugal Fan of 636HS [13]



Figure 14 Green Machine 636HS during discharging [13]



Figure 15 Tennant Green Machines 636HS [13]

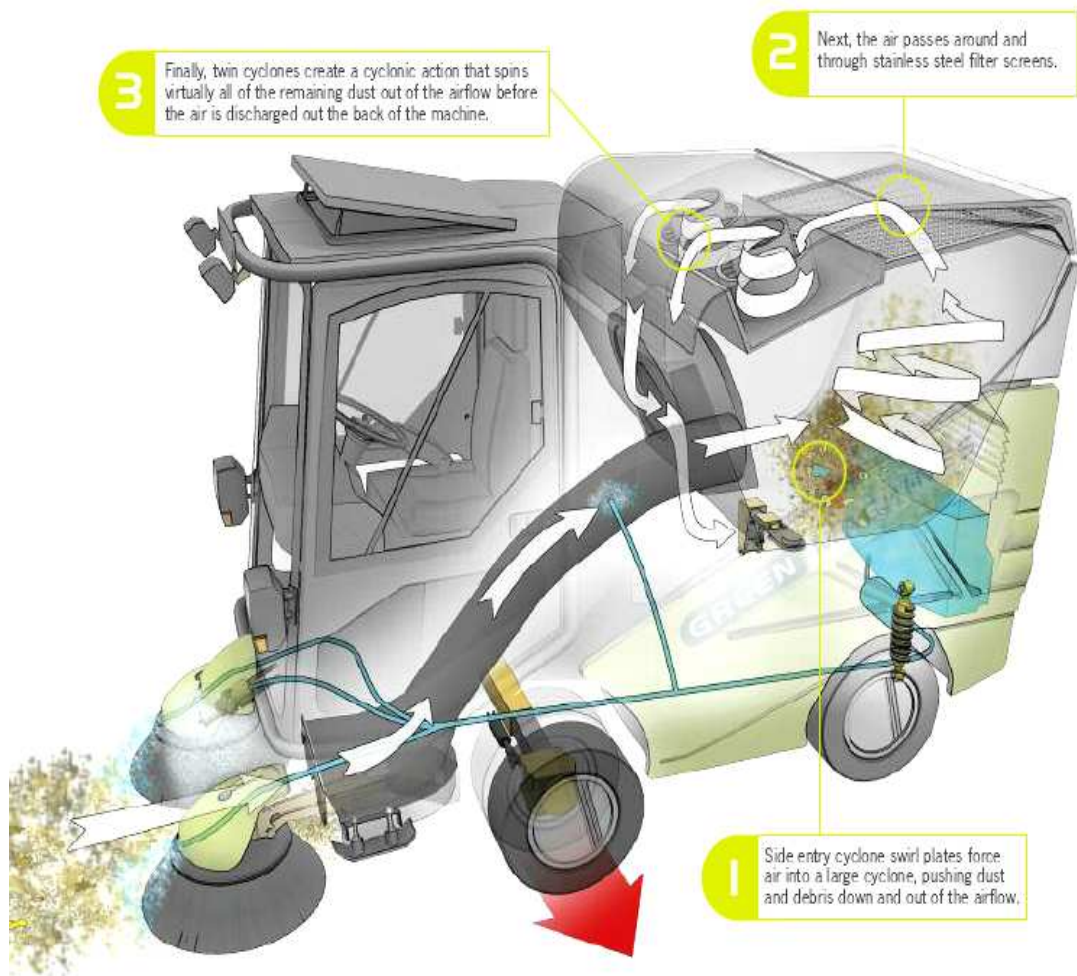


Figure 16 Vacuum System of Tennant Green Machines 636HS [13]

In Table 1 and Table 2, former MÜSAN VSM 060 sweeper is compared with the other sweepers in the market according to their technical specifications. The new sweeper can only be compared according to the structural and mechanical specifications of it. But, by the completion of the manufacturing of the prototype vehicle, and after the real-time experiments, exact and reliable results will be available and comparison is going to be done.

New VSM 080 has some advantages due to utilizing electro-hydraulic propulsion system. The turning radius of the vehicle is zero as the sweeper is capable of making sideward movement by the aid of hydraulic cylinders which are assembled in between front and back wheel pairs. The operation speed of the vehicle is approximately 15 km/h and it can climb to a terrain which has gradient of 26 % with this speed.

CHAPTER 3

OBJECT OF THE PRESENT INVESTIGATION

Combining all of systems (broom, fan and elevator) together effectively, efficient and powerful cleaning system of a vacuum sweeper is acquired. Today, customer demands are rising on sweeper vehicles generating less dust particles, less air pollution (health), less sound and optimum costs. Sweeper manufacturers are working on the 'Cleaning System' especially to gain active role on market and also to supply customer demands.

Street sweepers do much more than just keep things clean. They are used to pick up trash, dirt, chemicals, motor oil and more that can contaminate the environment. If these potentially harmful substances are removed from the street, the construction site, or the parking lot before they reach storm drains, they are less likely to end up in rivers and lakes.

The United States government has already taken steps to combat this type of pollution. In 1987, the Environmental Protection Agency (EPA) enacted the PM-10 standard, which regulates the number of particles in the air that are 10 microns (.001 centimeters, or .0004 inches) or smaller. These particles are small enough to affect our health because of their ability to reach the lower regions of our lungs. Most street sweeper manufacturers have several machines that meet PM-10 removal standards, keeping our streets cleaner and our lungs clearer. Street sweeper makers need to stay on their toes, though the EPA is considering lowering the standard to PM-2.5, or regulating particle matter measuring 2.5 microns (.0001 inches) or less. [2]

The purpose of this thesis is to develop a vacuum sweeper having an efficient cleaning system with low noise level, less air pollution, low costs and aesthetic design surely.

Throughout the SAN-TEZ research and development project, the following subjects were investigated and following subjects got start for programming and manufacturing:

- Redesign of VSM 060 vacuum sweeper with aesthetic approach.
- Redesign of traction and motion control system

And for the cleaning system, an elevator system was designed and manufacturing process completed before the tests of whole cleaning system. Following subjects related with the Vacuum (Fan) System were also investigated and some of them are continuing:

- CFD (Computational Fluid Dynamic) analysis of fans (present and new design)
- Designing and manufacturing of fan test set-up.
- Determining performance results of former fans (diameter of 250 mm, straight and curved type blades) which are already being used by Müsan A.Ş.
- Comparing former straight and curved blade fans
- Designing and manufacturing of a new fan and testing of it (diameter of 400 mm, straight type blade).

SAN-TEZ project group members comprise of METU and MÜSAN A.Ş. (Makina Üretim Sanayi) and other project members and responsibilities are as follows:

Punto Mühendislik Ltd. Şti. is responsible for the computational fluid dynamics (CFD) analyses of the cleaning system and system components of the former and new vehicle.

PEKAS is responsible for all electrical subsystems, driver and firmware designs. Component procurements and integration of electrical systems to the vehicle will also be done by the firm.

MÜSAN A.Ş. is responsible for the detailed design of some subsystems and components, drafting for the designed parts and manufacturing.

Design requirements for the new sweeper can be listed as follows:

- Length of elevator will be 1200 mm (difference between pulley axes)
- Total length of elevator will be 1610 mm with the (protective sheets)
- Elevator System dimensions will be approximately (1610x1170x400 mm)
- For the vacuum system, during the tests, it was proved that the centrifugal fan with straight blades is more efficient than the curved one. Therefore, a straight blade centrifugal fan will be used due to its easy manufacturing as well.
- Straight blade fans (non-radial, backward) with 400 mm diameter and 50 mm width were manufactured.
- Fan diameter will be 400 mm and width of fan is 50 mm. Expected flow rate is about 1 m³/s. according to power consumption results of fan (individual and system test will be performed), two centrifugal fans may be used.
- Expected power consumption is approximately in the range of 2-2.5 kW for one fan.

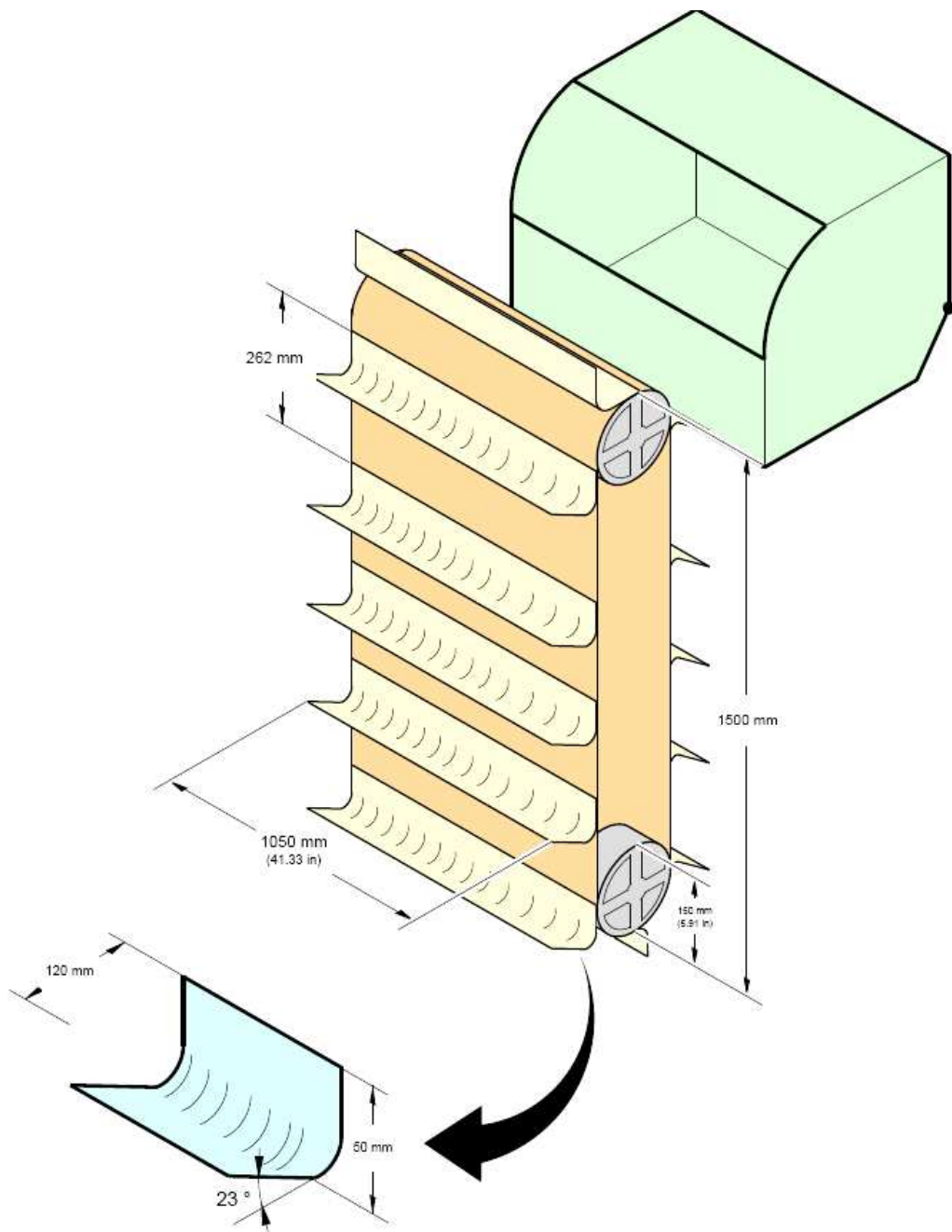


Figure 17 An Illustration of Elevator System

CHAPTER 4

DESIGN AND MANUFACTURING OF THE CLEANING SYSTEM

4.1. Elevator System

4.1.1. Applications of Elevator Systems

Elevator systems are used to transport products such as foodstuff in food production widely and it is also used to elevate other materials such as heavy iron parts, coal pieces in mines or cement dusts. These systems have also a wide usage area in 'Cleaning Operations' to elevate garbage on streets or in some other places. Nowadays, 'Street Sweepers' are widely using this system.

Street sweepers are heavy duty cleaners, available as either ride-on or walk-behind models. While the ride-on models are easiest on the user and require little energy to use, the walk-behind street sweepers provide a great alternative for those with a smaller street sweeper budget. [13]

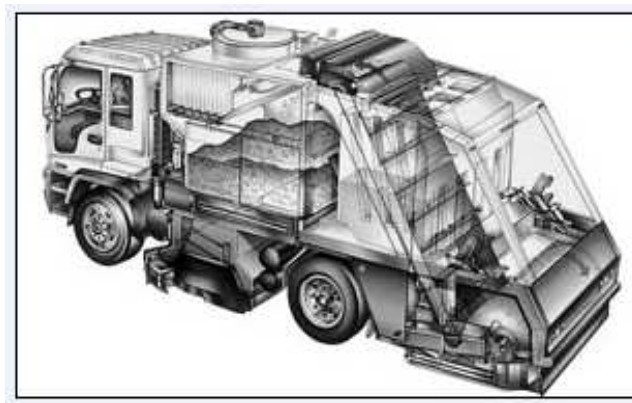


Figure 18 Inclined Elevator Conveyor of Centurion Sweeper [14]



Figure 19 Elevator Conveyor Parts [14]

- 1-Sweeper
- 2- Mounting Frame
- 3- Front Wheel
- 4-Back Wheel
- 5- Rotary brush
- 6- Sweeping Brush
- 7-Rubbish
- 8-Picked (lifted) rubbish area
- 9-Elevating Bucket (Blade)

The elongated pick-up rotary brush, extending transversally, adjacent to the lower end of the elevating conveyor for sweeping the rubbishes on the ground and projecting them into transporting elements of the elevating conveyor.

4.1.2. Theory and Calculations

In elevator design, during the calculations, all available data is used and linear velocity, diameter of pulley, average density of material, capacity of per bucket, distance between pulleys (distance mass that will be raised), bucket dimensions (length, width and depth), distance between each buckets are able to be used as inputs and eventually; rotation, polar distance, discharge type [centrifugal, gravitational or both], maximum static stress, torque and power can be obtained.

Therefore, necessary inputs are these;

- Linear Velocity [m/s] $v = 1$
- Diameter of Pulley [m] $r_b = 0.15$
- Average Density of Material [g/cm³] $\rho = 1.2$
- Capacity of per Bucket [l] $i_0 = 10$
- Distance between pulleys [m] $H = 1.5$
- Bucket Dimensions (Length, Width and Depth) [mm]
 $(B, A, h) = (1050, 120, 50)$
- Distance between each Buckets [mm] $a = 262$

There are basically three types of buckets used in elevators: Deep, Shallow and V-type buckets. Theoretically, dimensions of these types of buckets are determined according to the DIN standards.

In many applications, buckets can be designed according to the customer requirements. Therefore, unlike the buckets given in standards (type and dimension limits) shape of buckets of the elevator system is designed to obtain efficient discharging. [16]

Dimensions and determining the motion type of system is the effective process for an 'Elevator Design'. It is necessary to specify "I-POLAR DISTANCE" for determining the type of motion (Discharging Type).: Centrifuge, Gravitational or both.

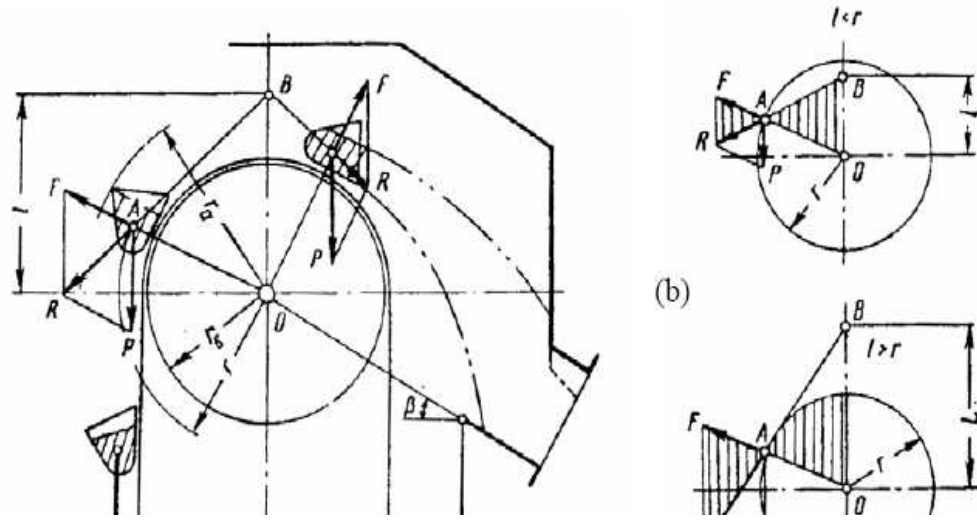


Figure 20 Determining the Polar Distance (l) [16]

F: Centrifugal Force, **P:** Loading, **g:** Acceleration of gravity

$$P = m \times g$$

m: Total mass of full bucket

v: Linear Velocity of Load

$$F = m \frac{v^2}{r}$$

$$\frac{l}{r} = \frac{P}{F} = \frac{mg}{m \frac{v^2}{r}}$$

Linear velocity (v) of the elevator is taken 1 m/s due to prevent gravitational discharging of buckets. Centrifugal or 'Centrifugal-Gravitational' discharging is required. [16]

As it is seen in the Figure 20:

r_b : Pulley radius (75 mm)

$r = 2r_b + A/2$ (where A is width of blade)

r : Rotation radius (210 mm)

r_a : Outside radius (270 mm)

$$r_a = 2r_b + A$$

As a result: If $l < r_b$ discharging is by centrifuge, If $l > r_a$ discharging is by gravitational and if $r_b < l \leq r_a$ then it is also proved by experimentally that discharging of pieces in the bucket will be discharged by centrifugal and gravitational motion.

Rotation per Minute:

$$n = \frac{30 \times v}{\pi \times r_a} [rpm] \text{ where } r_a \text{ is outside radius}$$

Since $r_a = 270$ mm and $v = 1$ m/s

$$n = 73.14 \text{ rpm}$$

Polar Distance:

$$\frac{I}{r} = \frac{P}{F} \Rightarrow$$

$$I = \frac{gr^2 30^2}{\pi^2 r^2 n^2} = \frac{895}{n^2}$$

$$I = 167.3 \text{ mm}$$

Since $r_b < I = 167.3 \text{ mm} < r_a$, pieces in the blade will be discharged by centrifugal-gravitational motion.

Carrying Capacity:

Q (carrying capacity) is found from the equation given below by using the capacity of per bucket and distance between each buckets. (ρ is average density of materials and ψ is loading coefficient can be taken either 0.75 or 0.85 in common)

$$\frac{i_0}{a} = \frac{Q}{3.6v\rho\psi} [l / m]$$

Where;

Average density [g/cm^3] $\rho = 1.2$

Capacity of per blade [l] (approximately blade volume) $i_0 = 10$

Distance between each buckets [mm] $a = 262$

$$\Psi = 0.85$$

After that, Q (carrying capacity) is found as $Q = 123.6 \text{ l/h}$

Later, q_0 (unit mass of belt and buckets) and q (unit mass of elevator) values are calculated using the formula given below where K_2 is the resistance coefficient of the Bucket-Belt (or chain) intersection and m is the mass of overall buckets where ρ is density and i_0 capacity of per bucket ;

$$q_0 = K_2 \times Q$$

K_2 is taken 0.50 from the Table 7.

$$q_0 = 61.8 \text{ kg/m}$$

Table 7 Resistance and Manufacturing Coefficients [16]

$Q [\text{ t/h }]$	ELEVATOR TYPE					
	Belt		Single Chain		Double Chain	
	Bucket Types					
	Deep and Shallow	V- type	Deep& Shallow	V-type	Deep& Shallow	V-type
	K_2					
<10	0.6	-	1.1	-	-	-
10-25	0.5	-	0.8	1.1	1.2	-
25-50	0.45	0.6	0.6	0.85	1.0	-
50-100	0.4	0.55	0.5	0.7	0.8	1.1
>100	0.35	0.5	-	-	0.6	0.9
K_1	2.5	2.0	1.5	1.25	1.5	1.25
K_3	1.6	1.1	1.3	0.8	1.3	0.8

l : Length of belt can be found from the formula shown below:

$$l = 2H + 1.143 \times r_b / 1000$$

$$l \cong 3 \text{ m}$$

t : Blade quantity can be found as follows:

$$t = 1000 \times l / a \text{ (Where } a \text{ is the distance between each blades and } a = 262 \text{ mm)}$$

$$t \cong 12$$

Mass of overall blades:

$$m = \rho \times i_0 \times t \text{ (where } t=11 \text{ and } i_0=10)$$

$$m=145.2 \text{ kg/m}$$

Unit mass of the elevator is calculated by assuming each blade is full.

q (Unit mass of elevator) is found by $q=q_0+m$

$$q=207 \text{ kg/m}$$

The calculation is continued with maximum static stress. Empirical formulas are generally used for the calculations of elevator parameters. These equations give approximate results instead of exact due to lifting main resistance loadings in elevators. For vertical elevators, maximum static stress of tensile component is given as:

$$S_{\max} = 1.15 \times H \times (q + K_1 q_0) \text{ [Mpa]}$$

Since given value of K_1 in Table 7 is maximum, K_1 can be taken as 1.5

$$S_{\max} = 380 \text{ Mpa}$$

Required power on the rotating shaft of vertical elevators (neglecting the losses of working mechanism) can be determined by the equation given below:

$$N_0 \cong \frac{1.15QH}{367} + \frac{K_3 q_0 H v}{367} = \frac{QH}{367} (1.15 + K_2 K_3 v) \text{ [kW]}$$

Required power for lifting the charged mass (include a safety factor of 1.15) is determined from the first part of equation and friction resistance of dynamic component is determined from the second part of equation as shown above. Approximately K_1 and K_2 coefficients are taken from Table 7.

$$N_0 \cong 0.84 \text{ kW}$$

Overall design results can be determined as shown below:

Table 8 Overall Results of Elevator System

ELEVATOR DESIGN RESULTS			
<u>INPUTS:</u>			
Bucket Type	=	V-type (BLADE)	
Linear Velocity	=	1	m/s
Capacity of per Bucket	=	10	lt
Width of Bucket	=	120	mm
Depth of Bucket	=	50	mm
Length of Bucket	=	1050	mm
Distance between Each Bucket	=	262	mm
Diameter of Pulley	=	150	mm
Distance between Pulleys	=	1.5	m
Average Density	=	1.2	g/cm ³
<u>OUTPUTS:</u>			
DISCHARGING TYPE	=	CENTRIFUGE+GRAVITATIONAL	
Polar Distance	=	167.31	mm
Belt Length	=	3.17	m
Bucket Quantity	=	12	
Carrying Capacity	=	123.66	l/h
Unit Mass of Belt and Bucket	=	61.83	kg/m
Unit Mass of Elevator	=	207.09	kg/m
Rotation	=	73.14	rpm
Maximum Static Stress	=	380.37	MPa
Torque	=	112.24	Nm
Power	=	0.84	kW

4.1.3. Elevator System Design

As shown in the Figure 21, vacuum system, shown in red color, will be positioned on the sweeper and elevator system will be settled at the middle of the sweeper.

The main aim of the system is lifting the debris such stones, cans and bottles. Elevator system can also be named 'continuum system' in consideration of it has many blades mounted on a belt-pulley system which are rotating continuously.

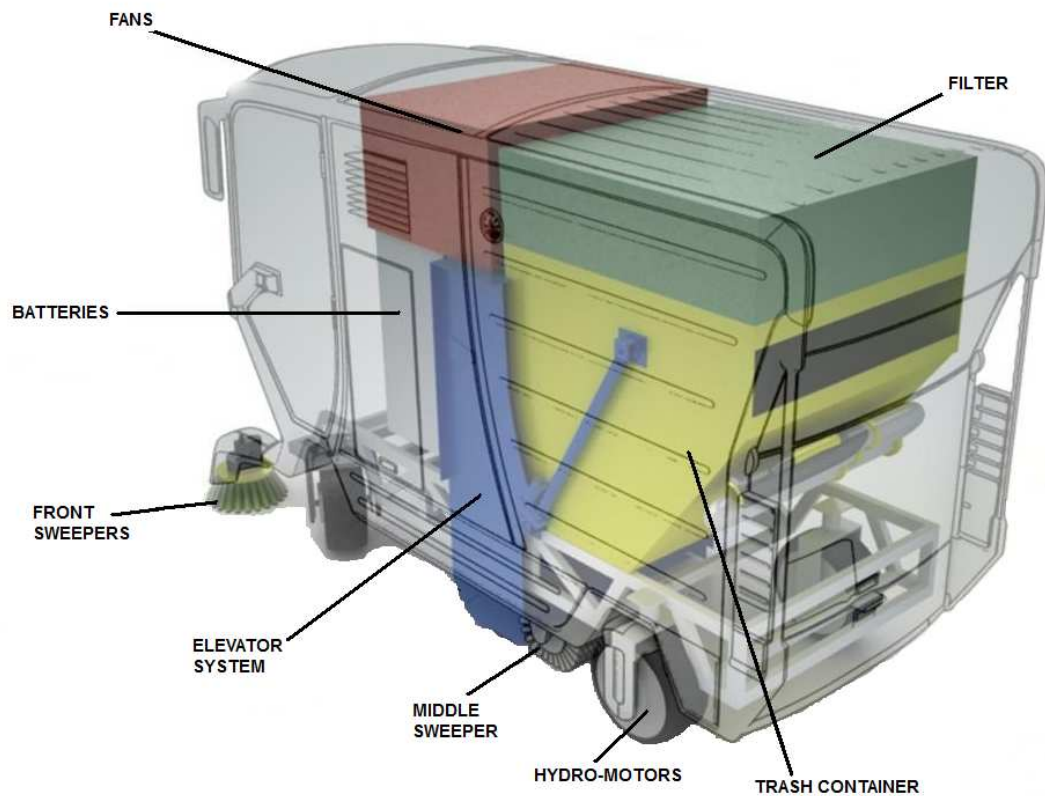


Figure 21 Locations of Sub-Assemblies on Sweeper

Details of the elevator system are:

- Length of elevator will be 1200 mm (difference between pulley axes)
- Total length of elevator will be 1610 mm with the (protective sheets)
- Elevator system dimensions will be approximately (1610x1170x400 mm)
- There will be 11 blades (bucket) on elevator system with 1050 mm width and 262 mm distance every two blades.
- Linear velocity of the elevator system is 1-1.25 m/s and for rotation power, electric motors (2x440 W) were used. After the experiments, hydraulic motor (BR 160) will be preferred.
- For the bumpy roads, sliding mechanism was designed. Hydraulic piston will be used to hold the system.

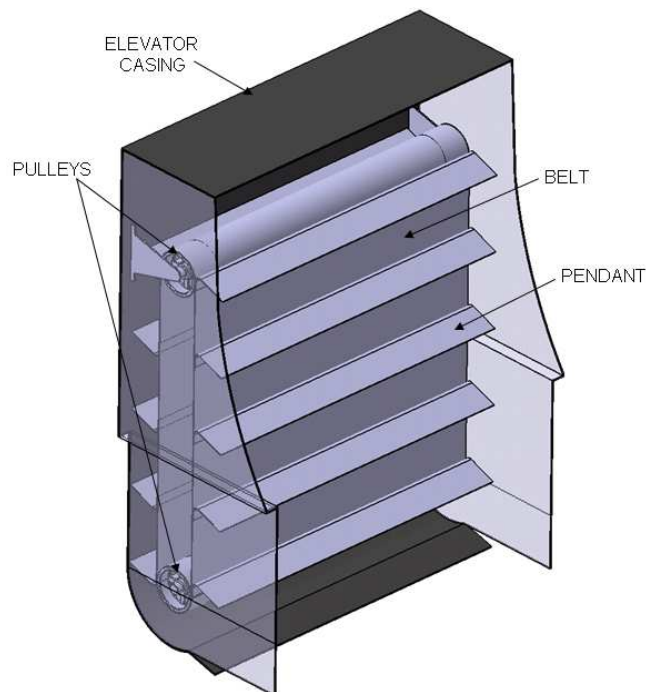


Figure 22 Initial Design of Elevator System

As given in the theory and calculation section, diameter of pulley and linear velocity of the elevator system are the critical inputs due to acting on the discharging type. By taking the diameter of pulley 150 mm and linear velocity of 1 m/s, all of the detailed outputs are obtained such number of blades, overall dimensions of the system and electric motor to be used. Other detailed outputs for the design can be reached in previous section.

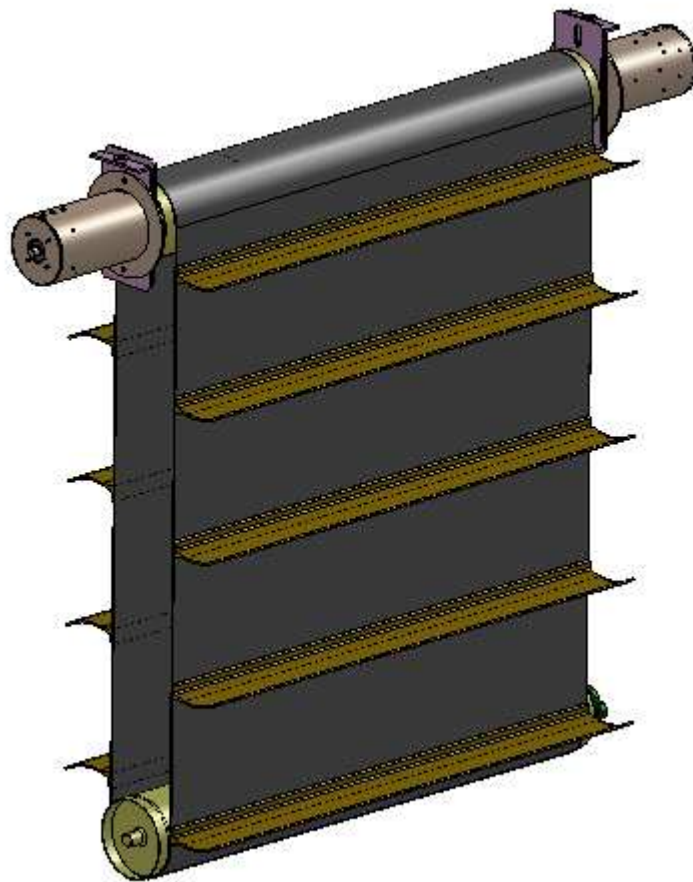


Figure 23 Second Design of Elevator System with Electric Motors

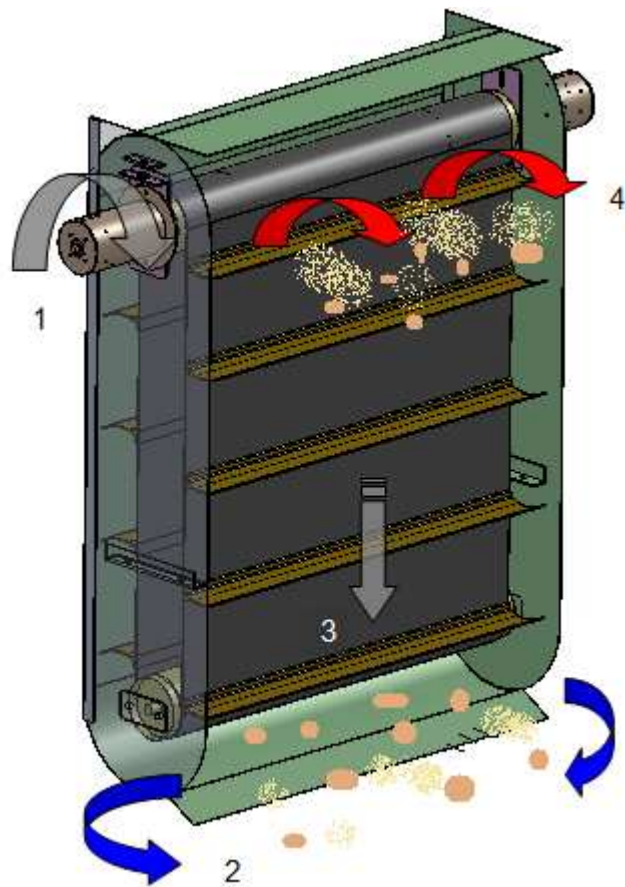


Figure 24 Operation principle of the Elevator System

- 1- Electric motors supply power for 50-60 rpm.
- 2- Gutter brooms direct the debris on the path towards the main broom and the main broom impels the garbage towards the system.
- 3- Elevator blades collect the garbage.
- 4- After one period of the system, blades discharge the garbage. And fan system vacuum the dust due to discharging.

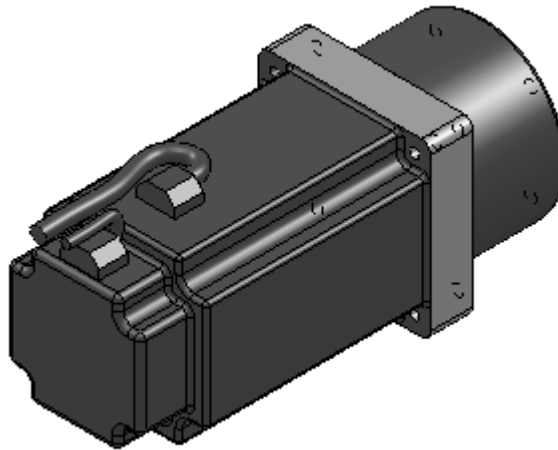


Figure 25 Previous Electric Motor for Elevator System (440W DC)

After changing the overall system from electrical to electro-hydraulic, hydraulic motor is preferred instead of the electric one as given below:



Figure 26 Hydraulic Motor (BR 160) for the Elevator System

4.1.4. Manufacturing of the Elevator System

As it can be seen from the following Figures, to prevent the dispersing of the trashes and dust, protective side sheets are manufactured from the Aluminum material. Then, pulley-band system and blades assembled onto the band are manufactured and assembled to the inside of the Al protective sheets.



Figure 27 Assembly of the Protective Side Sheets to the Chassis

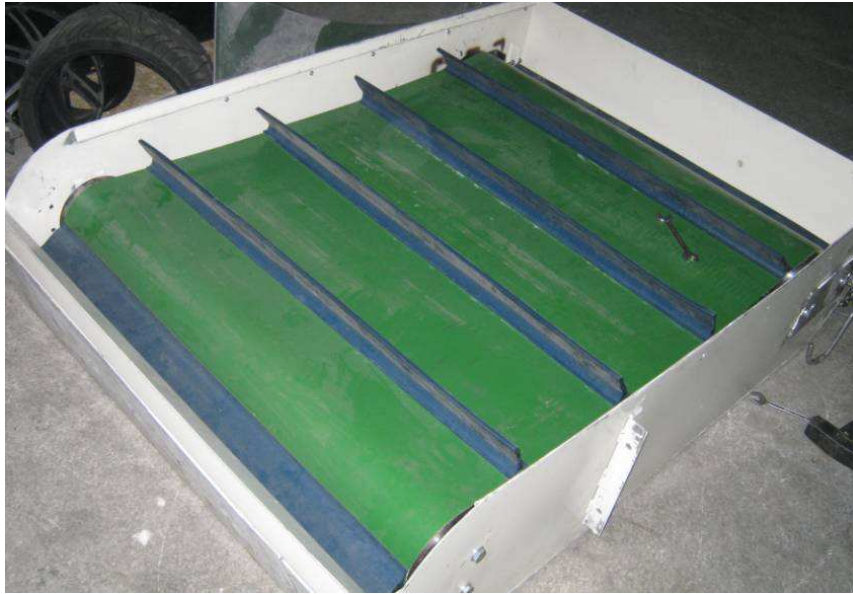


Figure 28 Elevator System and Band & Blades



Figure 29 Assembly of Hydraulic Motor to System

4.2. Vacuum System Development and Design

4.2.1. Fan Selection and Centrifugal Fans

As it has been presented in Chapter 2, most of the sweeper manufactures have been using centrifugal fans in their designs. Fans should be efficiently used and correctly selected to satisfy the conditions. Therefore, fan selection is a complex process that starts with a basic knowledge of system operating requirements and conditions such as airflow rates, temperatures, pressures, airstream properties. The variability of these factors and other considerations, such as cost, efficiency, operating life, maintenance, speed, material type, space constraints, drive arrangements, temperature, and range of operating conditions, complicate fan selection. [16]

Centrifugal fans have an entirely different design. Instead of passing straight through, the air makes a 90-degree-angle turn as it travels from the inlet to the outlet and is "thrown" from the blade tips. Centrifugal fans have more mass farther from the axle, which requires more starting torque, but they are generally quieter than axial fans.

There are several arrangements of fan blades for centrifugal impellers. The highest-efficiency centrifugal fans use airfoil or backward-curved impeller blades. Airfoil blades are curved backward and have an airfoil shape (similar to a cross section of an airplane wing), while backward-curved blades are of a single thickness of metal. Straight radial fan blades are used mostly in industrial applications. The main advantage of radial blades is that they permit the passage of foreign objects in the airstream such as sawdust, metal filings, and other debris.

Forward-curved fan blades have low efficiency and are typically used to move high volume against low pressure in applications such as window air conditioners and hotel unitary packages. Low purchase cost and compactness are the principal advantages of fans with forward-curved blades; they are still being built and installed in great quantities. [18]

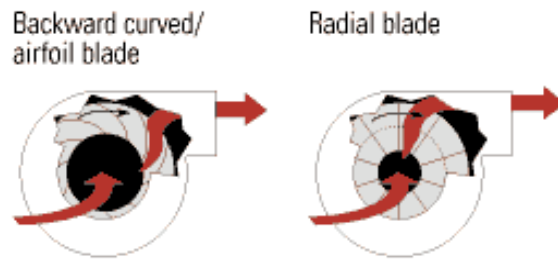


Figure 30 Centrifugal Fan Blades [18], [19]

Industrial vacuum sweepers use radial type centrifugal fans commonly due to high speeds, low volumes, and high pressures. MÜSAN Company has two fans (curved and straight type blade) with diameter of 250 mm and 8 blades. The former sweeper (VSM 06) is using backward curved one. These fans are compared and detailed results are given in next chapters.

As it has already been proved by experiments and presented in Chapter 5, former straight blade fan provides better results with respect to the former curved one. Fan(s), which is considered to be used at new Müsan Sweeper (VSM 08), is 'straight' type and non-radial.

4.2.2. Characteristics of Former Müsan Fans

In fan design, although the best results are obtained experimentally; using some inputs such inlet and outlet radius, inlet and outlet blade angles, width of the fan, outputs can be obtained for different speeds of the fan such as estimated optimum blade quantity and tip speed of blades.

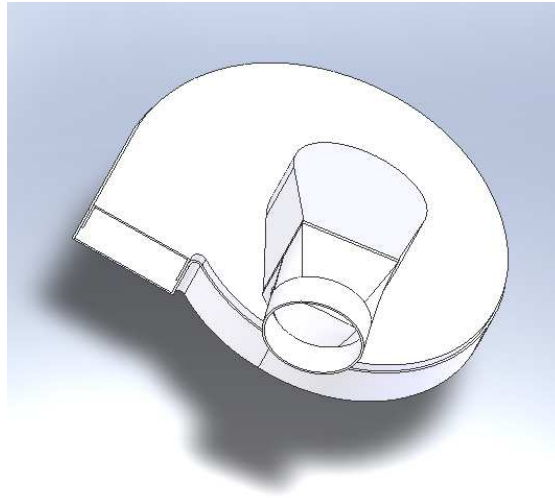


Figure 31 Centrifugal Fan of Former Müsan Sweeper

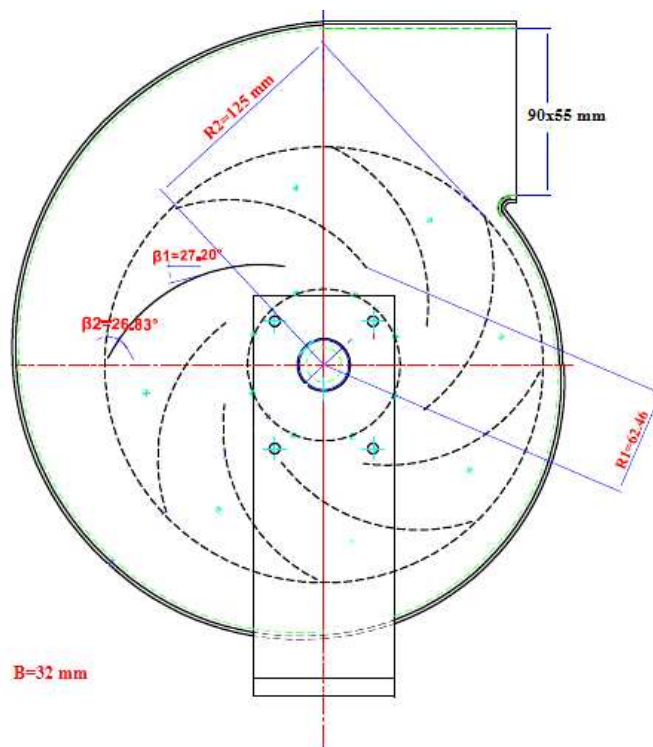


Figure 32 2D View of the Former Fan

4.2.3. Velocity Vector Diagrams and Triangles

The main purpose of determining the available fan performance results is that, these results will be evaluated for development process of the new fan(s). Using the velocity triangles and inputs given in Table 8; tip speeds, pressure difference and power are calculated.

Former MÜSAN fan (Figure 33, backward and 8 Blades) is approximately driven at 3078, 4926 and 6198 rpm. Therefore, velocity vectors will be determined initially and then pressure differences and required power to drive fan will be calculated for different rotations in next pages.

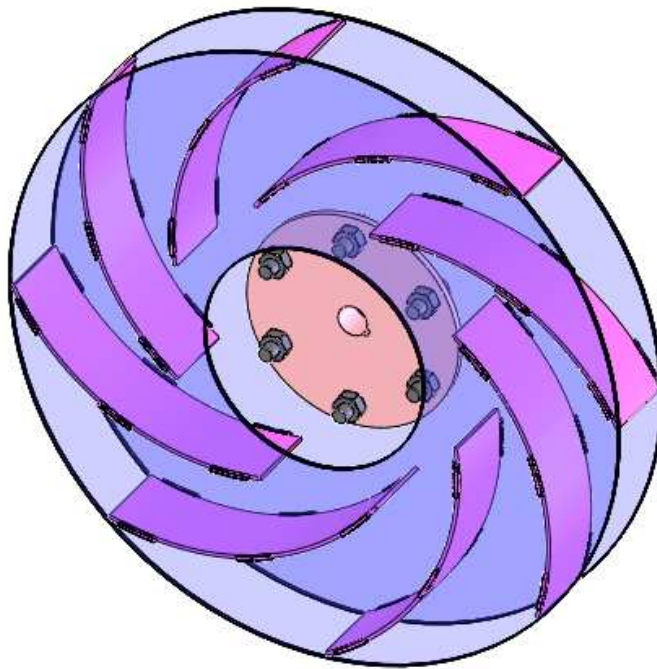


Figure 33 Former Rotor (to be drawn velocity triangles)

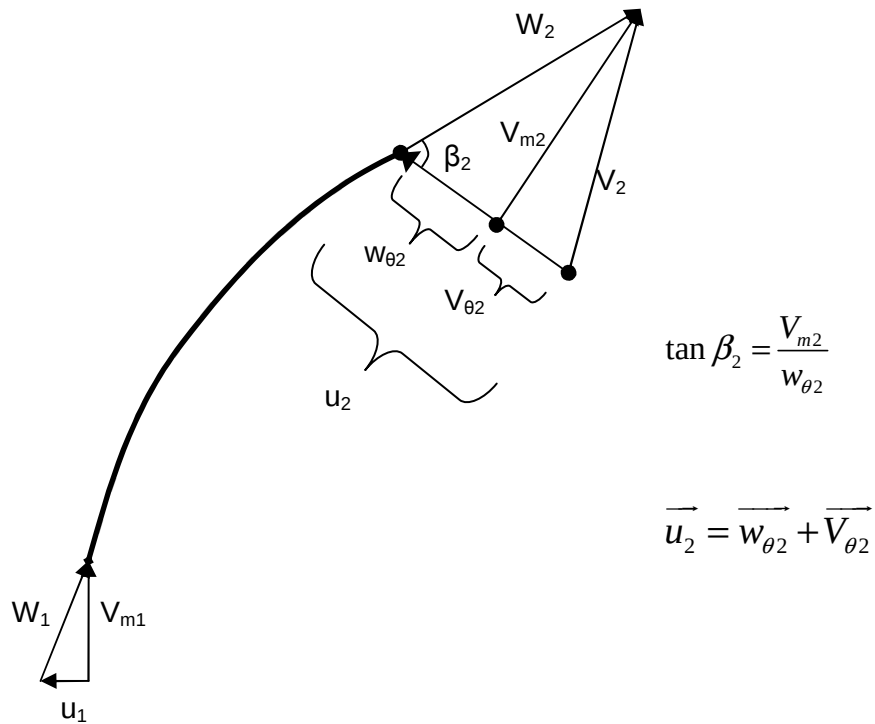


Figure 34 Velocity Triangles on Blade Tips

Table 9 Properties of Former Fan Blade

BLADE INLET ANGLE (deg)	β_1	29.5
BLADE OUTLET ANGLE (deg)	β_2	26
INLET RADIUS (mm)	R_1	63.5
OUTLET RADIUS (mm)	R_2	122.8
WIDTH (mm)	B	32

Physical meanings of velocity vectors (W_2 , V_{m2} , u_2 , $V_{\theta 2}$ and $w_{\theta 2}$) has been given in Figure 34. u_2 is the peripheral velocity, W is the relative velocity, V_m is the meridional velocity and V_2 is the absolute velocity.

$$N_1 = 3078 \text{ rpm}$$

Flow rate is also experimentally verified that $Q = 0.076 \text{ m}^3/\text{s}$ @ 3078 rpm,

Required inputs are taken from the Table 9,

$$V_{m2} = \frac{Q}{2\pi R_2 B}$$

$$V_{m2} = \frac{0.076}{6.28 \times 0.122 \times 0.032} \Rightarrow V_{m2} = 3.08 \text{ m/s}$$

$$u_2 = W_2 \times R_2 = \left(\frac{2\pi N}{60} \right) \times R_2 \Rightarrow u_2 = 39.57 \text{ m/s}$$

$$\tan \beta_2 = \frac{V_{m2}}{w_{\theta 2}} \Rightarrow \tan 26^\circ = 0.48 = \frac{V_{m2}}{w_{\theta 2}}$$

$$w_{\theta 2} = 6.31 \text{ m/s and since } \vec{u}_2 = \vec{w}_{\theta 2} + \vec{V}_{\theta 2}$$

$$V_{\theta 2} = 33.26 \text{ m/s}$$

$$H = \frac{1}{g} \times u_2 \times V_{\theta 2} = \frac{1}{9.81} \times 39.57 \times 33.26 \Rightarrow H = 134.16 \text{ m}$$

$$\Delta P = \rho g H = 1.2 \times 9.81 \times 134.16 \Rightarrow \Delta P = 1.58 \text{ kPa}$$

$$P_f = \Delta P \times Q = 1.58 \times 0.076 \Rightarrow P_f = 0.12 \text{ kW}$$

$$N_2 = 4926 \text{ rpm}$$

Flow rate is determined for 4926 rpm by using the 'Similarity Equation' which is given below:

$$\frac{Q_2}{Q_1} = \left(\frac{N_2}{N_1} \right) \left(\frac{D_2}{D_1} \right)^3 \text{ (for geometrically similar machines)}$$

For the Müsan VSM 06 Sweeper, Fan diameter $D_1=D_2=250$ mm (similarity for the same fan rotor $D_1/D_2=1$) and $Q_1 = 0.076 \text{ m}^3/\text{s}$, Therefore $Q = Q_2 = 0.116 \text{ m}^3/\text{s}$ at 4926 rpm and other required inputs are taken from the Table 9,

$$V_{m2} = \frac{Q}{2\pi R_2 B}$$

$$V_{m2} = \frac{0.116}{6.28 \times 0.122 \times 0.032} \Rightarrow V_{m2} = 4.7 \text{ m/s}$$

$$u_2 = W_2 \times R_2 = \left(\frac{2\pi N}{60} \right) \times R_2 \Rightarrow u_2 = 63.3 \text{ m/s}$$

$$\tan \beta_2 = \frac{V_{m2}}{w_{\theta 2}} \Rightarrow \tan 26^\circ = 0.48 = \frac{V_{m2}}{w_{\theta 2}}$$

$$w_{\theta 2} = 9.65 \text{ m/s and since } \vec{u}_2 = \vec{w}_{\theta 2} + \vec{V}_{\theta 2}$$

$$V_{\theta 2} = 53.65 \text{ m/s}$$

$$H = \frac{1}{g} \times u_2 \times V_{\theta 2} = \frac{1}{9.81} \times 63.3 \times 53.65 \Rightarrow H = 346.18 \text{ m}$$

$$\Delta P = \rho g H = 1.2 \times 9.81 \times 346.18 \Rightarrow \Delta P = 4.07 \text{ kPa}$$

$$P_f = \Delta P \times Q = 4.07 \times 0.116 \Rightarrow P_f = 0.473 \text{ kW}$$

$$N_3 = 6198 \text{ rpm}$$

Flow rate is determined for 6198 rpm by using the 'Similarity Equation' which is given below:

$$\frac{Q_2}{Q_1} = \left(\frac{N_2}{N_1} \right) \left(\frac{D_2}{D_1} \right)^3 \text{ (for geometrically similar machines)}$$

Therefore, $Q = Q_3 = 0.152 \text{ m}^3/\text{s}$ at 6198 rpm and other required inputs are taken from the Table 9,

$$V_{m2} = \frac{Q}{2\pi R_2 B}$$

$$V_{m2} = \frac{0.152}{6.28 \times 0.122 \times 0.032} \Rightarrow V_{m2} = 6.16 \text{ m/s}$$

$$u_2 = W_2 \times R_2 = \left(\frac{2\pi N}{60} \right) \times R_2 \Rightarrow u_2 = 79.72 \text{ m/s}$$

$$\tan \beta_2 = \frac{V_{m2}}{w_{\theta 2}} \Rightarrow \tan 26^\circ = 0.48 = \frac{V_{m2}}{w_{\theta 2}}$$

$$w_{\theta 2} = 12.64 \text{ m/s and since } \vec{u}_2 = \vec{w}_{\theta 2} + \vec{V}_{\theta 2}$$

$$V_{\theta 2} = 67.08 \text{ m/s}$$

$$H = \frac{1}{g} \times u_2 \times V_{\theta 2} = \frac{1}{9.81} \times 79.72 \times 67.08 \Rightarrow H = 545 \text{ m}$$

$$\Delta P = \rho g H = 1.2 \times 9.81 \times 545.12 \Rightarrow \Delta P = 6.42 \text{ kPa}$$

$$P_f = \Delta P \times Q = 6.42 \times 0.152 \Rightarrow P_f = 0.975 \text{ kW}$$

Using the same equations, blade speeds of the former fans (curved and straight type blades) can be calculated. After the initial results, straight type fan provides higher speeds as shown the table given below.

Table 10 Overall Results of Former Fans

BLADE SPEEDS OF CURVED BLADE FAN (m/s)					
ROTATION rpm	V _{m2}	w _θ	u ₂	V _θ	H (m)
3098	3.03	6.21	40.53	34.323	141.8
4926	4.62	9.47	64.45	54.98	361.2
6198	6.05	12.41	81.10	68.67	567.7
BLADE SPEEDS OF STRAIGHT BLADE FAN (m/s)					
ROTATION rpm	V _{m2}	w _θ	u ₂	V _θ	H (m)
3098	5.57	5.58	40.53	34.95	144.4
4926	9.95	9.96	64.45	54.49	358
6198	12.34	12.35	81.09	68.74	568.2

The results given in the previous table, state the requirement of the experimental comparison. Therefore, verification of the results will also be done and advantage of the straight type blade will be proved by experimentally in next Chapter.

4.2.4. CFD Analyses

CFD analyses for the overall system were made by Punto Mühendislik Ltd. Şti. by using CFdesign program to obtain the fan characteristics of the currently used fan and to understand the air flow rates, vacuum level and pressure losses in the whole cleaning system of current sweeper.

Before the experiments, CFD analyses express the approximate dimensions of the fan providing required flow rate and pressure difference.

CFD results also express the system curve approximately for the required flow rate, pressure difference and power.

Table 11 Properties of the Current Müsan Fan

Fan Diameter (mm)	250
Blade Width (mm)	95
Hub Pitch (mm)	30
Pitch (mm)	95
Number of Blades	8
Sense of Rotation	CCW
Rotational Speed	5747
Suction Diameter (mm)	90
Blow Diameter (mm)	96
Suction Velocity (m/s)	29,5
Blow Velocity (m/s)	37
Suction Area (mm ²)	6361.7
Blow Area (mm ²)	4512
Flow Rate (m ³ /h)	675,6

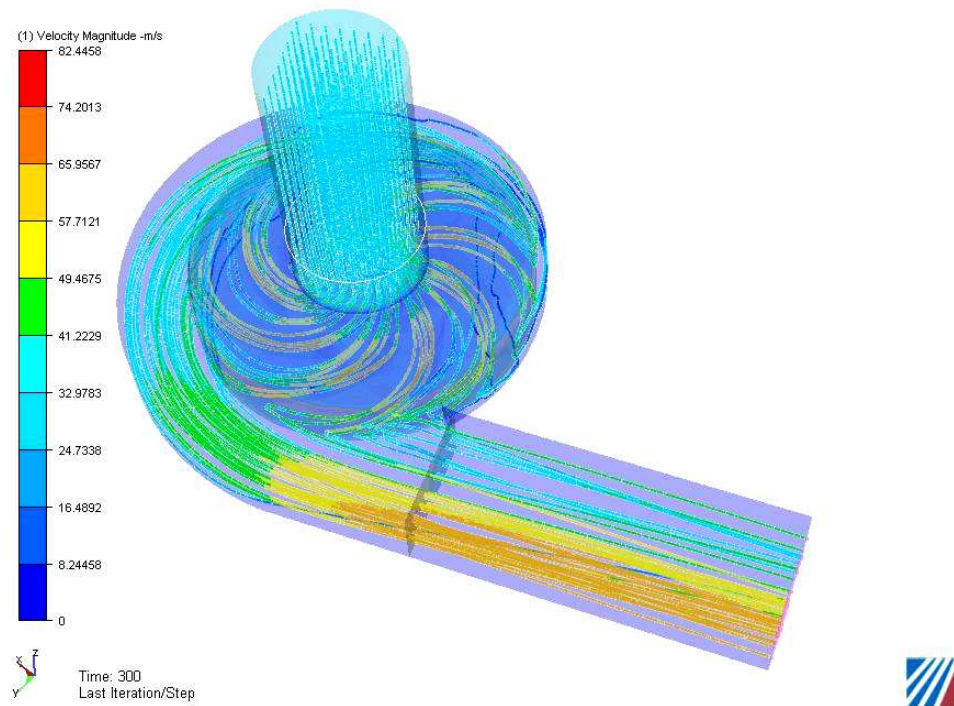
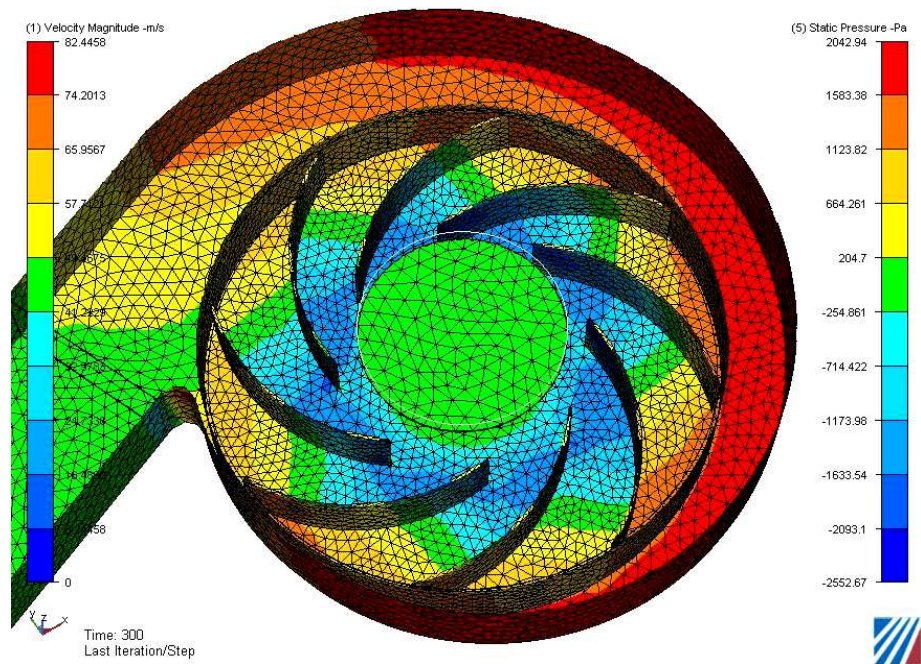


Figure 35 Pressure&Velocity Distributions of Former Fan (D=250 mm)

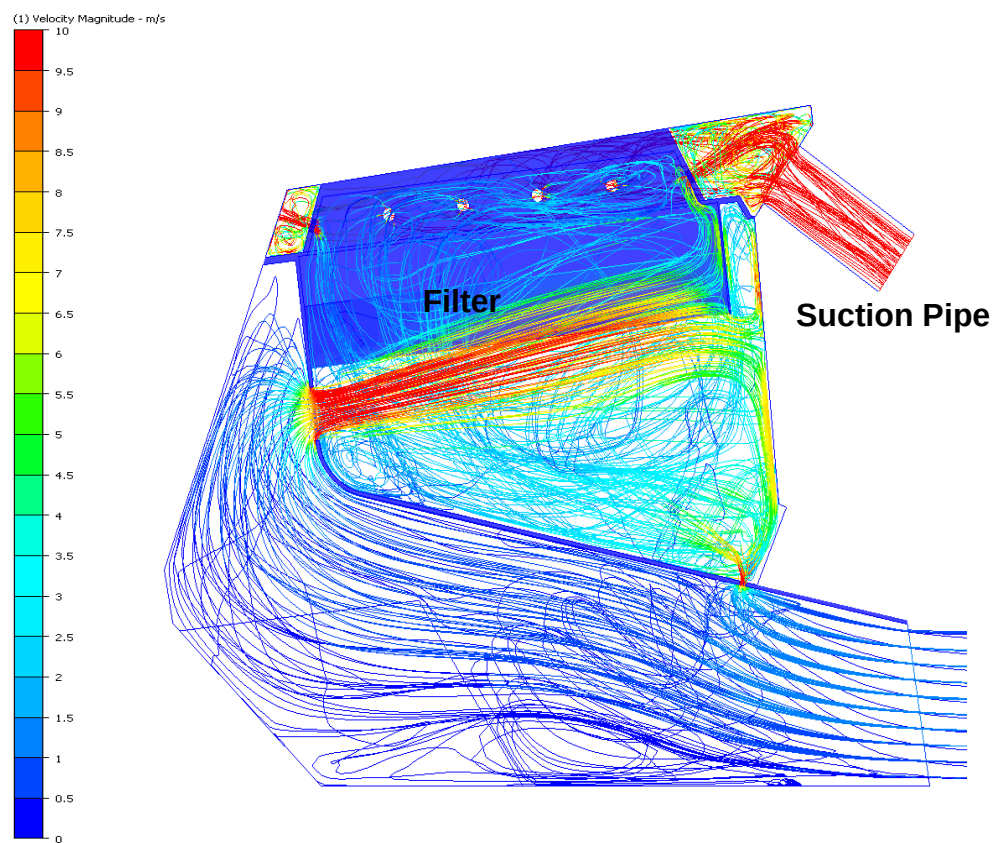
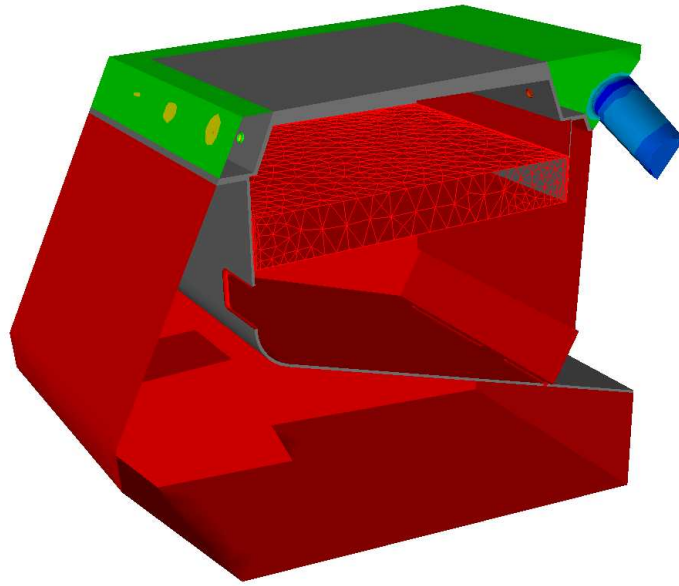


Figure 36 Velocity Distribution of Trash Container of the Current Sweeper

The Figures (Figure 35 and Figure 36) show that the velocity distribution in the trash container of the current MÜSAN Sweeper is not uniform. Tip speed of the curved blade fan is approximately 60-70 m/s at 6000 rpm and it converges to the calculated results given in the Table 10.

According to the initial analysis results, following steps are required to increase cleaning performance and to continue design activities;

- Air flow rate must be increased as much as possible.
- Pressure loss must be minimized throughout the cleaning system.
- Design must be done to prevent air pressure losses.
- Air flow through the filter must be uniform in order to avoid blockage on some areas.
- Appropriate fan(s) must be determined (in the range of the required values such flow rate, pressure difference and power)
- Fan performance tests and system tests (during all sub-systems are working on vehicle) must be performed.

After the 3D modeling of other systems such trash container, elevator system and side sheets by using Catia and Solid Works, CFD analyses are run to develop fan and filtration system for the new sweeper design.

To decide the particle carrying air velocity, data given in Table 12 and 13, typical solid particle sizes and the pneumatic solid transportation velocities for some solid particles are investigated. It is decided to obtain an air velocity of 10-15 m/s at the hopper inlet. The fan capacity to obtain this velocity is calculated and fan is sized by the help of CFD analysis. [2]

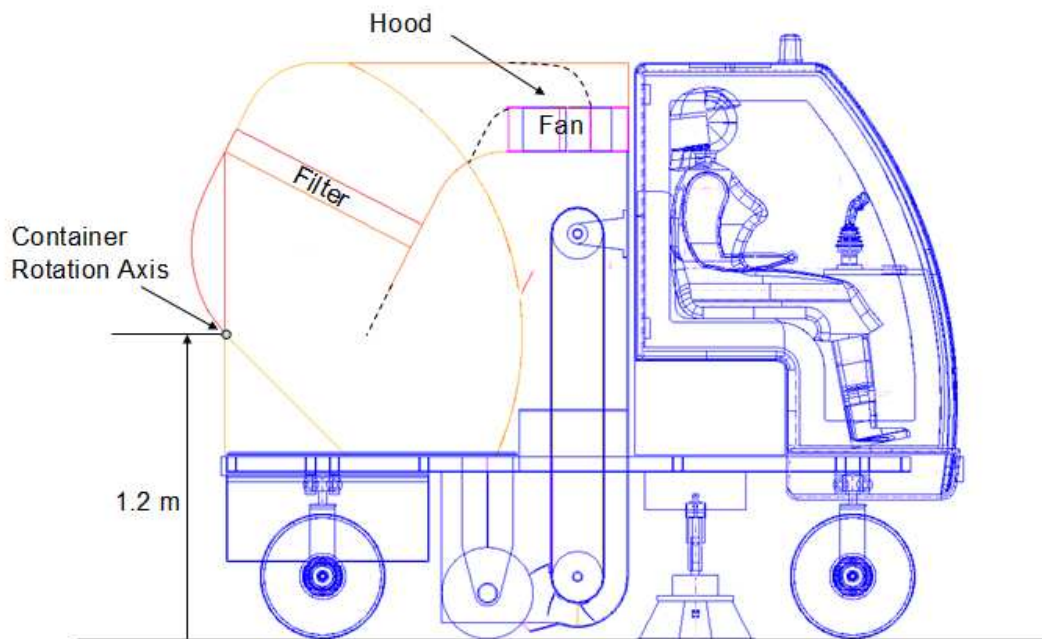


Figure 37 Arrangements of the Vacuum System Components

Table 12 Typical solid particle sizes [2]

Product	Typical Particle Size (micron = 0.001 mm)
Outdoor dust	0.5
Sand blasting	1.4
Foundry dust	1.0 - 200
Granite cutting	1.4
Coal mining	1.0
Raindrops	500 - 5000
Mist	40 - 500
Fog	1 - 40
Fly ash	3 - 70
Pulverized coal	10 - 400

Table 13 Pneumatic solid transportation velocities [2]

Product	Carrying Velocities (m/s)
Ashes, powdered clinker	30 - 43
Cement	30 - 46
Coal, powdered	20 - 28
Coffee beans	15 - 20
Cork	17 - 28
Corn, wheat, rye	25 - 36
Cotton	22 - 30
Flour	17 - 30
Grain dust	10 - 15
Grinding and foundry dust	17 - 23
Jute	22 - 30
Lead dust	20 - 30
Leather dust	8 - 12
Lime	25 - 36
Limestone dust	10 - 15
Metal dust	15 - 18
Oats	22 - 30
Plastic molding powder	15 - 17
Plastic dust	10 - 12
Pulp chips	22 - 36
Rags	22 - 33
Rubber dust	10 - 15
Sand	30 - 46
Sandblast	17 - 23
Sawdust and shavings, light	10 - 15
Sawdust and shavings, heavy	17 - 23
Textile dust	10 - 15
Wood chips	20 - 25
Wool	22 - 30

The analysis is run according to the air velocity of 10-15 m/s at the hopper inlet but from these results it is concluded that this value must be reconsidered. And also the fan volume can be reduced by choosing a fan with higher speed. **But as the fan input power varies with the cube of the fan speed ratio, an increase at the speed will result increasing of the power requirement as well.** Therefore, as it will be presented in the Chapter which is related with the experiments, an optimum solution is found according to the air velocity requirement at hopper inlet. After the experiments, it will be considered to use one or more than one fan in the system.

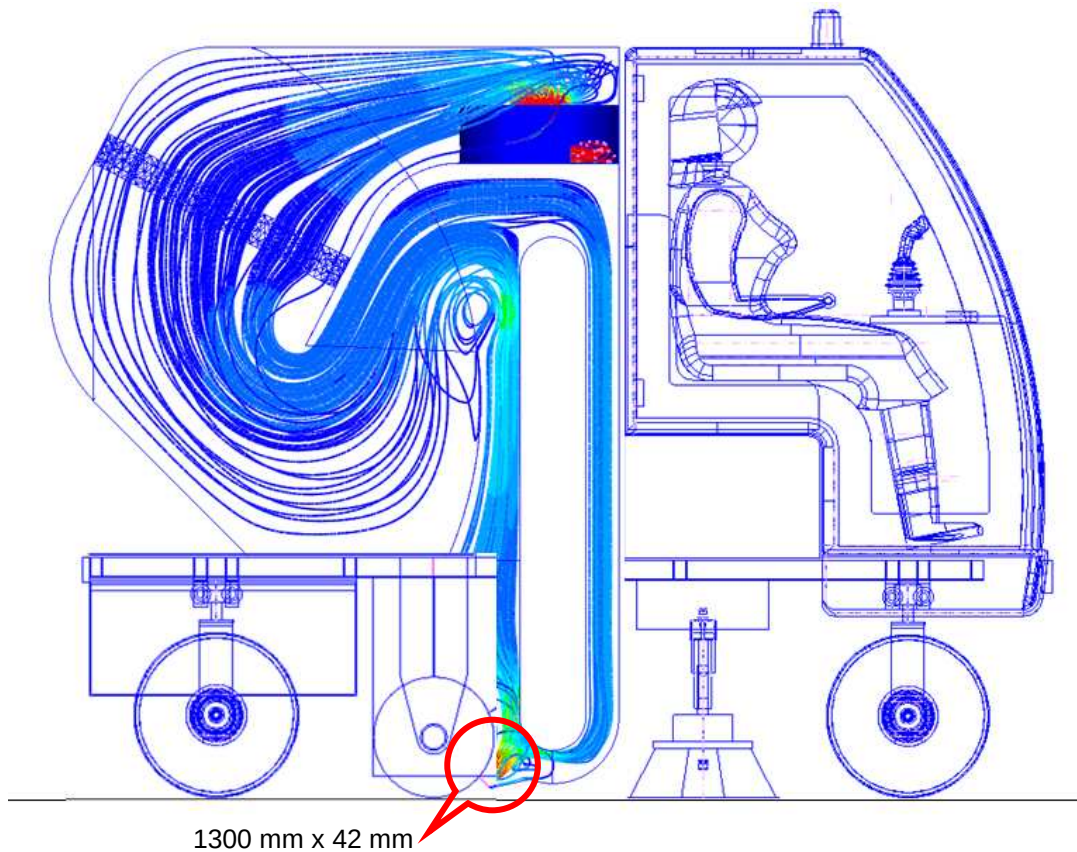


Figure 38 CFD Analysis of System (Determining approximate requirements)

$$A = 1300 \text{ mm} \times 42 \text{ mm} = 0.0546 \text{ m}^2$$

$$Q = A \times V$$

$$Q_{40\text{m/s}} = 0.0546 \text{ m}^2 \times 40 \text{ m/s} = 2.2 \text{ m}^3/\text{s}$$

$$Q_{30\text{m/s}} = 0.0546 \text{ m}^2 \times 30 \text{ m/s} = 1.6 \text{ m}^3/\text{s}$$

Flow Rate of the former MÜSAN Sweeper: $650 \text{ m}^3/\text{h} = 0.18 \text{ m}^3/\text{s}$

After the initial results, it seems that the required flow rate ($5200 \text{ m}^3/\text{h} - 7800 \text{ m}^3/\text{h}$) is more than 8-10 times than the flow rate of the fan used on current sweeper.

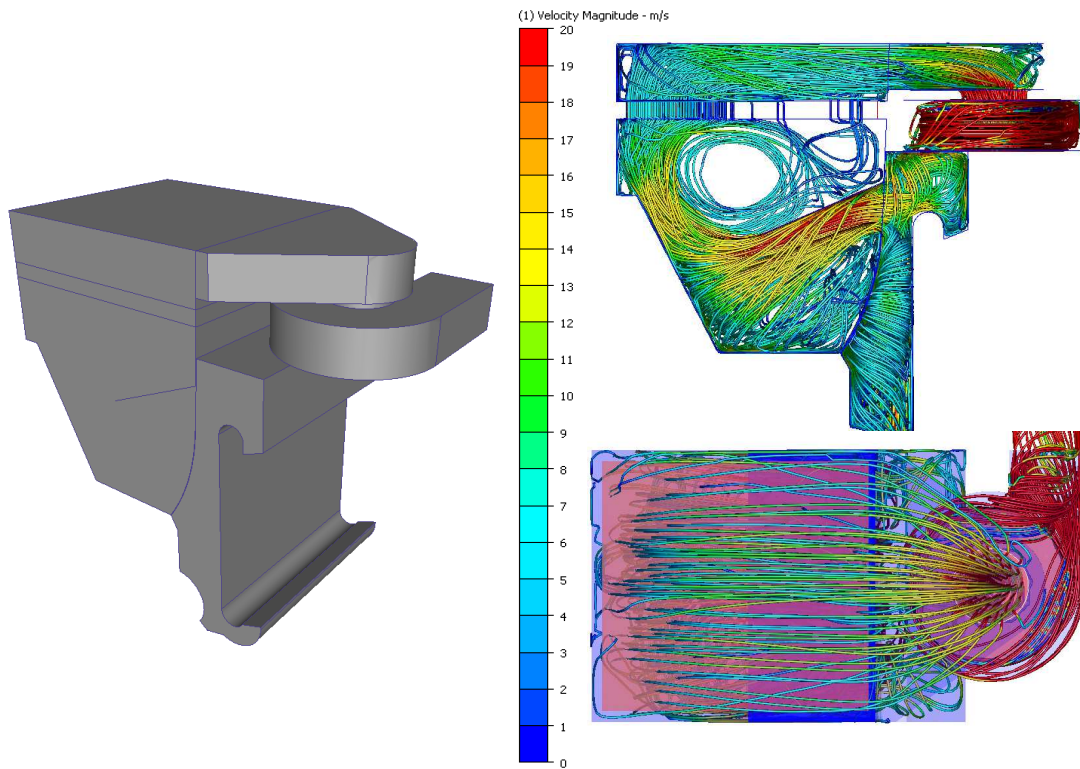


Figure 39 Streamlines of New Design

After the initial experiments, 3600-5200 m³/h flow rate needs much more power than expected (10-15 kW and this means using all energy for fan system) and since vacuum system is for the dust suction, fan development and dimension analysis is performed for the flow rate in the range of 2500-3600 m³/h.

Subsequent to this decision, market search will be done for procurement. In the case of not finding a fan to comply the requirements then a new fan prototype will be designed and constructed.

4.2.5. Fan Sizing and Development

Before the experiments, using the similitude equations which include flow rate, pressure and power parameters, dimension analysis and comparing of the results are done.

Table 14 Properties of the Former Fan (Backward-Curved Type Blade)

EXPEREMANTALY MEASURED (D=250 mm)			
RPM	H (Pa)	Q (m ³ /s)	P _f (W)
3000	392.4	0.11	41.35
4000	667.08	0.15	97.28
5000	981	0.20	196.26
6000	1393.02	0.24	332.89

Table 15 Properties of the Former Fan (Backward-Straight Type Blade)

EXPEREMANTALY MEASURED (D=250 mm)			
RPM	H (Pa)	Q (m ³ /s)	P _f (W)
3000	568.98	0.14	79.66
4000	981	0.19	182.47
5000	1657	0.25	414.25
6000	2315	0.31	717.65

As given in the previous section, flow rate and pressure are determined by using the velocity triangles (Table 10), these results are also determined experimentally (Table 14 and 15) and these verified results can be used as a reference for the determination of the characteristic dimensions of the fan(s) to be used at new design.

Assuming the new Diameter (D_2): 300 mm for the Curved Type Blade

$$D_2/D_1 = 300/250 = 1.2$$

$$5000 \text{ rpm: } Q_1=0.2 \text{ m}^3/\text{s}; H_1=981 \text{ Pa and } P_1=196.3 \text{ W}$$

For the same speed:

$$\frac{Q_2}{Q_1} = \left(\frac{N_2}{N_1} \right) \left(\frac{D_2}{D_1} \right)^3 \Rightarrow Q_2=0.35 \text{ m}^3/\text{s}$$

$$\frac{H_2}{H_1} = \left(\frac{N_2}{N_1} \right)^2 \left(\frac{D_2}{D_1} \right)^2 \Rightarrow H_2=1413 \text{ Pa}$$

$$\frac{P_2}{P_1} = \left(\frac{\rho_2}{\rho_1} \right) \left(\frac{N_2}{N_1} \right)^3 \left(\frac{D_2}{D_1} \right)^5 \Rightarrow P_2=488.4 \text{ W}$$

Assuming the new Diameter (D_2): 300 mm for the Straight Type Blade

$$D_2/D_1 = 300/250 = 1.2$$

$$3000 \text{ rpm: } Q_1=0.25 \text{ m}^3/\text{s}; H_1=1657 \text{ Pa and } P_1=414.3 \text{ W}$$

For the same speed:

$$\frac{Q_2}{Q_1} = \left(\frac{N_2}{N_1} \right) \left(\frac{D_2}{D_1} \right)^3 \Rightarrow Q_2 = 0.43 \text{ m}^3/\text{s}$$

$$\frac{H_2}{H_1} = \left(\frac{N_2}{N_1} \right)^2 \left(\frac{D_2}{D_1} \right)^2 \Rightarrow H_2 = 2386 \text{ Pa}$$

$$\frac{P_2}{P_1} = \left(\frac{\rho_2}{\rho_1} \right) \left(\frac{N_2}{N_1} \right)^3 \left(\frac{D_2}{D_1} \right)^5 \Rightarrow P_2 = 1030.8 \text{ W}$$

Table 16 Comparison of Results (Rotor Diameter is 300 mm)

CURVED BLADE FANS						
EXP. MEASURED (D=250 mm)				SIMILARITY (D=300 mm)		
RPM	H (Pa)	Q (m ³ /s)	P _f (W)	H (Pa)	Q (m ³ /s)	P _f (W)
5000	981	0.20	196.26	1412.64	0.35	488.36
6000	1393.02	0.24	332.89	2005.949	0.41	828.34
STRAIGHT BLADE FANS						
EXP. MEASURED (D=250 mm)				SIMILARITY (D=300 mm)		
RPM	H (Pa)	Q (m ³ /s)	P _f (W)	H (Pa)	Q (m ³ /s)	P _f (W)
5000	1657	0.25	414.25	2386.08	0.43	1030.79
6000	2315	0.31	717.65	3333.6	0.54	1785.74

Using the velocity triangle equations given in the previous section, speed values of the rotor (D=300 mm) are also checked for the determination of fan(s) to be run at the new vehicle.

Table 17 Blade Speeds (Rotor Diameter is 300 mm)

ROTATION (rpm)	V_{m2} (m/s)	w_θ (m/s)	U_2 (m/s)	V_θ (m/s)
3098	6.70	6.71	48.64	41.93
4926	11.73	11.74	77.34	65.6
6198	15.36	15.38	97.34	81.93

According to the initial results, straight type blade seems more productive due to higher velocity values (also proved experimentally and will be given in the next chapter in detail). The other rotor (diameter of 300 mm) which is not manufactured, that seems not to be sufficient as flow rate and pressure values.

As it can be seen from the following results (Table 18), when the fan rotor is enlarged to the diameter of 400 mm, flow rate and pressure values converges to the required operation point. (Approximately 1 m³/s flow rate and 952 Pa pressure difference)

U_2 is the peripheral velocity (tip speed) and higher values of tip speed may cause noise problems. It is observed that after the 100-120 m/s, noise problems occur. Therefore, in addition to the calculation of the internal flow parameters, tip speed (U_2) is also calculated for the rotors which will be manufactured before the experiments.

Table 18 Comparison of Results (Rotor Diameter is 400 mm)

STRAIGHT BLADE FAN ROTOR						
CALCULATION & MEASUREMENT (D=250 mm)				SIMILARITY& MEASUREMENT (D=400 mm)		
RPM	H (Pa)	Q (m ³ /s)	Pf(W)	H (Pa)	Q (m ³ /s)	Pf(W)
4000	981	0.19	182.47	2511.36	0.76	1913.29
5000	1657	0.25	414.25	4241.92	1.02	4343.73
6000	2315	0.31	717.65	5926.4	1.27	7525.11

Table 19 Blade Speeds (Rotor Diameter is 400 mm)

ROTATION (rpm)	V_{m2} (m/s)	w_{θ} (m/s)	U_2 (m/s)	V_{θ} (m/s)
3098	9.07	9.08	64.85	55.76
4926	16.24	16.25	103.12	86.86
6198	20.22	20.24	129.74	109.51

Besides the shape of the blade, blade quantity also takes an important role for the efficiency of fan(s). As it is examined in next title, optimum blade quantity is calculated theoretically and these results can also be verified experimentally. So therefore, comparison of different number of blades (obtained performance curves after the tests) has been examined in Chapter 5.

4.2.6. An Investigation on Number of Blades

Unlike the case with axial impeller there is no mathematical method to determine an optimum number of blades for a radial impeller. [20] As an average we can assume the blade length l to be approximately 50% longer than the radial depth $r_2 - r_1$. The passage depth is $a = t \sin \beta_2$; with $l = 1.5(r_2 - r_1)$ we get:

Table 20 General Properties of New Fan Blade

BLADE INLET ANGLE (deg)	β_1	45
BLADE OUTLET ANGLE (deg)	β_2	45
INLET RADIUS (mm)	R_1	80
OUTLET RADIUS (mm)	R_2	190
WIDTH (mm)	B	50

$$\frac{l}{a} = \frac{1.5(r_2 - r_1)}{t \sin \beta_2} = \frac{1.5(r_2 - r_1)z}{2r_2\pi \sin \beta_2} = 2 \Rightarrow z = \frac{4\pi}{1.5} \frac{\sin \beta_2}{\left(1 - \frac{r_1}{r_2}\right)} \approx 8.5 \frac{\sin \beta_2}{1 - \frac{r_1}{r_2}}$$

$$z = \frac{8.5 \sin 45}{1 - \frac{D_1}{D_2}} = \frac{8.5 \times 0.707}{1 - \frac{160}{380}} \Rightarrow z \cong 10.38$$

Thus the number of blades depends only on the discharge angle and the radial ratio. The formula gives an approximate indication of the number of blades required for normal radial impellers, while multivane impellers require more specialized treatment. However, the optimum number of blades of a radial impeller can only be truly ascertained by experiment.

As it can be seen in the section of performance experiments, it is also proved and compared experimentally that we can apply this formula for the non-radial (straight type and while β_2 is not equal to zero) centrifugal fans with the around 35% of convergence. [20]

4.2.7. Manufacturing of the Rotors and Vacuum Fans

A typical centrifugal (vacuum) fan consists of mechanical components such fan wheel or rotor (composed of a number of fan blades, mounted around the hub) and housing. The diameter of the former MÜSAN fan rotor was 250 mm and there were 8 blades mounted around the rotor. MÜSAN VSM 06 sweeper uses the backward-curved blade fan to vacuum. There is also another fan to be examined experimentally in next chapter. These fans (former MÜSAN fans to be tested and to be checked by the calculation and CFD results as well) are shown in the following figures.



Figure 40 Former Müsan Fan (D= 250 mm)

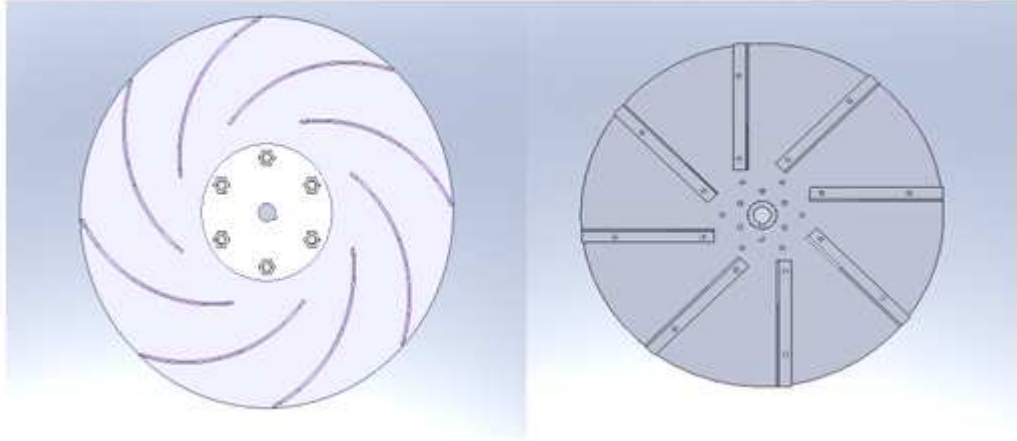


Figure 41 Former Müsan Rotors (Curved & Straight)

As shown in the previous titles, tip speeds (peripheral velocity on blade tip) and the flow parameters such flow rate, pressure and power are calculated and considering the power consumption (as low as possible), required operation point and probable sound problems, diameter of 400 mm fan rotors are manufactured from the steel material before the experiments.



Figure 42 Welding of Blades around the Fan Rotor (D=400 mm & 8 Blades)



Figure 43 Welding of Blades around the Fan Rotor (D=400 mm & 10 Blades)

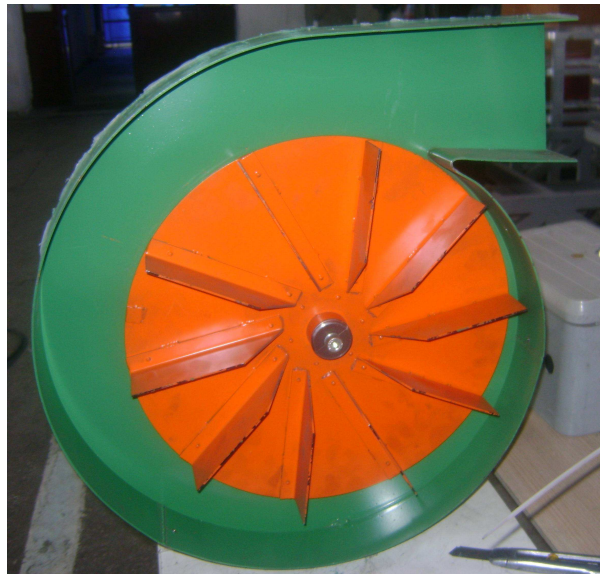


Figure 44 Mounting of the Rotor to the Housing

Before the system experiment is performed, the rotors (Non-radial, backward and straight type with 8, 10 and 12 number of blades) and cases are manufactured and an assembly box is manufactured from the aluminum material due to carrying of the fans and to provide mounting of the fans onto the sweeper.

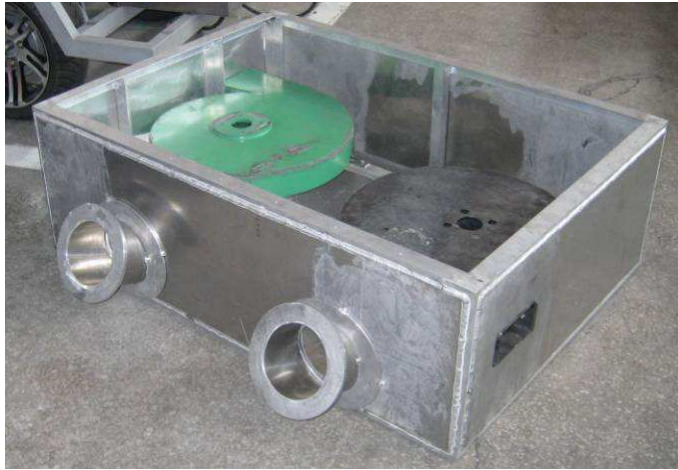


Figure 45 Assembly Box for the Fans (Initial Design)

Initially, 48V electric motors (brushless DC) were considered (experiments were also done with electric motors) to run fans, but then, by the change of the system from electric to electro-hydraulic, hydraulic motors were preferred for vacuum system.

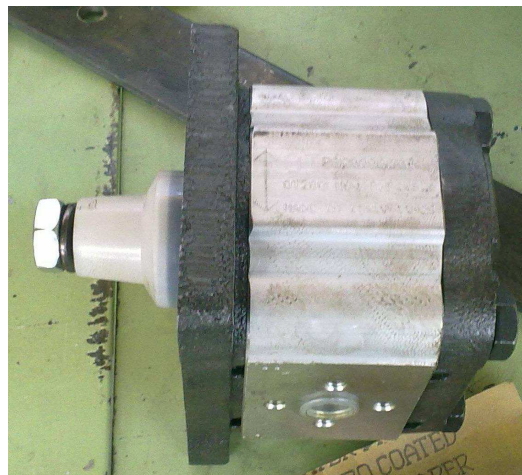


Figure 46 Hydraulic Motor for Fans

As final design, after the results of the system experiments given in the next chapter in detail, assembly box is cancelled and two fans are directly connected to the container as given in the Figure 47.

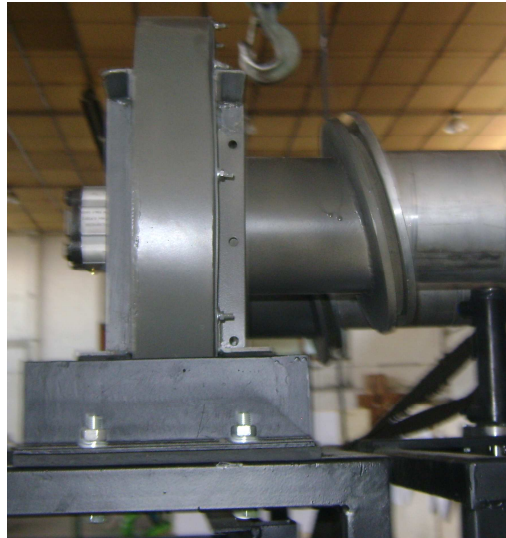


Figure 47 Arrangement of the fans directly (final design)

Screws are taken to outside of the case to prevent the resistances against air and rubber pads are used in between all metal-metal contacts to prevent sound and air leakages as well. The other important change is position of the fans. Fans are mounted to the chassis with supports as the outlets of the fans are positioned vertically. Gravels or any other small particles will be discharged to atmosphere vertically and possible injuries will be avoided by this way.



Figure 48 Mounting of the Fan group to Sweeper

4.3. Manufacturing of the Other Sub-Systems before Experiments

4.3.1. Trash Container and Filter

As it can be seen from the following Figures, trash container is manufactured from the Al sheets and combined together by using the welding techniques. Elevator is collecting the trashes with its blades and discharging them to the trash container assembled at the back of the sweeper. In Figure, it is seen that there is a slot for 'Electro mechanic Thruster' to overturn the container for discharging.



Figure 49 Trash Container and Thruster Slot (Initial Design)



Figure 50 Electro mechanic Thruster (Initial Design)

There will be two filters for new design mounted inside the trash container to filter particles. 'Filter Lost Coefficient' is also calculated by determining the static and dynamic pressure while the system is working. This topic will be examined under the section named 'System Experiments'.



Figure 51 Inside of the Trash Container and Filters

By the change of the system from electric to electro-hydraulic, trash container was manufactured over again. Hydraulic piston and cylinder was preferred instead of the electro-mechanic thruster (Figure 50). For the assembly of the hydraulic cylinder, slot is cut off as shown in the following figure.



Figure 52 Trash Container and Hydraulic Cylinder Slot

4.3.2. Battery Cassette

Power is supplied from the batteries (16x 48V) to energize all of the sub-systems. These batteries are placed in the middle of the sweeper. Therefore, to hold those batteries, a battery cassette (or frame) is manufactured by using Al profiles due to the lower weight of Aluminum and fine strength properties.

The other property of this cassette is that the side profiles have slots which provide to house supports (wheels are down when the cassette is pulled) for changing of the batteries with new (charged) one.



Figure 53 Battery Cassette (with slots)



Figure 54 Placement of the Battery Cassette on the Sweeper

4.3.3. Cabin

By using the CATIA and SW 3D models, detailed cabin design is improved and manufactured from the circular Al profiles as shown in the following Figures.

MDF material is preferred for the doors, polycarbonate material is used for the front and side glasses due to its endurance structure and to provide comfortable view for operator with transparent property of Polycarbonate.

By the change of the system from electric to electro-hydraulic, cabin frame which was manufactured from cylindrical aluminium profiles has also been changed. New profiles are rectangular and dimension of the section is smaller than the old one. It decreases the mass as well.



Figure 55 Cabin Framework (Initial Design)



Figure 56 The Sweeper before Experiments

CHAPTER 5

EXPERIMENTS

For the design of the MÜSAN VSM 080 sweeper under SAN-TEZ project, commercial 3D modeling tool Catia™ and Solid Works are used. Technical drawings of related parts and assemblies with overall dimensions are given in Appendix.

5.1. AMCA Standard to be Used

AMCA International publishes and distributes standards, references, and application manuals for specifiers, engineers, and others with an interest in air systems to use in the selection, evaluation, and troubleshooting of air system components. Many of AMCA International's standards are accepted as American National Standards. A description of AMCA International's publications is contained in the Publications Catalogue. [21]

Manufacturing of the test set-up and experiments are performed according to the AMCA Standard 210-74. For this process, initially, an appropriate test set-up is chosen utilized from the AMCA Standard (ASHRAE 51-75 & AMCA 210-74). Then, using the flow rate and velocity relation, pipe diameter is found. Other dimension of the set-up is obtained according to the pipe diameter.

Determining the characteristic of a fan takes an important role for the fan design. Therefore, measurement points and locations of the test devices must be placed correctly to obtain reliable results. A schematic of the test set-up, which is used for the experiments in MÜSAN A.Ş., is given in the following Figure.

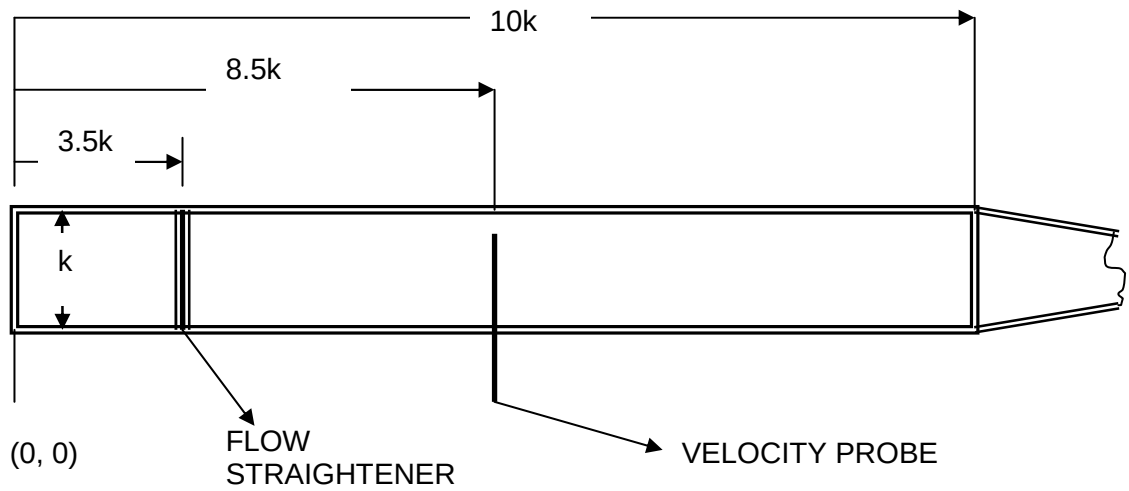


Figure 57 Schematic of the Test Set-up

In Figure 57;

k: diameter of the pipe (determined according to maximum velocity and capacity of the velocity probe)

(0, 0) point is the origin where conical vane is mounted.

5.2. Manufacturing of Test Set-up

Experiments are performed by using two test setups. Former MÜSAN sweeper is already using a backward centrifugal fan with 250 mm diameter and 8 blades. Determining the characteristics of the current fan is the initial step for a design. Hence, before the experiments, an appropriate test setup must be manufactured according to the AMCA standard and required velocity range.

According to the AMCA standard, test setup includes parts such pipe (average velocity calculation is performed for the selection of appropriate diameter), conical section of the pipe, throttling valve and flow straightener (inside of the pipe).

Using the obtained flow parameters, given in the previous chapter, 0.15 m³/s flow rate is taken as an approximate result belonging to the former Müsan Fan. The current test device (hot wire probe) is working in the range of 0-30 m/s, diameter of the setup can be determined from here. This diameter will also affect the length of the pipe.

If D=100 mm

$$V = \frac{0.150}{\frac{\pi * 0.1^2}{4}} = 20 \text{ m/s}$$

Since this result is about the device limit, Therefore, D=150 mm is taken. Thus,

$$V = \frac{0.150}{\frac{\pi * 0.15^2}{4}} = 8.5 \text{ m/s (estimated approximate velocity)}$$

Pipe is supplied from the market as PVC material. In standard, PVC pipe is found with the diameter of 156 mm. Throttling valve is wood material which is cut as the same diameter with pipe outlet. Straightener is manufactured by welding of aluminums (Al) sheets, which is placed inside of the pipe.

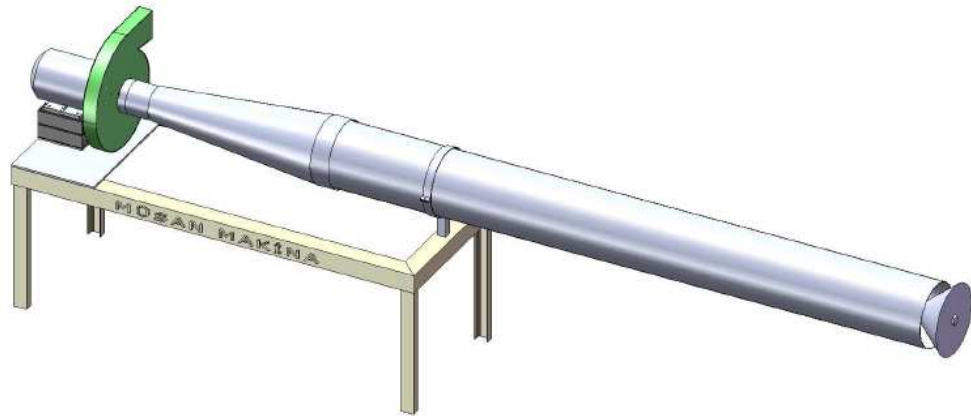


Figure 58 3D View of Test Set-up

Velocity measurements are very critical to determine flow rate through the pipe cross section. Therefore, to obtain reliable results, a probe fixer is manufactured and assembled to the test setup as shown in Figure 59.

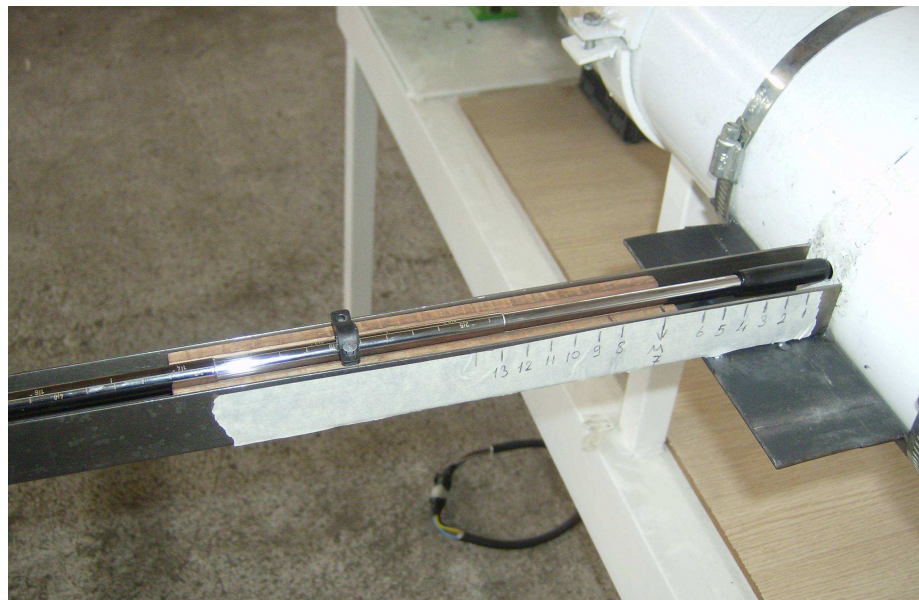


Figure 59 Probe Fixer mounted on the Pipe



Figure 60 Test Set-Up for the Experiments of Current Fans

Enlarged fan rotors (400 mm with different blade quantities) could not be tested with the available test setup due to much higher velocity values and limits of the measurement device as well.

Therefore, another test setup is manufactured by keeping the proportional limits in AMCA standard. Pipe diameter of the new test setup is determined as shown below:

$$Q_{\max}=1.5 \text{ m}^3/\text{s} \text{ (Similarity results give approximately } 1 \text{ m}^3/\text{s)}$$

$$D=300 \text{ mm}$$

$$V=\frac{4*1.5}{\pi*0.3^2}=21.2 \text{ m/s}$$

In standard, since there is PVC pipe with the inside diameter of 302.6 mm, Pipe diameter can be taken 302.6 mm. Unlike first test setup, throttling valve of the new one is manufactured from Aluminium (Al) material and mounted to the pipe with a worm screw throughout the pipe axis.

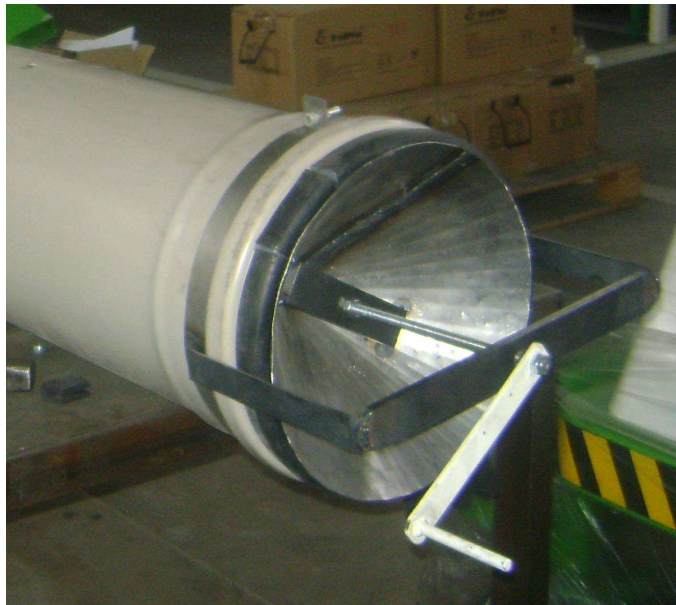


Figure 61 Throttling Valve (Conical)



Figure 62 Flow Straightener

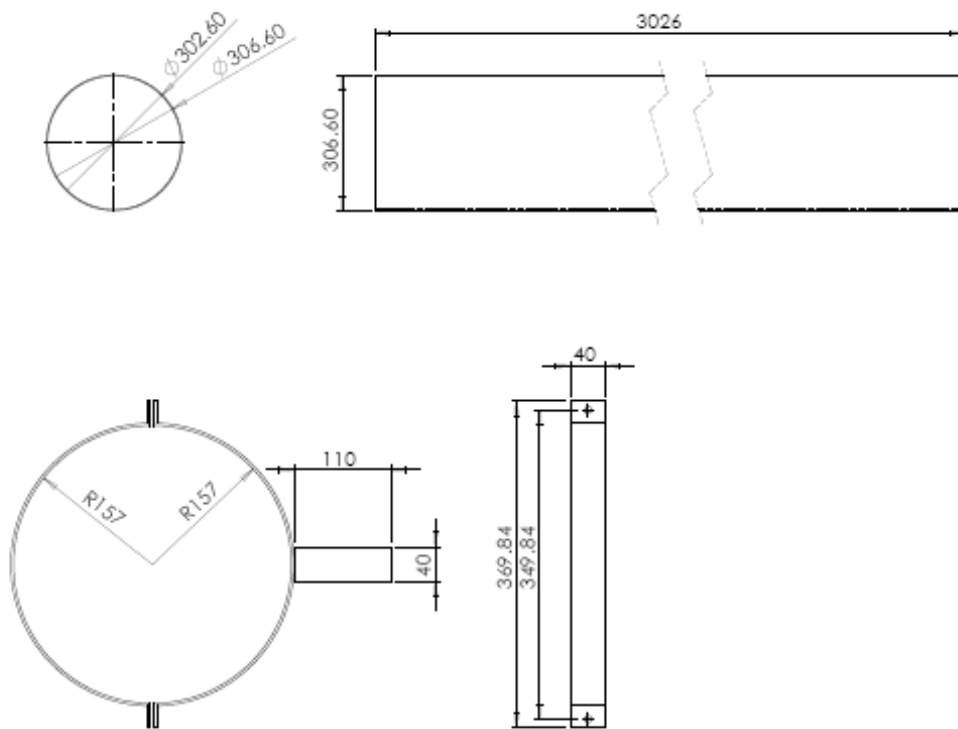


Figure 63 Outer Dimensions of the Test Setup



Figure 64 Test Setup for the New Fans

5.3. Properties of Test Devices

Technical properties of the measurement devices, which are used for the experiments, are given in the following pages.

- TESTO 435-4 combine measurement device is used for the velocity and temperature measurements.
- Pitot tube is used for the velocity and dynamic pressure measurements where the velocity is higher than 30 m/s.
- Electric Motor, wattmeter, inverter and energy analyzer are also used throughout the experiments.

Detailed information and technical properties belonging to test devices can be given as follow:

Testo 435-4 Combine Measurement Device:

Properties of the Testo 435 (with hot wire probe) can be stated as follows:

Testo 435-4; multi-function measurement instrument with built-in differential pressure measurement for A/C, ventilation and Indoor Air Quality with reading memory, PC software and USB data transmission cable, with battery and calibration protocol.

Velocity : 0 to 20 m/s (thermal)
0.2 to 60 m/s (air propeller)

Temperature: -50 to +150°C (NTC)
-200 to +1.370°C

Humidity : 0 to %100 RH

Measurement of CO₂ : 0 to 10.000 ppm

Absolute pressure : 0 to 2.000 hPa (mbar)

Light Intensity : 0 to 100.000 Lux

Pressure Difference : 0 to 25 hPa (mbar)

Many types of probes can be used for these devices but general properties of the probe, which is used throughout the experiments in Müsan, is given below:

Hot wire probe for 0 to 25 m/s (30 m/s can be measured as limit),

-50 to +150°C temperature measurement,

Ø7.5 mm probe head with telescopic handle (maximum 820 mm)



Figure 65 Testo-435 Measurement Device [22]

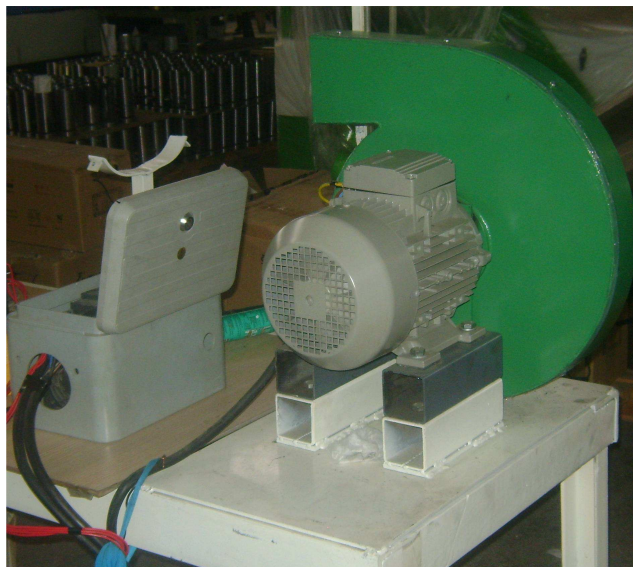


Figure 66 Mounting of Electric Motor and Invertor

Energy Analyzer:

Initially, magnetic ampermeter and voltmeter are used for measuring the motor power. Results are obtained using the Formula $P=V \times I$ (where V is Volt and I is current on the electric motor). But due to the unsteady values of ampermeter, an analog wattmeter is connected to the system for measuring the power.

Analog wattmeter could not give the requested steady results, too. Eventually, Energy analyzer (with integrated transformers) is mounted to system as shown in the following figure and very steady results are obtained.



Figure 67 Energy Analyzer for Power Measurement

Pitot tube:

Stainless steel, L (Length) =500 mm, Ø (Diameter)=7 mm, $T_{\max}=600^{\circ}\text{C}$

5.4. Testing and Calculation Methods

Tests and calculation processes are performed by following the procedure written below and given as shown in figures.

- Pipe (which is used in experiments) section is separated to sectors. In the following figure, through the inside pipe section, expected velocity distribution is also given.

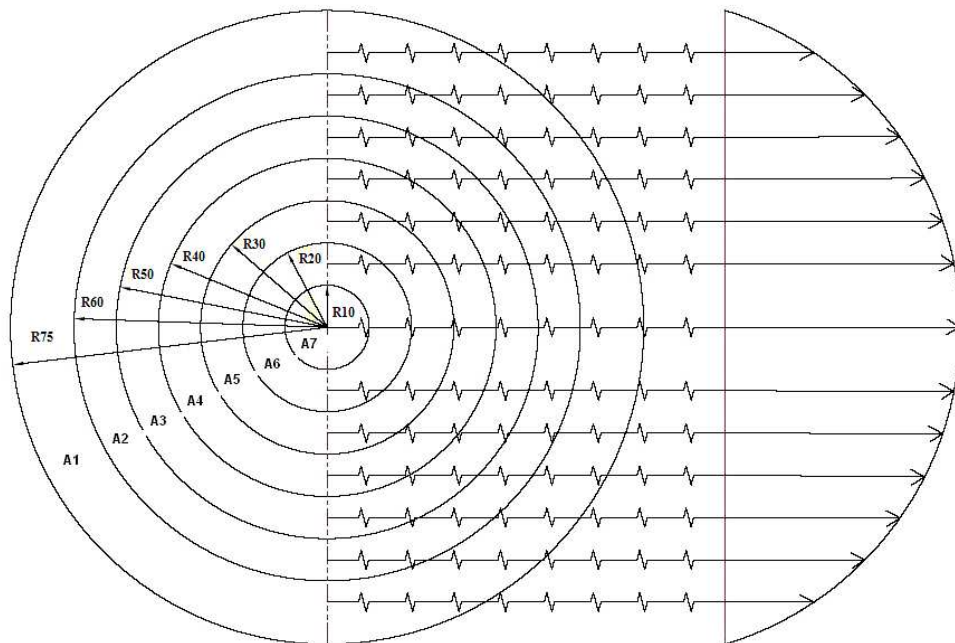


Figure 68 Area Sectors and Velocity Distribution through the Pipe Section

- Each area is calculated using the diameters of sectors as given below.

Table 21 Diameter & Area Values of the Initial Test-Setup

CIRCULAR AREA DISTRIBUTION						
D1 (m)	D2 (m)	D3 (m)	D4 (m)	D5 (m)	D6 (m)	D7 (m)
0.156	0.126	0.106	0.086	0.066	0.046	0.018
A1 (m ²)	A2 (m ²)	A3 (m ²)	A4 (m ²)	A5 (m ²)	A6 (m ²)	A7 (m ²)
0.006641	0.0036424	0.0030144	0.0023864	0.0017584	0.00140672	0.000254

- Velocity measurement is taken from these sectors (four different valve position and four different speed) and flow rate per sector is found ($Q_i = A_i V_i$ where $i=1$ to 7 and $i=7$ is the center measurement point). This measurement is done by using the TESTO 435 which is fixed onto the 'probe fixer' as shown in Figure 59.
- Then integrating the results gives flow rate for that valve position and for that speed specifically.

$$\Sigma Q = V_1 A_1 + V_2 A_2 + V_3 A_3 + \dots + V_7 A_7 = V_{13} A_{13} + V_{12} A_{12} + V_{11} A_{11} + \dots + V_7 A_7$$

- Measurement Table can be formed as shown in the following table:

Table 22 A sample Measurement Table belonging to Former Müsan Fans

MEASUREMENT TABLE (VELOCITY, FLOW RATE)

A (m²)		V (m/s)		Q (m³/s)
A ₁	0.0066411	V ₁	10.1	0.067
A ₂	0.0036424	V ₂	10.41	0.03
A ₃	0.0030144	V ₃	10.57	0.031
A ₄	0.0023864	V ₄	10.75	0.025
A ₅	0.0017584	V ₅	10.96	0.0192
A ₆	0.00140672	V ₆	11	0.015
A ₇	0.00025434	V ₇	11.04	0.0028
A ₈	0.00140672	V ₈	10.78	0.015
A ₉	0.00140672	V ₉	10.59	0.014
A ₁₀	0.0023864	V ₁₀	10.42	0.024
A ₁₁	0.0030144	V ₁₁	9.75	0.029
A ₁₂	0.0036424	V ₁₂	9.56	0.034
A ₁₃	0.0066411	V ₁₃	9.37	0.062
			ΣQ	0.200

- ΔP is measured from the inlet of the fan, where a hole is drilled onto the pipe of the test setup near the fan inlet and by using a U-type manometer (which has flexible hosepipe) , ΔP can be determined.

Then, P_f is obtained with the formula $P_f = \Sigma Q \times \Delta P$

- The values Volt (V), Current (I) are taken from the electric motor, P_m is obtained. As presented in the device properties, this technique is ignored and instead of it, energy analyzer is used for the power measurements.

Efficiency is determined. ($\eta = P_f / P_m$)

- And flow rate, pressure, power and efficiency results can be compared.

5.5. Experiments of Former Müsan Fans

According to the CFD results (Figure 69) and calculations given in the previous chapter, former fans (driven by VSM 06) which are manufactured by MÜSAN A.Ş. Company, are not sufficient. Therefore, to specify the performance characteristics of these fans and to compare CFD results and calculation results, performance experiments are performed.

As it can be seen from the following figure, performance curve of the former fans (current vehicle is being run by the curved blade type) are well below the desired operation point.

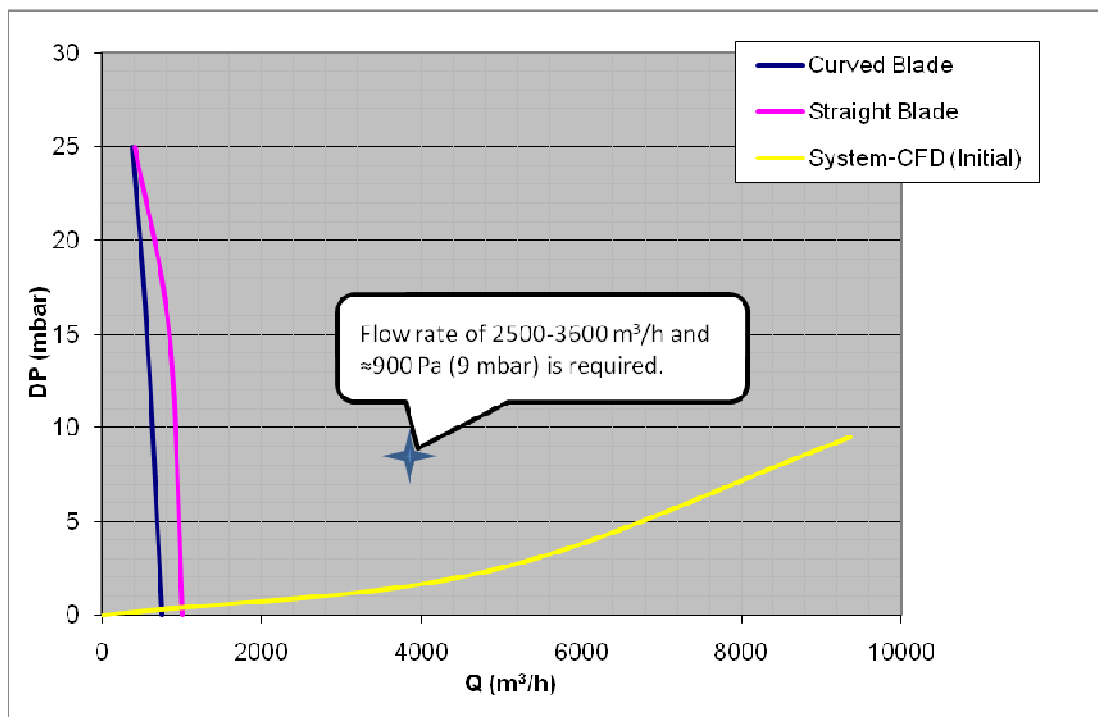


Figure 69 Former Fans and Initial System Curve (CFD)

Another considerable point is that, initially envisaged operation point is in the range of 3600-5200 m³/h of flow rate and due to the high power consumption, this value is reduced to range of 2500-3600 m³/h.

Determining the performance characteristic of the fans (curved and straight blade types) provides a comparison between the CFD results and experiment results.

5.5.1. Testing of Curved and Straight Blade Fans

VSM 06 Cleaner has a centrifugal fan (diameter of 250 mm and curved type blade) to vacuum. The other fan, diameter of 250 mm and straight type blade, is also manufactured by Müsan A.Ş. and wasn't tested until the experiments whether it is better than the curved one or not.

The procedure given in the previous title named as "Test Method" is followed step by step. Experiments are performed for different rpms (3000, 4000, 5000 and 6000 rpm) and for the different vane position (four different positions).



Figure 70 Set-up and Experiments of the Former Fans

5.5.2. Comparison of Curved and Straight Blade Fans

The curved and straight (blade type) fans can be compared according to the variables given below.

Table 23 Experiment Results (Former Fans) and Comparison Table

CURVED TYPE BLADE (D=250 mm & $\beta_2=26^\circ$)						
5000 RPM						
Position No	Valve Position	Flow Rate (m ³ /h)	ΔP (Pa)	ΔP (mbar)	P_f (W)	η (%)
1	Shut-Off	65.225	2805.660	28.06	50.83	
2	Open (75 mm)	586.75	1946.180	19.46	317.20	54.00
3	Open (90 mm)	650.01	1177.200	11.77	212.55	47.00
4	Open (160 mm)	720.23	981.000	9.81	196.26	42.00
6000 RPM						
Position No	Valve Position	Flow Rate (m ³ /h)	ΔP (Pa)	ΔP (mbar)	P_f (W)	η (%)
1	Shut-Off	113.052	3924.00	39.24	123.23	
2	Open (75 mm)	695.296	2654.40	26.54	512.66	53.60
3	Open (90 mm)	768.525	1628.46	16.28	347.64	49.40
4	Open (160 mm)	860.290	1393.02	13.93	332.90	42.00
STRAIGHT TYPE BLADE (D=250 mm & $\beta_2=45^\circ$)						
5000 RPM						
Position No	Valve Position	Flow Rate (m ³ /h)	ΔP (Pa)	ΔP (mbar)	P_f (W)	η (%)
1	Shut-Off	226.897	2786.040	27.86	175.596	
2	Open (75 mm)	811.399	2382.760	23.83	537.047	52.50
3	Open (90 mm)	854.234	1834.470	18.34	435.296	50.50
4	Open (160 mm)	912.367	1657.890	16.58	420.168	52.50
6000 RPM						
Position No	Valve Position	Flow Rate (m ³ /h)	ΔP (Pa)	ΔP (mbar)	P_f (W)	η (%)
1	Shut-Off	280.434	4022.100	40.22	313.315	
2	Open (75 mm)	981.024	3568.320	35.68	972.391	57.50
3	Open (90 mm)	983.690	2413.260	24.13	659.417	51.50
4	Open (160 mm)	1112.681	2315.160	23.15	715.565	55.40

After the initial results, it is observed that the straight blade fan has higher performance. As it can be seen from the previous table, straight type blade results in higher flow rate, pressure and efficiency as well. The following figures represent the superiority of the straight type blade clearly.

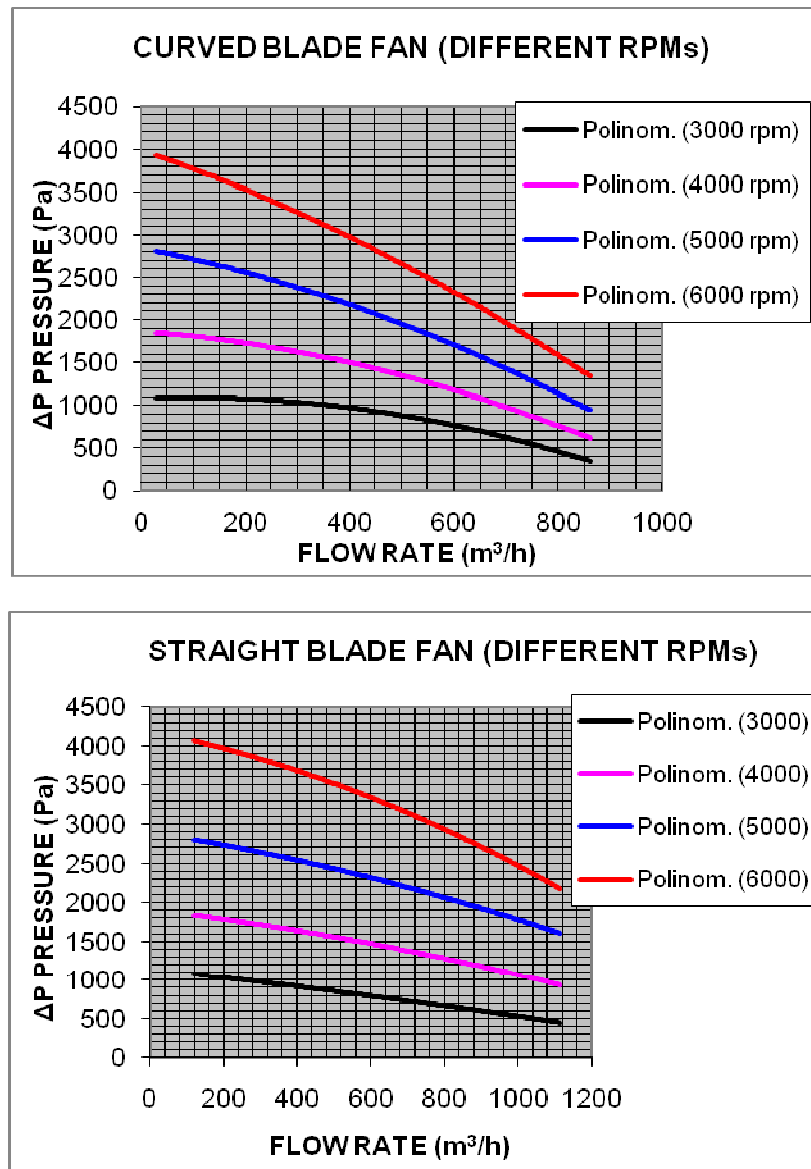


Figure 71 Performance Curves of the Former Fans

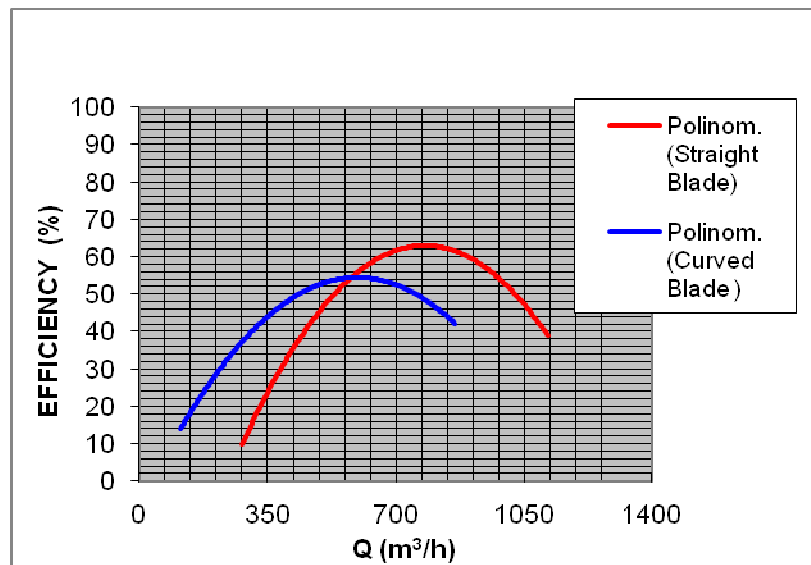


Figure 72 Efficiency Curves of the Former Fans

As it can be seen from the figures given above, the superiority of the straight type blade (as to former curved one) is also proved experimentally besides CFD and initial calculation results. Under the cover of the literature survey and experiments (CFD as well), fan rotors are considered to be manufactured with straight blades which can also be manufactured easier. (Different rotors are tested as mentioned in the next title).

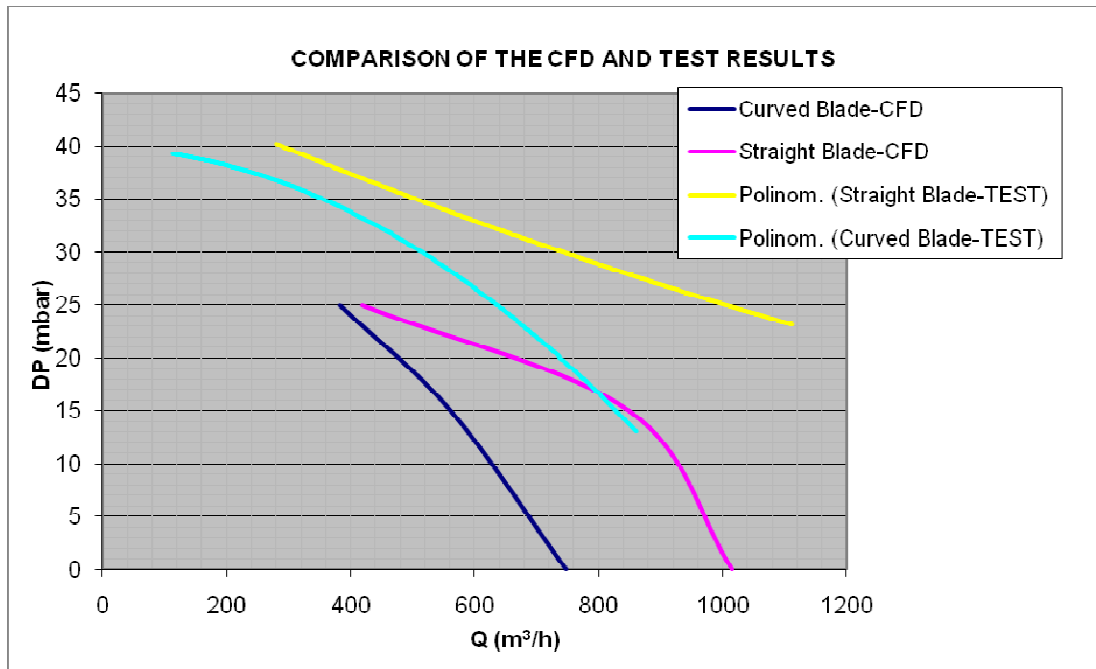


Figure 73 Comparison of the CFD and Experiment Results (Former fans)

5.6. Experiments of New Fans

Initially envisaged operation point is in the range of 3600-5200 m³/h of flow rate. Due to the high power consumption and considering the required vacuum conditions, this value is reduced to range of 2500-3600 m³/h.

As it is mentioned in manufacturing part, three different fan rotors are manufactured and performance experiments are performed at four different vane positions for new fan rotors. (Straight and 8, 10 and 12 blades) These experiments are performed on new test setup described in previous section.

5.6.1. Experiments of Straight Blade Fans

The procedure given in the previous title named as “Test Method” is followed step by step. For the 8, 10 and 12 bladed rotors, flow rate-pressure and flow rate-efficiency graphics are drawn. Using the whole datum, appropriate rotor is chosen and new fan is fixed to the system with the other subsystems before the system experiments start.

Experiments are performed for different RPMs (1000, 2000, 3000 and 3672) and for the different vane positions (four different positions).



Figure 74 Test Setup for New Fans

Table 24 Test Results (New Fans) and Comparison Table

12 BLADES						
1000 RPM						
Position No	Valve Position	Flow Rate (m ³ /s)	Flow Rate (m ³ /h)	ΔP (Pa)	P _f (W)	η (%)
1	Open (160 mm)	0,284	1023,991	196,200	55,807	62,01
2	Open (85 mm)	0,246	886,574	245,250	60,398	67,11
3	Open (45 mm)	0,166	599,070	294,300	48,974	69,96
4	Shut-Off	0,018	63,667	510,120	9,022	
2000 RPM						
Position No	Valve Position	Flow Rate (m ³ /s)	Flow Rate (m ³ /h)	ΔP (Pa)	P _f (W)	η (%)
1	Open (160 mm)	0,433	1560,396	941,760	408,200	59,16
2	Open (85 mm)	0,345	1241,966	1079,100	372,279	56,41
3	Open (45 mm)	0,324	1165,852	1373,400	444,773	68,43
4	Shut-Off	0,046	165,912	1863,900	85,901	
3000 RPM						
Position No	Valve Position	Flow Rate (m ³ /s)	Flow Rate (m ³ /h)	ΔP (Pa)	P _f (W)	η (%)
1	Open (160 mm)	0,553	1990,481	2648,700	1464,496	64,09
2	Open (85 mm)	0,468	1685,827	2844,900	1332,225	58,18
3	Open (45 mm)	0,348	1252,558	4022,100	1399,420	63,04
4	Shut-Off	0,148	532,948	5003,100	740,664	
3672 RPM						
Position No	Valve Position	Flow Rate (m ³ /s)	Flow Rate (m ³ /h)	ΔP (Pa)	P _f (W)	η (%)
1	Open (160 mm)	0,586	2111,073	4022,100	2358,596	57,53
2	Open (85 mm)	0,534	1920,624	4218,300	2250,492	55,84
3	Open (45 mm)	0,431	1552,036	5493,600	2368,407	59,51
4	Shut-Off	0,134	481,036	6965,100	930,685	
10 BLADES						
1000 RPM						
Position No	Valve Position	Flow Rate (m ³ /s)	Flow Rate (m ³ /h)	ΔP (Pa)	P _f (W)	η (%)
1	Tam Açık	0,283	1017,116	196,200	55,433	61,59
2	Açık (85 mm)	0,244	879,720	245,250	59,931	74,91
3	Açık (45 mm)	0,179	644,410	274,680	49,168	70,24
4	Tam Kapalı	0,019	67,818	490,500	9,240	
2000 RPM						
Position No	Valve Position	Flow Rate (m ³ /s)	Flow Rate (m ³ /h)	ΔP (Pa)	P _f (W)	η (%)
1	Open (160 mm)	0,434	1561,680	882,900	383,000	58,03
2	Open (85 mm)	0,345	1242,803	1079,100	372,530	57,31
3	Open (45 mm)	0,288	1036,988	1314,540	378,656	59,17
4	Shut-Off	0,049	177,084	1863,900	91,685	

Table 24 Continued

3000 RPM						
Position No	Valve Position	Flow Rate (m ³ /h)	Flow Rate (m ³ /h)	ΔP (Pa)	P_f (W)	η (%)
1	Open (160 mm)	0,548	1972,022	2550,600	1397,177	60,75
2	Open (85 mm)	0,462	1661,899	2746,800	1268,029	55,37
3	Open (45 mm)	0,348	1252,558	4022,100	1399,420	63,04
4	Shut-Off	0,153	551,762	5003,100	766,811	
3672 RPM						
Position No	Valve Position	Flow Rate (m ³ /h)	Flow Rate (m ³ /h)	ΔP (Pa)	P_f (W)	η (%)
1	Open (160 mm)	0,580	2087,917	3924,000	2275,829	55,78
2	Open (85 mm)	0,526	1893,580	4022,100	2115,602	53,16
3	Open (45 mm)	0,428	1539,158	5454,360	2331,978	58,89
4	Shut-Off	0,147	530,398	7063,200	1040,642	
8 BLADES						
1000 RPM						
Position No	Valve Position	Flow Rate (m ³ /s)	Flow Rate (m ³ /h)	ΔP (Pa)	P_f (W)	η (%)
1	Open (160 mm)	0,208	748,970	156,960	32,655	32,66
2	Open (85 mm)	0,199	717,877	196,200	39,124	43,47
3	Open (45 mm)	0,179	644,964	235,440	42,181	60,26
4	Shut-Off	0,018	65,566	588,600	10,720	
2000 RPM						
Position No	Valve Position	Flow Rate (m ³ /s)	Flow Rate (m ³ /h)	ΔP (Pa)	P_f (W)	η (%)
1	Open (160 mm)	0,347	1248,182	686,700	238,100	36,63
2	Open (85 mm)	0,334	1202,395	686,700	229,357	35,84
3	Open (45 mm)	0,280	1008,496	1177,200	329,778	54,06
4	Shut-Off	0,045	160,913	2256,300	100,852	
3000 RPM						
Position No	Valve Position	Flow Rate (m ³ /s)	Flow Rate (m ³ /h)	ΔP (Pa)	P_f (W)	η (%)
1	Open (160 mm)	0,540	1944,808	2354,400	1271,905	58,88
2	Open (85 mm)	0,440	1582,601	2550,600	1121,273	52,40
3	Open (45 mm)	0,348	1252,117	3825,900	1330,688	62,77
4	Shut-Off	0,122	438,107	4316,400	525,290	
3672 RPM						
Position No	Valve Position	Flow Rate (m ³ /s)	Flow Rate (m ³ /h)	ΔP (Pa)	P_f (W)	η (%)
1	Open (160 mm)	0,567	2041,492	3727,800	2113,965	51,81
2	Open (85 mm)	0,549	1977,888	3924,000	2155,898	53,76
3	Open (45 mm)	0,492	1772,594	4610,700	2270,250	54,57
4	Shut-Off	0,113	408,380	7259,400	823,498	

5.6.2. Performance and Efficiency Curves

All of the values are taken from the previous table and using curve fitting method for four different points, performance curves can be obtained.

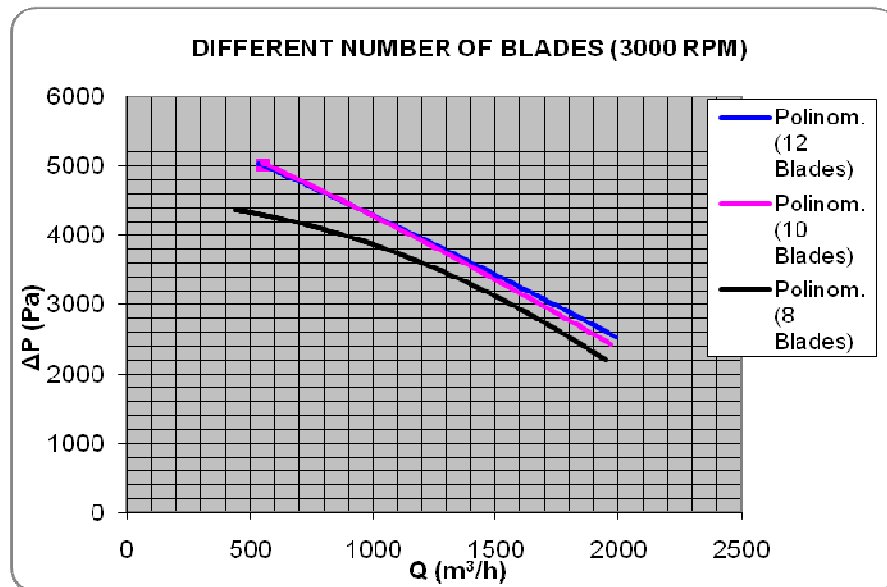


Figure 75 Comparison of the Different Number of Blades (3000 rpm)

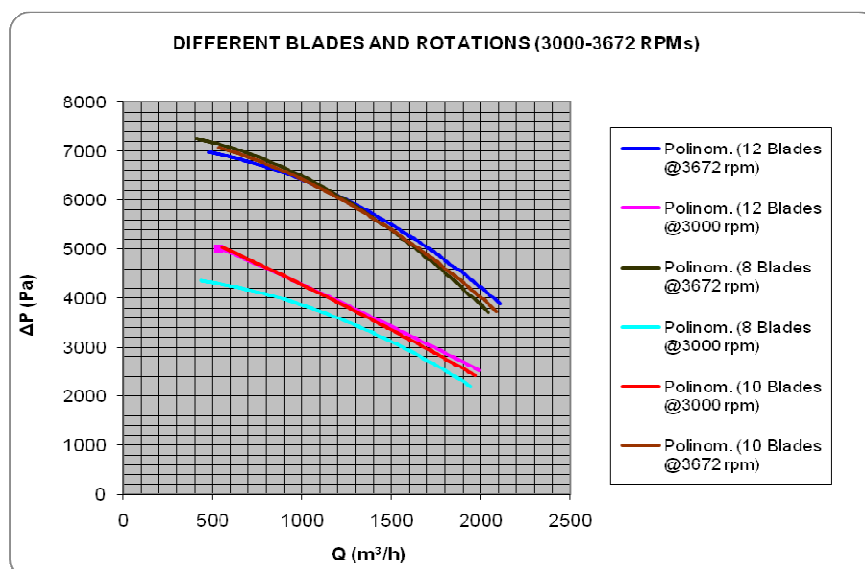


Figure 76 Comparison of the Different Number of Blades & Rotations

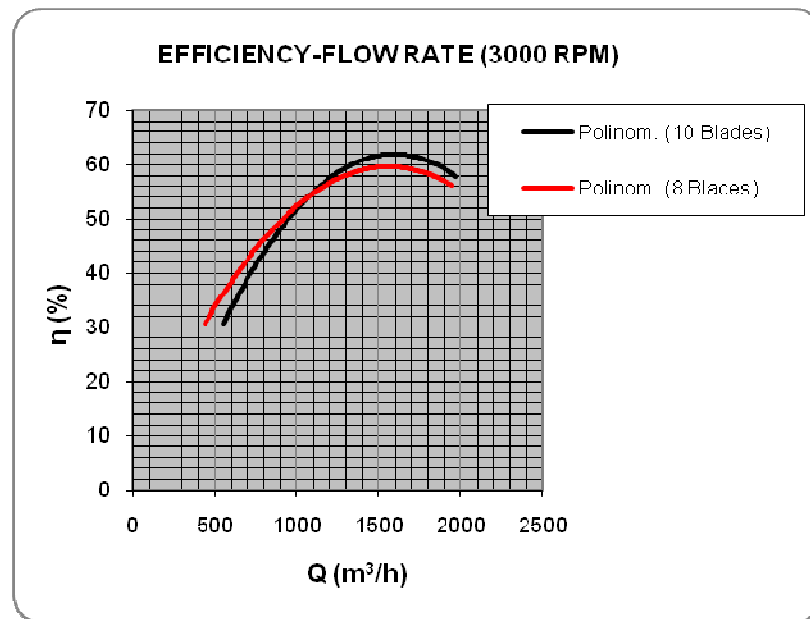


Figure 77 Efficiency Curves of Different Blades of the VSM 08

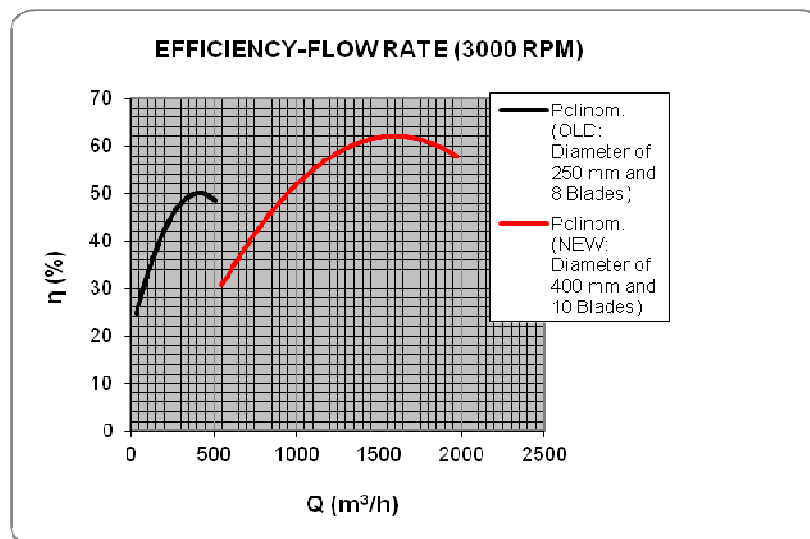


Figure 78 Comparison of VSM 08 (NEW) and VSM 06 (OLD) Fans

5.7. System Tests and Operation Points

After the performance experiments and manufacturing of the whole sub-systems, two different system experiments are conducted. One is that, using the 'fan assembly box' and the other, according to the results of the first experiment, neglecting the 'fan assembly box' and connecting the entrance of vacuum fan to the vehicle (trash container) directly. These two different experiment results are examined and finally, comparing the differences between different system experiments, optimum operation points (vacuum level) are found out.

5.7.1. System Test Procedure and Test Devices

- For the system tests, an inclined manometer and pitot tube are used to determine the velocities (flow rate and pressure difference consequently);



Figure 79 Measuring Air Speed at Fan Outlet with Pitot Tube

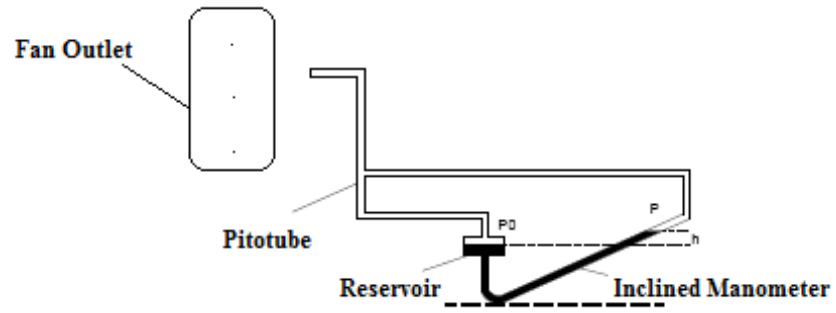


Figure 80 An Illustration of the Measurement Style

In the following Figure, pitot tube can be seen which is used for the system tests.

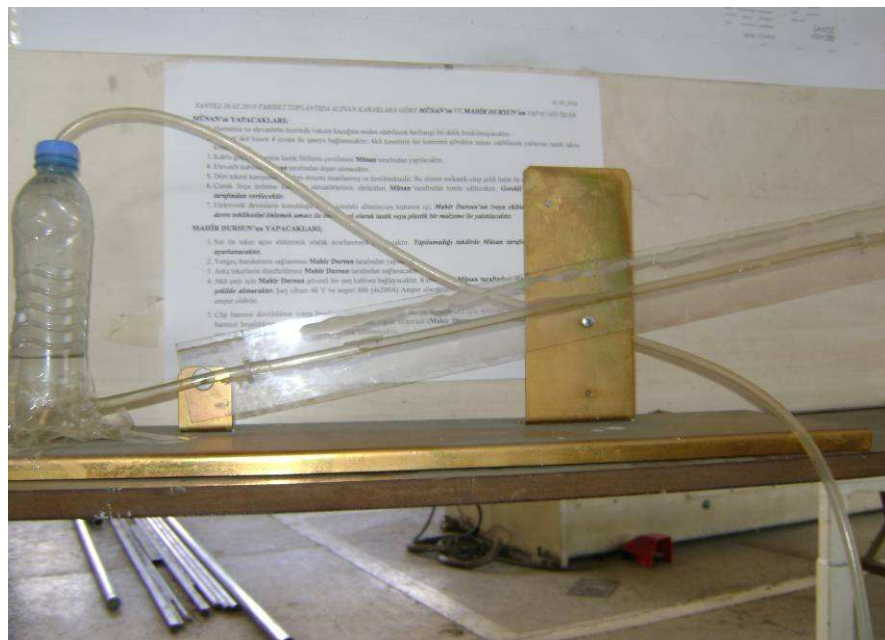


Figure 81 Inclined Manometer and Pitot Tube

- U type manometer is also used to measure the pressure difference inside the container (filter lost coefficient can be obtained).



Figure 82 Measuring with 'U' Type Manometer

- To measure these data, some holes are drilled at different areas on sweeper such container, fan assembly box (for the first experiment)

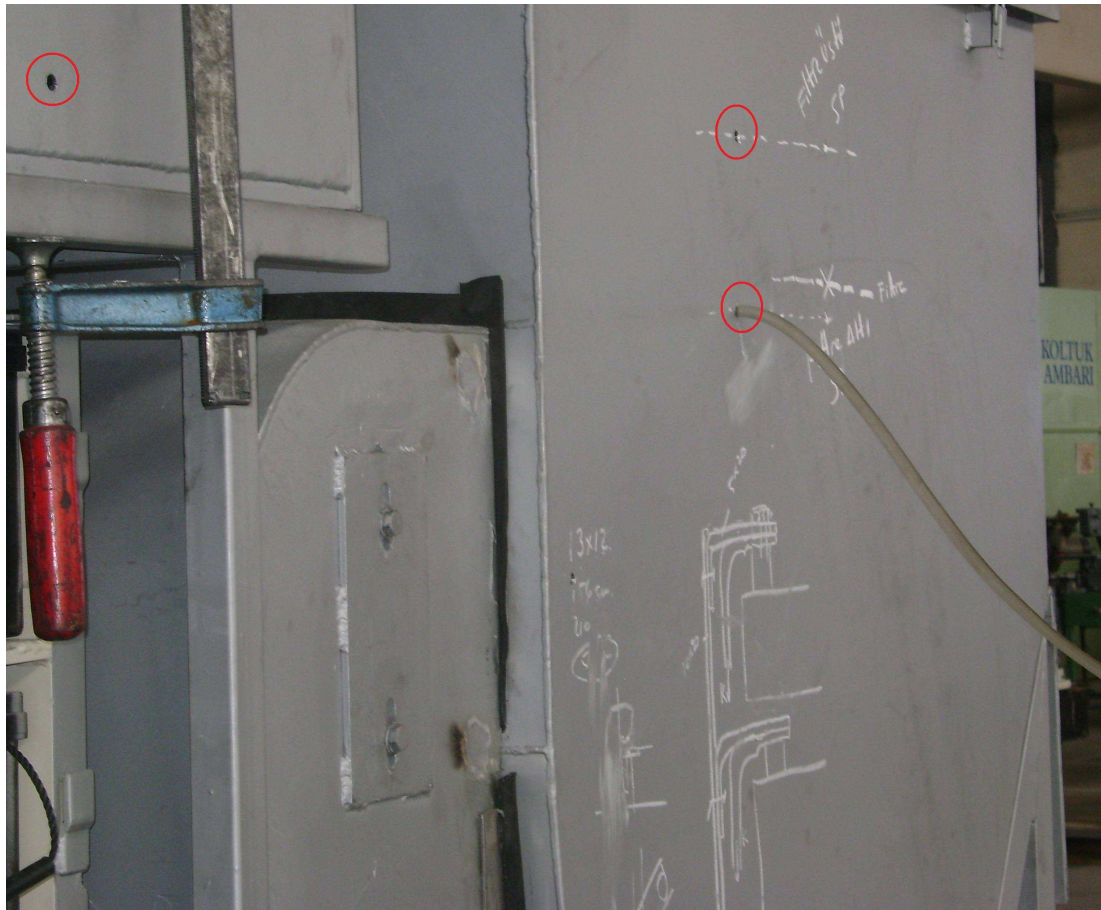


Figure 83 Drilled Holes at Vehicle

- TESTO Hotwire device is also used to measure the velocities at different locations of the sweeper such elevator inlet and outlet, filter inlet and outlet.

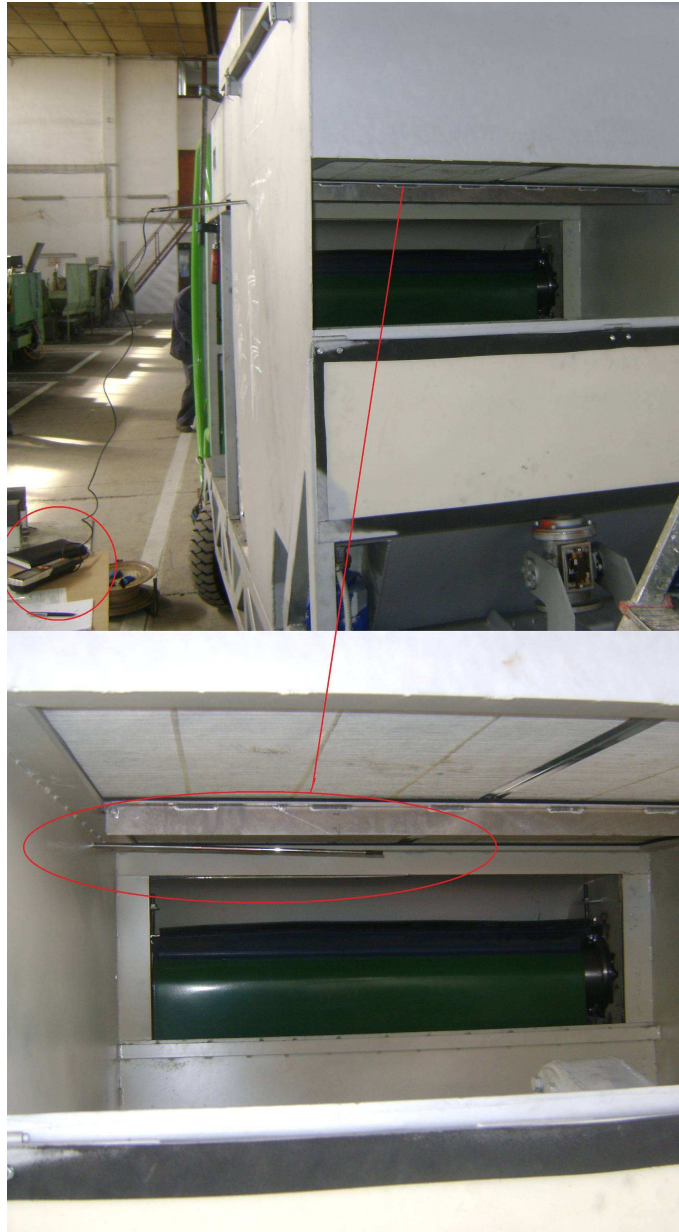


Figure 84 Measuring Air Speeds inside of the Container (Filter Inlet)

- Then, using all of the data taken from the system; flow rate, pressure, filter lost coefficient and air velocities at different areas of sweeper can be calculated and can be written as a 'Result Table' before drawing the system characteristic curve.
- Finally, 'Operation Point (O.P.)' is determined and all of the results can be compared.

5.7.2. System Experiment 1: With 'Fan Assembly Box'

As an initial work, system is checked out whether there is an adaptation problem of the mechanic components with electrical connections or not. When the synchronization is provided, system experiments started. After that, first measurements are taken while the elevator system is not working. But it is observed that the effect of the elevator blades is almost zero to the experiment results. So therefore, whole experiment data are started to be taken while the whole system is working.

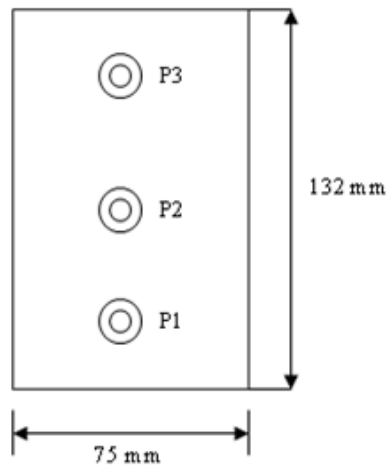


Figure 85 Measurement Points (Fan Outlet)

5.7.2.1. Calculations of System Experiment (1)

As it can be seen from the previous figure, measurements are taken from the fan outlets through the outlet section at three points by using pitot tube and inclined manometer. Then, the results are represented as a table including “U Manometer” measurement results as well.

Table 25 Experiment Data Table at Outlet

DATA ($\theta=11.5^\circ$ Inclined Manometer)		
	SYSTEM IS ALL WORKING	
	FAN(1)	FAN(2)
P1 (mm)	500	500
P2 (mm)	530	530
P3 (mm)	490	490
$\rho_w(kg/m^3)$	1000	
$\rho_{air}(kg/m^3)$	1,2	
$\sin 11.5$	0,19	

By using results given at previous table, calculations are performed as shown in the following page.

$$h = L \sin \theta$$

$$L=500 \text{ mm and } \sin(11.5)=0.19$$

$$\Delta P = P_0 - P = \Delta P = \rho_w g h = \rho_w g L \sin \theta \quad \text{where } (\theta=11.5)$$

$$P_0 = P + \frac{\rho_{air} V^2}{2}$$

$$\Delta P = \frac{\rho_{air} V^2}{2}$$

$$V = \sqrt{\frac{2\Delta P}{\rho_{air}}} \Rightarrow V = \sqrt{\frac{2\rho_w g L \sin \theta}{\rho_{air}}}$$

And finally, velocity at P1 point can be obtained as:

$$V = \sqrt{\frac{2 \times 1000 \times 9.81 \times 0.5 \times 0.19}{1.2}} \Rightarrow V = 39.41 \text{ m/s (Velocity at P1)}$$

Velocities for the other points (P1, P2) throughout the fan outlet section are also obtained by using the same procedure.

Average Outlet Velocity V_{ave} (m/s) is obtained as:

$$V_{ave} = \frac{V_{P1} + V_{P2} + V_{P3}}{3} \text{ (m/s)}$$

$V_{ave1} = 39.67 \text{ m/s}$ (V_{ave2} can be determined using the same formulas)

Since there will be two fans (Backward, 10 straight blades, 3000 rpm, 400 mm) at sweeper:

V_{ave1} : Average velocity of Fan(1)

V_{ave2} : Average velocity of Fan(2)

A: Outlet Area (m)

$$A = 0.132 \times 0.075 = 0.0099 \text{ m}^2$$

Q: Flow Rate (Total)

$$Q = A \times (V_{ave1} + V_{ave2})$$

$$Q = 0.772 \text{ m}^3/\text{s}$$

By the aid of U manometer:

$h=215$ mm (suction inlet of the fan)

$$\Delta P = \rho_w gh$$

$$\Delta P = 1000 \times 9.81 \times 0.215$$

$$\Delta P = 2109.15 \text{ Pa}$$

5.7.2.2. Determining the System Characteristic and Operation Point

Using the values obtained from the experiments, calculations are done and overall results are obtained as shown below.

Table 26 Experiment Results Table

3000 rpm	ELEVATOR DOES NOT RUN		ELEVATOR RUNS	
	FAN(1)	FAN(2)	FAN(1)	FAN(2)
V_{ave} (m/s)	39,01	39,01	39,67	39,67
Q (m ³ /s)	0,772		0,785	
ΔP (Pa)	2011,05		2109,15	

To draw the system curve and for determining the operation point, Q (flow rate), ΔP (pressure difference) and constant K (parabola coefficient) values are used. Firstly, K value is calculated from the initial experimental results then the system curve is fitted by using four different Q - ΔP values.

To obtain the K (parabola coefficient) value:

$$\Delta P = K \cdot Q^2$$

$$Q \text{ (m}^3/\text{h)} = 2827.51$$

$$\Delta P \text{ (Pa)} = 2109.15$$

$$K = 0.000264$$

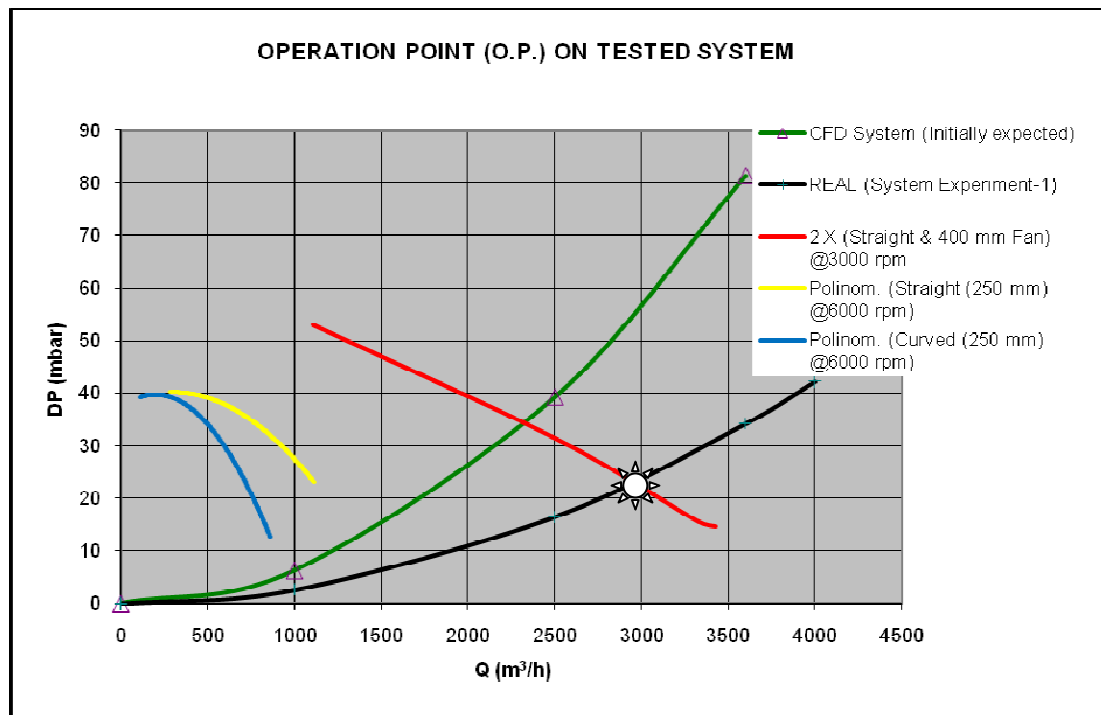


Figure 86 Characteristics Curve and Operation Point

Operation point: $0.81 \text{ m}^3/\text{s}$ ($2900 \text{ m}^3/\text{h}$) and 2250 Pa (2.25 mbar)

5.7.2.3. Speeds and Pressure in Fan Assembly Box

Arrangement of two fans and the electric motors in the assembly box can be seen in the following Figures. After the completing of this first experiment, to prevent the circulations in box and to overcome the losses (vacuum especially), box is canceled and relating to this new situation (no assembly box), new experiments are performed and these will be examined in next parts.

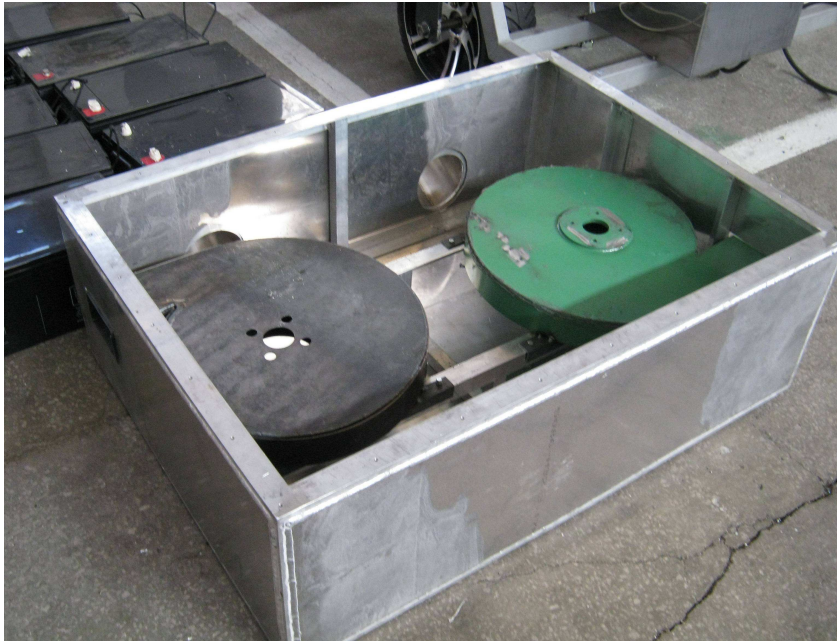


Figure 87 Mounting of the Fans



Figure 88 Mounting of the Electric Motors

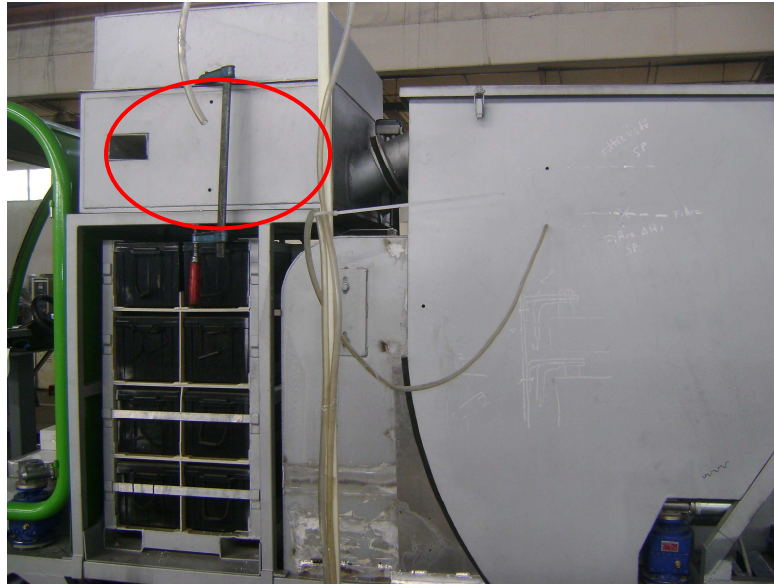


Figure 89 Assembly Box is on the Vehicle

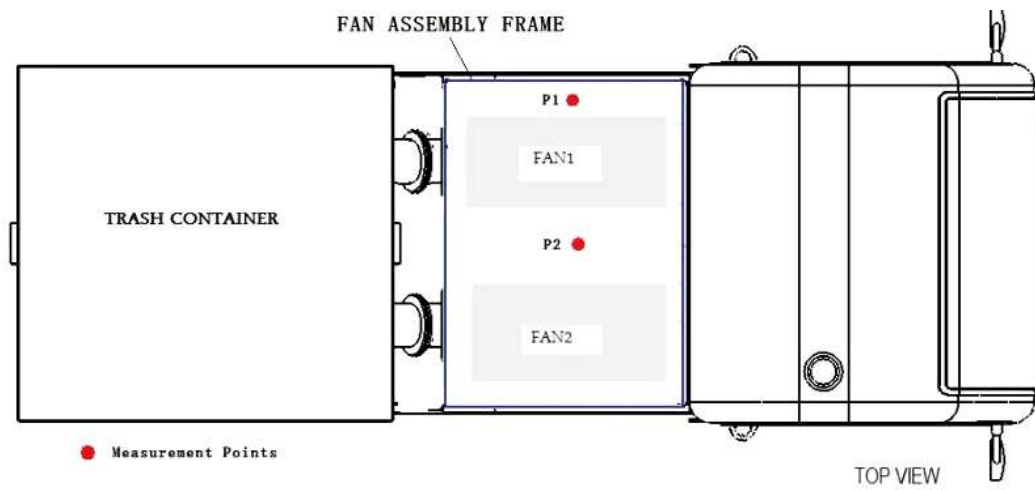


Figure 90 Measurement Points on Fan Assembly Box

Two different holes are drilled side surface of the fan assembly box and measured static and dynamic pressures and velocities as well using the hotwire and 'U' manometer over these drilled points.

Near P1 point:

H= 160 mm (U manometer) and Velocity: 8-10 m/s

$$P_{\text{static}} = 1570 \text{ Pa}$$

$$P_{\text{dynamic}} = \frac{\rho_{\text{air}} V^2}{2} = 60 \text{ Pa}$$

$$P_{\text{total}} = 1630 \text{ Pa}$$

Near P2 point:

H= 200 mm (U manometer) and Velocity: 15-20 m/s

$$P_{\text{static}} = 1962 \text{ Pa}$$

$$P_{\text{dynamic}} = \frac{\rho_{\text{air}} V^2}{2} = 240 \text{ Pa}$$

$$P_{\text{total}} = 2200 \text{ Pa}$$

After the calculations, related to the 'Fan Assembly Box', different pressures and changeable instant speeds mean that there might be air circulations and pressure losses inside the box. Therefore, this fan assembly box is neglected and examined all of the values in the second system experiment without 'Fan Assembly Box'.

5.7.2.4. Calculation of the Filter Loss Coefficient (FLC)

System contains a pair of filters. These filters are combined with an eccentric metal frame carrying 250 W brushless electric motor as well. This motor is used to provide vibration to shake filter off when the filter becomes clogged.



Figure 91 Filters and Brushless Electric Motor (250W)



Figure 92 Assembly of the Filters to the Vehicle (Top View)



Figure 93 Inside of the Container (Filter Bottom View)

As it can be seen from the following two figures, two different holes (one on the upper side of the Filter and the other on the lower side) are drilled at trash container. And by using the 'U' type manometer, required values are measured and calculations are performed as presented in next page.



Figure 94 'U' Manometer and Drilled Measurement Holes

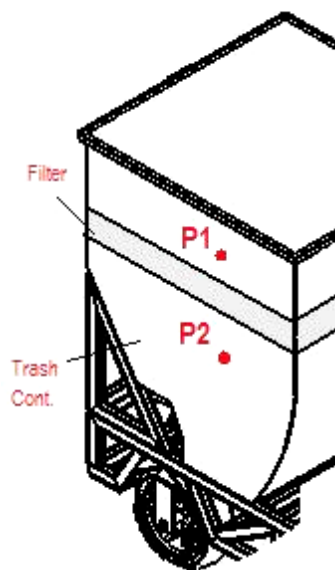


Figure 95 Illustration of Measurement Points and Container

As shown in the Figure 95, measurements are taken from the drilled holes named P1 and P2:

Table 27 Experiment Result Table for FLC

	P1 (Upper side of Filter) Point	P2 (Lower side of Filter) Point
ρ (kg/m ³)	1000	1000
H (mm)	50	48
P (Pa)	490.5	470.88
V_{avg} (m/s)	1.2	2.85

Calculating the Filter Loss Coefficient:

$$\Delta P = K \frac{\rho_{air} (V_{inlet}^2 - V_{exit}^2)}{2} \Rightarrow$$

$$V_{inlet} = 2.85 \text{ m/s}$$

$$V_{exit} = 1.2 \text{ m/s}$$

$$\Delta P = 19.6 \text{ Pa}$$

$$19.6 = K \frac{1.2 \times (2.85^2 - 1.2^2)}{2}$$

$$K = 4.87 \text{ (FLC)}$$

5.7.3. System Experiment 2: Directly Connected

After the initial system experiment, the results are evaluated and it is observed that pressure (vacuum) losses occur in the system due to fan assembly box. Therefore, secondly, this assembly box is removed from the system and another system experiment is performed following the same procedure.



Figure 96 Views of Connection of the Fans Directly



Figure 97 Electric Motors and Fans



Figure 98 Taking measurements during Experiment-2

5.7.3.1. Calculations of System Experiment (2)

As it is given in the previous experimental procedure, measurements are taken from the fan outlets through the outlet section at three points by using pitot tube and inclined manometer. Then, the results are represented as a table including U manometer measurements as well.

Table 28 Experiment-2 Data Table at Outlet

DATA ($\theta=11.5^\circ$ Inclined Manometer)		
	SYSTEM IS ALL WORKING	
	FAN(1)	FAN(2)
P1 (mm)	540	540
P2 (mm)	530	530
P3 (mm)	510	510
$\rho_w(\text{kg/m}^3)$	1000	
$\rho_{air}(\text{kg/m}^3)$	1,2	
$\sin 11.5$	0,19	

By using results given at previous table, calculations are performed as follows.

$$h = L \sin \theta$$

$$L=540 \text{ mm and } \sin(11.5) = 0.19$$

$$\Delta P = P_0 - P = \Delta P = \rho_w g h = \rho_w g L \sin \theta \quad \text{where } (\theta=11.5)$$

$$P_0 = P + \frac{\rho_{air} V^2}{2}$$

$$\Delta P = \frac{\rho_{air} V^2}{2}$$

$$V = \sqrt{\frac{2\Delta P}{\rho_{air}}} \Rightarrow V = \sqrt{\frac{2\rho_w g L \sin \theta}{\rho_{air}}}$$

And finally, velocity at P1 point can be obtained as:

$$V = \sqrt{\frac{2 \times 1000 \times 9.81 \times 0.54 \times 0.19}{1.2}} \Rightarrow V = 40.95 \text{ m/s (Velocity at P1)}$$

Velocities for the other points (P1, P2) throughout the fan outlet section are also obtained by using the same procedure.

Average Outlet Velocity V_{ave} (m/s) is obtained as:

$$V_{ave} = \frac{V_{P1} + V_{P2} + V_{P3}}{3} \text{ (m/s)}$$

$V_{ave1} = 40.45 \text{ m/s}$ (V_{ave2} can be determined using the same formulas)

Since there will be two (3000 rpm, 400 mm and 10 Blades) Fans at sweeper:

V_{ave1} : Average velocity of Fan(1)

V_{ave2} : Average velocity of Fan(2)

A: Outlet Area (m)

$$A = 0.132 \times 0.075 = 0.0099 \text{ m}^2$$

Q: Flow Rate (Total)

$$Q = A \times (V_{ave1} + V_{ave2})$$

$$Q = 0.801 \text{ m}^3/\text{s}$$

By the aid of U manometer:

H=350 mm (suction inlet of the fan)

$$\Delta P = \rho_w gh$$

$$\Delta P = 1000 \times 9.81 \times 0.35$$

$$\Delta P = 3433.5 \text{ Pa}$$

5.7.3.2. Determining the System Characteristic and Operation Point

Using the values obtained from the experiments, calculations are done and overall results can be obtained as shown below.

Table 29 Experiment-2 Results Table

3000 rpm	SYSTEM IS ALL WORKING	
	FAN(1)	FAN(2)
V_{ave} (m/s)	40,45	40,45
Q (m ³ /s)	0,801	
ΔP (Pa)	3433,50	

To draw the system curve and for determining the operation point, Q (flow rate), ΔP (pressure difference) and constant K (parabola coefficient) values are used. Firstly, K value is calculated from the initial experimental results then the system curve is fitted by using four different Q - ΔP values.

To obtain the K (parabola coefficient) value:

$$\Delta P = K * Q^2$$

$$Q = 2883.6 \text{ (m}^3\text{/h)}$$

$$\Delta P = 3433.5 \text{ (Pa)}$$

$$K = 0.000413$$

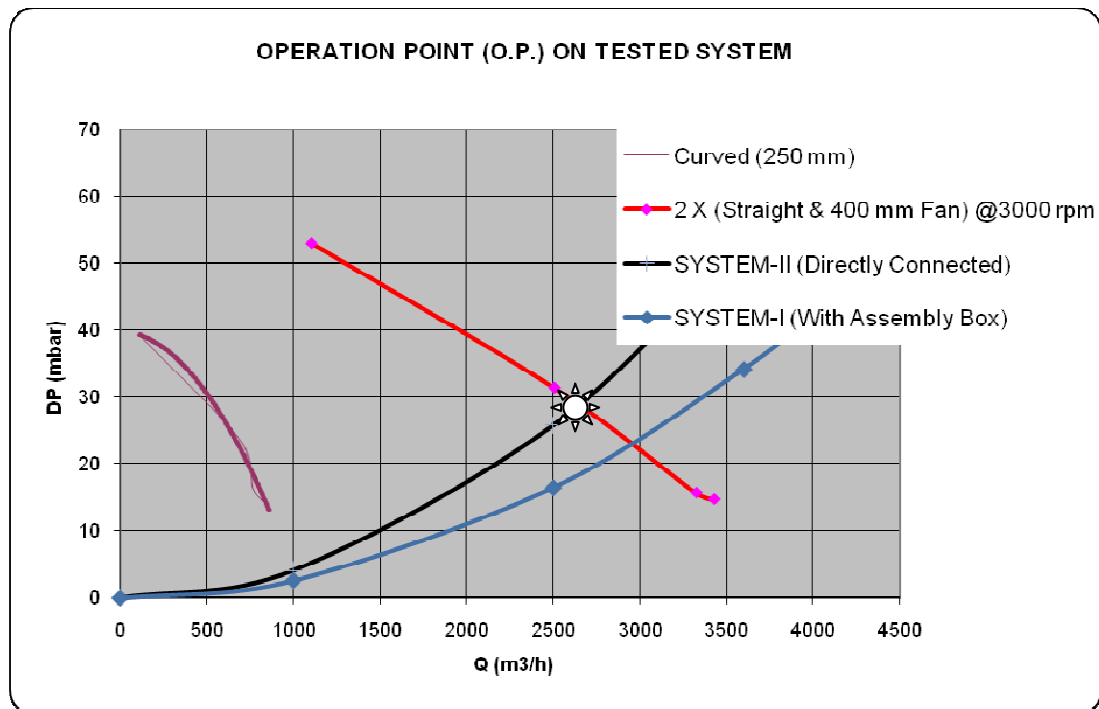


Figure 99 Characteristics Curve and Operation Point

Operation point: $0.74 \text{ m}^3/\text{s}$ ($2650 \text{ m}^3/\text{h}$) and 2950 Pa (2.95 mbar)

5.7.3.3. Calculation of the Filter Lost Coefficient (FLC)

As it can be seen from the previous experiment (1) part, two different holes (one is upper side of the filter and the other is lower) are drilled at trash container. And by using the 'U' type manometer, required values are taken and calculations are done as presented in next page.



Figure 100 Assembly of the Filters to the Vehicle (Top View)

As shown in the Figure 94 and Figure 95, measurements are taken from the drilled holes named P1 and P2:

Table 30 Experiment-2 Result Table for FLC

	P1 (Upper side of Filter) Point	P2 (Lower side of Filter) Point
ρ (kg/m ³)	1000	1000
H (mm)	51	48.5
P (Pa)	500	475
V_{avg} (m/s)	1.25	3.1

Calculating the Filter Lost Coefficient:

$$\Delta P = K \frac{\rho_{air} (V_{inlet}^2 - V_{exit}^2)}{2} \Rightarrow$$

$$V_{inlet} = 3.1 \text{ m/s}$$

$$V_{exit} = 1.25 \text{ m/s}$$

$$\Delta P = 24.53 \text{ Pa}$$

$$24.53 = K \frac{1.2 \times (3.1^2 - 1.25^2)}{2}$$

$$K = 5.081 \text{ (FLC)}$$

FLC value of the system experiment (1) was 4.87; and current FLC is found up as 5.081. So therefore, it is verified that the FLC is around 5.

CHAPTER 6

ELECTRO-HYDRAULIC TRACTION SYSTEM DESIGN

Traction system was initially considered to be run by using individual hub motors but after examining the domestic and foreign markets, it was seen that the electric car hub motor industry is not ready to provide torque requirements for low rotation and high torque applications. Therefore, force, power, rotation and torque values are calculated and then compared with the current hub motor (2.5 kW and 70 Nm) in graphics. Advantages and disadvantages of the hydraulic system are evaluated and electro-hydraulic system is preferred because of its cheaper, clean, wide and flexible usage field.

6.1. Determining the Torque Requirement

Total force includes; $F_T = F_r + F_g + F_a + m \times a_{\max}$ where rolling, gradient and air resistance forces and inertia acceleration forces respectively. Calculations are done for three different positions by changing variables acceleration, velocity and gradient. For the practice application and design process, position-1 and position-2 results are taken and hydro motors of the tractions system are started to be chosen according to these results.

Table 31 Position-1: ($a_{\max}=0$), ($V_{\max}=15$), Gradient ($\Theta=15^\circ$)

$V_{\max}$ (km/h)	TOTAL FORCE	POWER (W)	TORQUE (Nm)	ROTATION (rpm)
1	5974	414	388	10
5	5974	2074	388	51
7	5974	2904	388	71
15	5974	6222	388	153

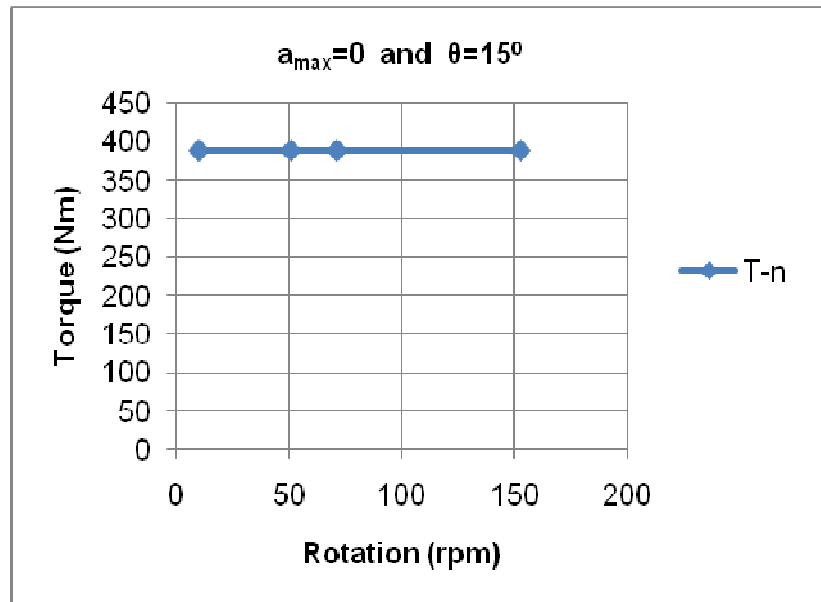


Figure 101 Torque-Rotation Graphic for the Position-1

In the Figure given above, constant torque value is required while the vehicle is going on uphill. In the next case, minimum torque and rotation range are going to be examined while the vehicle is on straight road with fully loaded container.

Table 32 Position-2: ($a_{\max}=2.5$), ($V_{\max}=15$), Gradient ($\Theta=0^\circ$)

$V_{\max}$ (km/h)	TOTAL FORCE	POWER (W)	TORQUE (Nm)	ROTATION (rpm)
1	5893	409	383	10
3	4242	883,	275	30
7	2592	1260	168	71
15	1491	1553	96	153

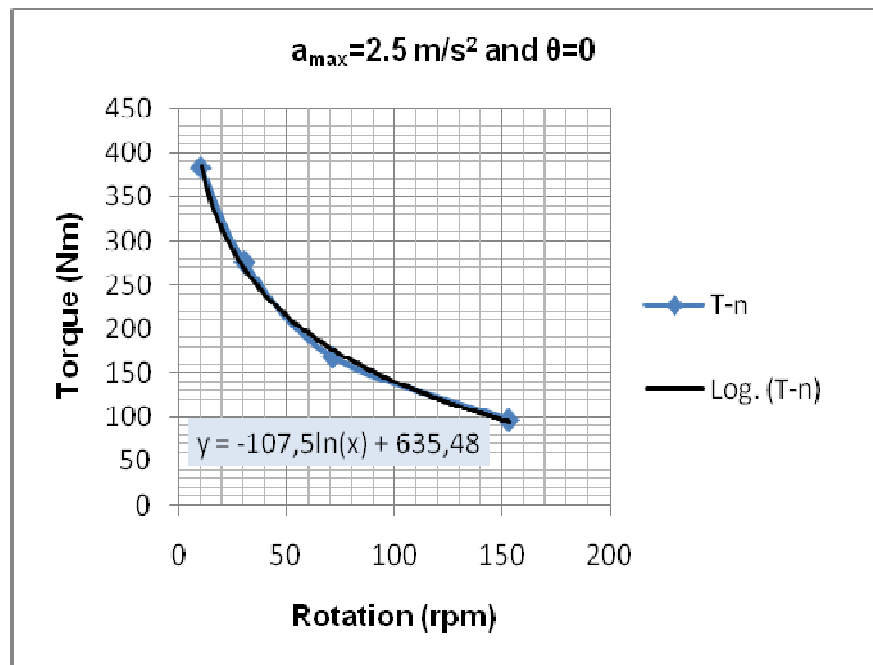


Figure 102 Torque-Rotation Graphic for the Position-2

In respect of the result tables and graphics given above, it is obtained that minimum torque value is 400 Nm and the rotation is in the range of 0-150 rpm. By the aid of these results, hydro-motors constituting the traction system are started to be determined.

6.2. Hydraulic Equipments

After determining the required torque and power, hydrostatic pump driven by an electric motor (with 15 kW, 2000 rpm and Brushless DC) is chosen. Main properties of the hydrostatic pump are given in the following table.

Table 33 Properties of the MD10V-28 Hydrostatic Pump (Samhydraulik)

SERIE/SERIES		MD10V 21/28
<i>Cilindrata⁽¹⁾</i> Displacement ⁽¹⁾	<i>cm³/giro</i> [in ³ /rev]	21-28 [1.28-1.71]
<i>Cilindrata pompa di sovralimentazione</i> Charge pump displacement	<i>cm³/giro</i> [in ³ /rev]	11 [0.671]
<i>Regime massimo di rotazione⁽²⁾</i> Max speed ⁽²⁾	<i>giri/min</i> rpm	3600
<i>Regime minimo di rotazione</i> Min speed	<i>giri/min</i> rpm	700
<i>Pressione nominale</i> Rated pressure	bar [psi]	Versione in Alluminio (fuori produzione) Aluminium version (out of production)
		250 [3625]
<i>Pressione di picco</i> Peak pressure	bar [psi]	Versione in Ghisa Cast iron version
		250 [3625]
<i>Pressione di sovralimentazione</i> Charge pressure	bar [psi]	Versione in Alluminio (fuori produzione) Aluminium version (out of production)
		300 [4350]
<i>Pressione massima in carcassa</i> Max case pressure	bar [psi]	Versione in Ghisa Cast iron version
		350 [5075]
<i>Pressione di aspirazione</i> Suction pressure	bar [psi]	15÷25 (standard 20) [218÷363] [Standard 290]
<i>Momento d'inerzia parti rotanti</i> Moment of inertia rotating parts	kg m ² [lbf·ft ²]	2 [29]
<i>Massa⁽³⁾</i> Weight ⁽³⁾	kg [lb]	≥ 0.8 [≥ 11.6]
		0.0018 [0.042]
<i>Massa⁽³⁾</i> Weight ⁽³⁾	kg [lb]	Versione in Alluminio (fuori produzione) Aluminium version (out of production)
		13 [28.65]
<i>Massa⁽³⁾</i> Weight ⁽³⁾	kg [lb]	Versione in Ghisa Cast iron version
		22 [48.48]

This hydrostatic pump, which is just for the back and forward traction system (four individual hydromotors assembled inside of the wheel rims), reaches to 56 lt/min at 2000 rpm. Besides the traction system, three pumps mounted to the same shaft with the hydrostatic pump which are for the cleaning system (brooms, fans and elevator) and all movement pistons (steering, covers on container, up-down of brooms and elevator).



Figure 103 MD10V-28 Hydrostatic Pump (Samhydraulik)

Speed of the vehicle is approximately 15 km/h and to provide this speed, hydromotors need to be rotated at 145 rpm. Therefore, appropriate motors are chosen and mounted to the inside of the wheel rims.

Table 34 Properties of the Wheel Type Hydromotor (HR L130)

Motore Motor	<i>Cilindrata</i> Displacement cm ³ /giro [in ³ /rev]	<i>Pressione diff. max.</i> Max. differential pressure bar [psi]		<i>Coppia max.*</i> Max. torque*		<i>Portata max.</i> Max. flow		<i>Velocità max.</i> Max. speed		<i>Potenza max.</i> Max. power	
				Nm [lbf-ft]		l/min [U.S. gpm]		[rpm]		kW [hp]	
HR 130	125.7 [7.66]	Cont	205 [2972]	Cont	375 [276]	Cont	75 [19.8]	Cont	596.5	Cont	23.4 [31.3]
		Int ¹⁾	310 [4495]	Int ¹⁾	545 [401]	Int ¹⁾	95 [25]	Int ¹⁾	755.5	Int ¹⁾	43.1 [57.7]
		Peak ²⁾	310 [4495]								

Due to the stock and delivery conditions of the supplier hydraulic company, HR 100 and HR 160 motors are used on prototype vehicle but it is decided to use HR 130 motor during mass production of the vehicle.



Figure 104 Wheel Type Hydromotor

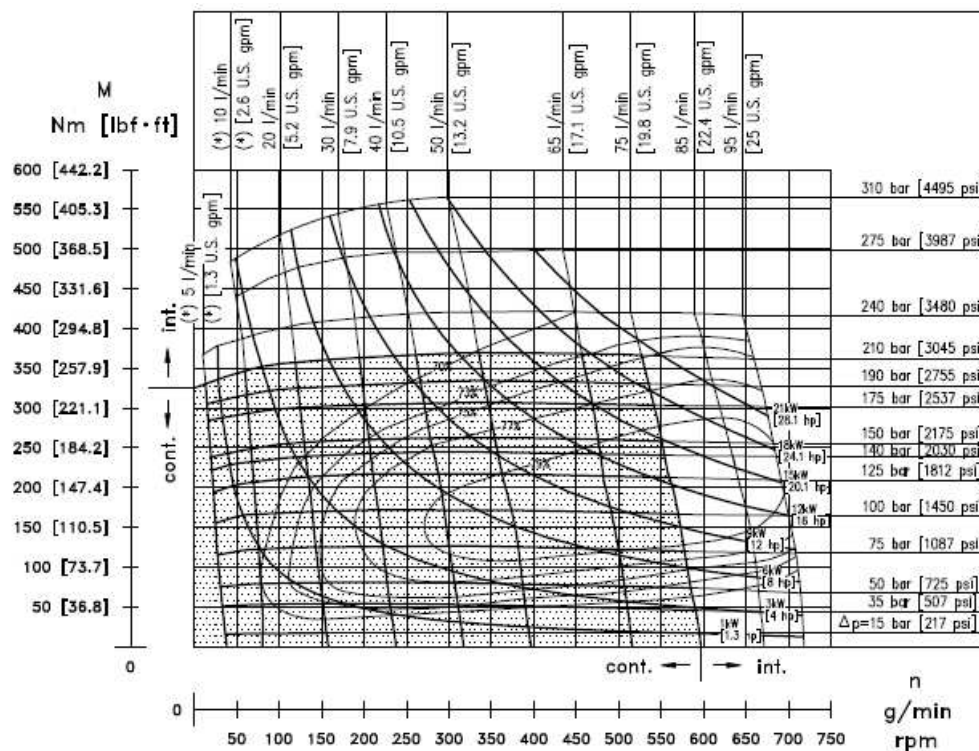


Figure 105 Characteristics of the Hydromotor

Overall hydraulic motor types and the hydraulic design schematic are given in the appendix section.

CHAPTER 7

ACCESSORIES

7.1. Vacuum Hose

When the cylindrical sweeping brushes and brooms are inadequate or cannot reach the edges of road and pavement areas to collect trashes, vacuum hose and connection adaptors are used. For this purpose, the literature survey is completed for similar usages. Then, the CFD analyses are run for different diameters and lengths by Punto Engineering.

Prior to the CFD analysis, for different usage patterns (MADVAC different models), observations are done. The vacuum arm is used in vehicles such as small and medium-sized cars also used in large tonnage scavenging.



Figure 106 Portable Vacuum Hose (MADVAC CN100)



Figure 107 Collecting the Trashes with Vacuum Hose (MADVAC) [12]



Figure 108 Elastic Vacuum Hose (MADVAC 101)

As a result of the investigations, new tools (vacuum hose and carrier arm) are designed to be used in new vehicle. The portable elastic vacuum hose is decided to be attached to the top of the trash container. When it will be used, hose is plugged to the valve (assembly adaptor at back side of the container) and vacuuming is realized. Due to determining of the optimum hose diameter and length, CFD analysis was carried out as well.

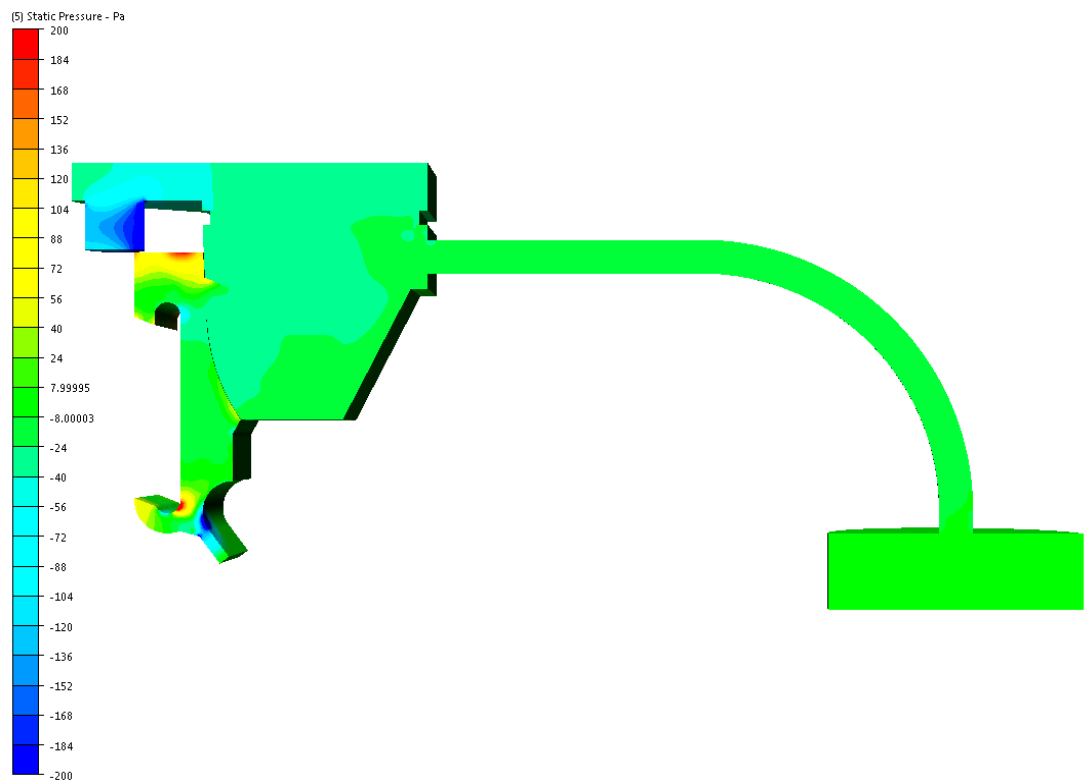


Figure 109 Pressure Distribution of the Section

The bigger volume trashes (such coke or water bottles) are considered to be collected by the vacuum hose. Therefore, diameter of the hose is chosen according to these needs and analyses are carried out to satisfy the requirements such pressure and velocities as well.

As a result, in the range of 100-180 mm of diameter (optimum and exact diameter will be determined according to the results of proto-type tests) and 4.5 m of length is agreed on the vacuum hose dimensions.

7.2. Power LED

Nowadays, usage field of the LED lighting is extremely increasing due to their efficiency and little power consumption. From home lighting to aircraft industry, LED lighting takes a very important place in the lighting industry.

A light-emitting diode (LED) is a semiconductor light source. LEDs are used as indicator lamps in many devices and are increasingly used for other lighting. Introduced as a practical electronic component in 1962, early LEDs emitted low-intensity red light, but modern versions are available across the visible, ultraviolet and infrared wavelengths, with very high brightness.

In 2011, LEDs have become more efficient, so that a 6 Watt LED can easily achieve the same results. A standard 40 W incandescent bulb has an expected lifespan of 1,000 hours while an LED can continue to operate with reduced efficiency for more than 50,000 hours, 50 times longer than the incandescent bulb.

One kilowatt-hour of electricity will cause 1.34 pounds (610 g) of CO₂ emission. Assuming the average light bulb is on for 10 hours a day, one 40-watt incandescent bulb will cause 196 pounds (89 kg) of CO₂ emission per year. The 6-watt LED equivalent will only cause 30 pounds (14 kg) of CO₂ over the same time span. [23]



Figure 110 LED daytime running lights of Audi A4 [23]



Figure 111 LED Aircraft Landing Light [24]



Figure 112 LED Aircraft Sidewall Light [24]

New Electro-hydraulic vacuum sweeper (VSM 080) consists of many technologic improvements and firsts as well. LED lighting is one of them.

By the completion of manufacturing of the main subsystems of sweeper, chosen LED lighting equipments are going to be fastened to the electronic (automation PLC) system. Overall schematic is given in the appendix section representing input-output voltages, electric-electronic components and any other detailed data.

CHAPTER 8

DISCUSSION OF RESULTS

Throughout the thesis study, the other vacuum sweepers available in the market were investigated and especially their cleaning systems were examined. It was observed that the centrifugal type fans are preferred and vehicles are driven by diesel engine. Therefore, they are not environment-friendly. Air and sound pollutions occur. Development of the efficient cleaning system (mechanical and vacuum) is the main purpose of the new sweeper design.

For the elevator design, overall calculations were done. So that the critical parameters such angular velocity of pulley, length of belt, number of blades and linear velocity were also specified. 3D model of the system was modeled and manufacturing was completed before the experiments as well. Aluminum sheets (as protective side sheets) and brushless electric motor were preferred initially. By the change of the system (to electro-hydraulic) belt material (textile), blade material and motor were all changed.

The other part of the cleaning system, vacuum system, consists of two equivalent centrifugal fans with the diameter of 400 mm, 10 blades (straight type). Before determining of these new fans, current MÜSAN fans (250 mm, 8 curved type blades) were also examined and overall characteristics were determined by setting up an experiment set-up according to the AMCA standard.

After specifying the requirements of the cleaning system (flow rate, pressure) using CFD analyses, calculations (pressure and flow rate, power, vector triangle speeds) were done concerning different rotor diameters to find up the appropriate rotor diameter and number of blades, considering the power limitations of battery capacity as well.

Two different experimental set-ups were designed and manufactured for testing of the former and new fans. Overall measurements were taken through the cross-section of the pipe and performance curves were drawn by the aid of these data. Performance experiments have enabled the comparison of test results with previous theoretical calculations.

Various types of fans (with different blade angles and different number of blades) were manufactured and tested on another set-up which was manufactured for the higher capacity fans. Performance and efficiency curves were determined and comparison of the former fans and new fans (at different rpms for individual rotors) were given on graphics.

In the next process, the operation point of the vacuum system was obtained after the completion of the manufacturing of the other subsystems before the system experiments. For this purpose, two equivalent fans were mounted inside the assembly box and then this box was assembled onto the sweeper. Due to some reasons such loss of air, friction, resistance of air and sound; the assembly (air mixture) box was cancelled and fans were directly connected to the outlet of the container with flexible hoses. These two individual situations represent that the second connection type (which is connected directly) gives more efficient results. We obtain higher vacuum.

Beside the cleaning system (elevator and fan system), other subsystems were also manufactured and assembled to the chassis before the experiments. Air leakages were avoided as much as possible and overall test results (with elevator, without elevator) recorded.

Due to the torque problem of hub motors were assembled to the wheel rim initially, system was changed from electric to the electro-hydraulic. For this purpose, hydrostatic and tip pumps, pistons of the brooms and container and hydraulic fan motors as well, were all determined. Schematic of the hydraulic system is given in the appendix section.

Using the parameters such mass, gradient and rolling resistances, acceleration and inertia forces, requirements of the hydraulic motors were calculated and torque-rotation graphs were stated. Two different cases were considered basically. In first case, 15° of gradient and zero acceleration are taken and for the second case, zero gradient and accelerated movement values are taken with the maximum speed of 15 km/h of the sweeper.

The results represent that the motor power should be around 15 kW as the trash container is full and while the other subsystems (fans, brooms etc.) are all running. So therefore, the electric motor (15 kW, DC, PM) which runs the hydrostatic pump at 2000 rpm, was specified and then required flow rate of hydraulic oil to be pumped (hydrostatic and tips) and displacement of motor (flow per minute) were determined as well.

Beside the hydraulic piston of container, to lift the elevator system, main and front brooms, mechanisms were used which consist of hydraulic pistons as well. To provide of the movement of the front brooms through the horizontal axis (x), hydraulic pistons were preferred. The front brooms can reach to the side of the pavements for directing the trashes to the middle of the sweeper. The larger trash will be collected by the elevator and dust is vacuumed by the fans.

CHAPTER 9

CONCLUSION

Since customer demands are rising on sweeper vehicles generating less dust particles, less air pollution (health), less sound and optimum costs basically, Sweeper manufacturers are working on the 'Cleaning System' especially to gain active role on market and also to supply customer demands.

Within the scope of this study, environment-friendly vacuum sweeper was designed and manufactured. Traction and cleaning (elevator and vacuum) systems of the sweeper (MÜSAN VSM 080) are run with hydraulic motors and hydraulic system (hydrostatic and tip pumps) energized from the batteries. The hydraulic system works as follows: the electric motor rotates the pump at 2000 rpm. Tip pumps which are on the hydrostatics pump distribute the oil to the subsystems such elevator system, main and front brooms, fans and hydraulic pistons.

It is essential to design an efficient and powerful cleaning system (fan and elevator system) for a street sweeper. Therefore, by following all of the steps such literature survey, calculations, analysis, experiments and manufacturing, efficient cleaning system of the new sweeper (MÜSAN VSM 080) has been revealed. After the completion of manufacturing of the prototype vehicle (with all accessories), mass production will begin by fixing the possible problems.

Unlike many industrial type vacuum sweepers, new sweeper has electric motor. It is also possible to use of solar energy to charge batteries in the future. In addition to this, LED lighting will be used for the interior and exterior lighting of the sweeper. From the point of view of the cleaning vehicle sector, new electro-hydraulic sweeper is adding value for the native and foreign industrial areas.

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APPENDIX A

OVERALL DIMENSIONS AND TECHNICAL DRAWINGS

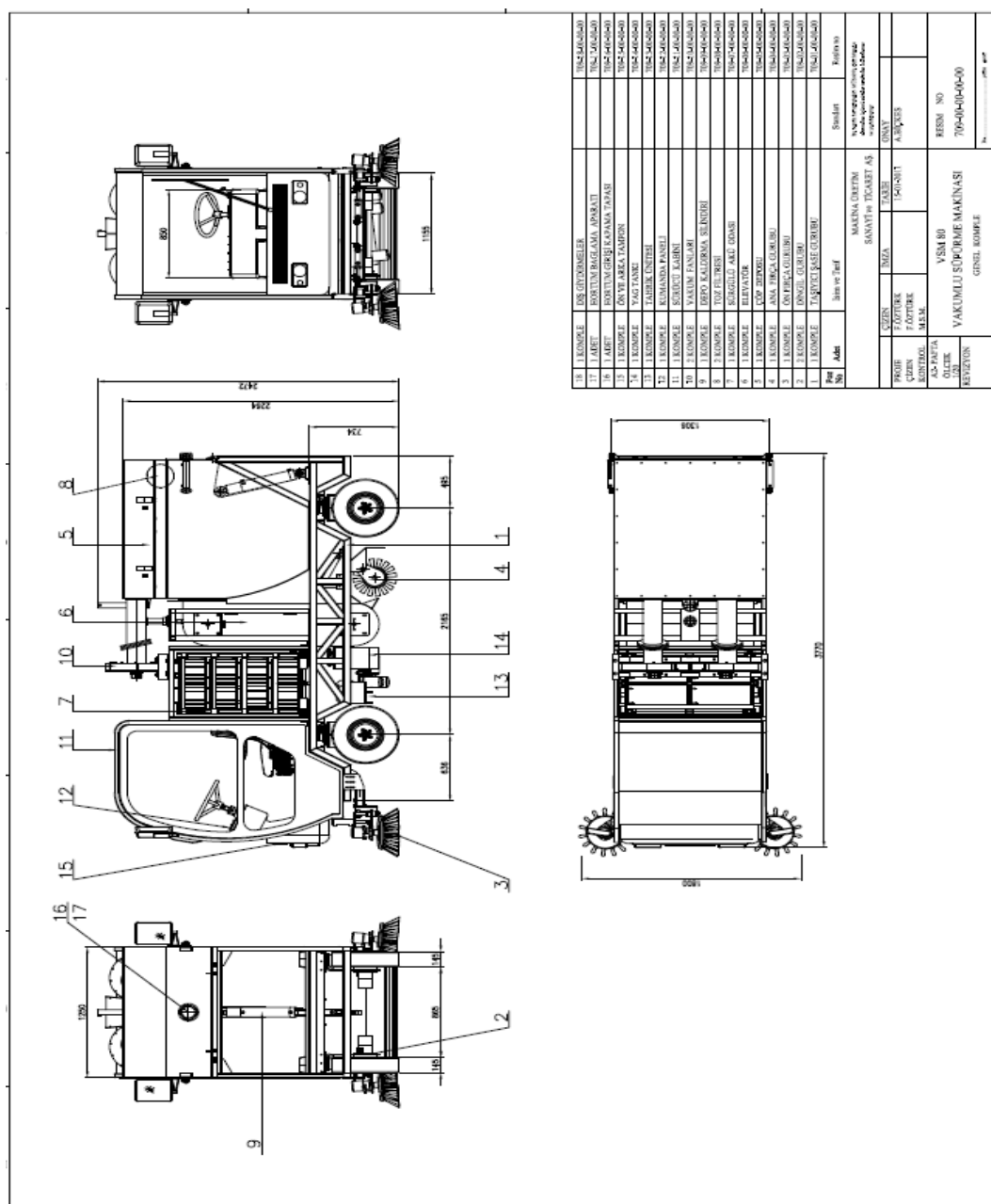


Figure 113 Overall Dimensions of the New Müsan Sweeper (VSM 08)

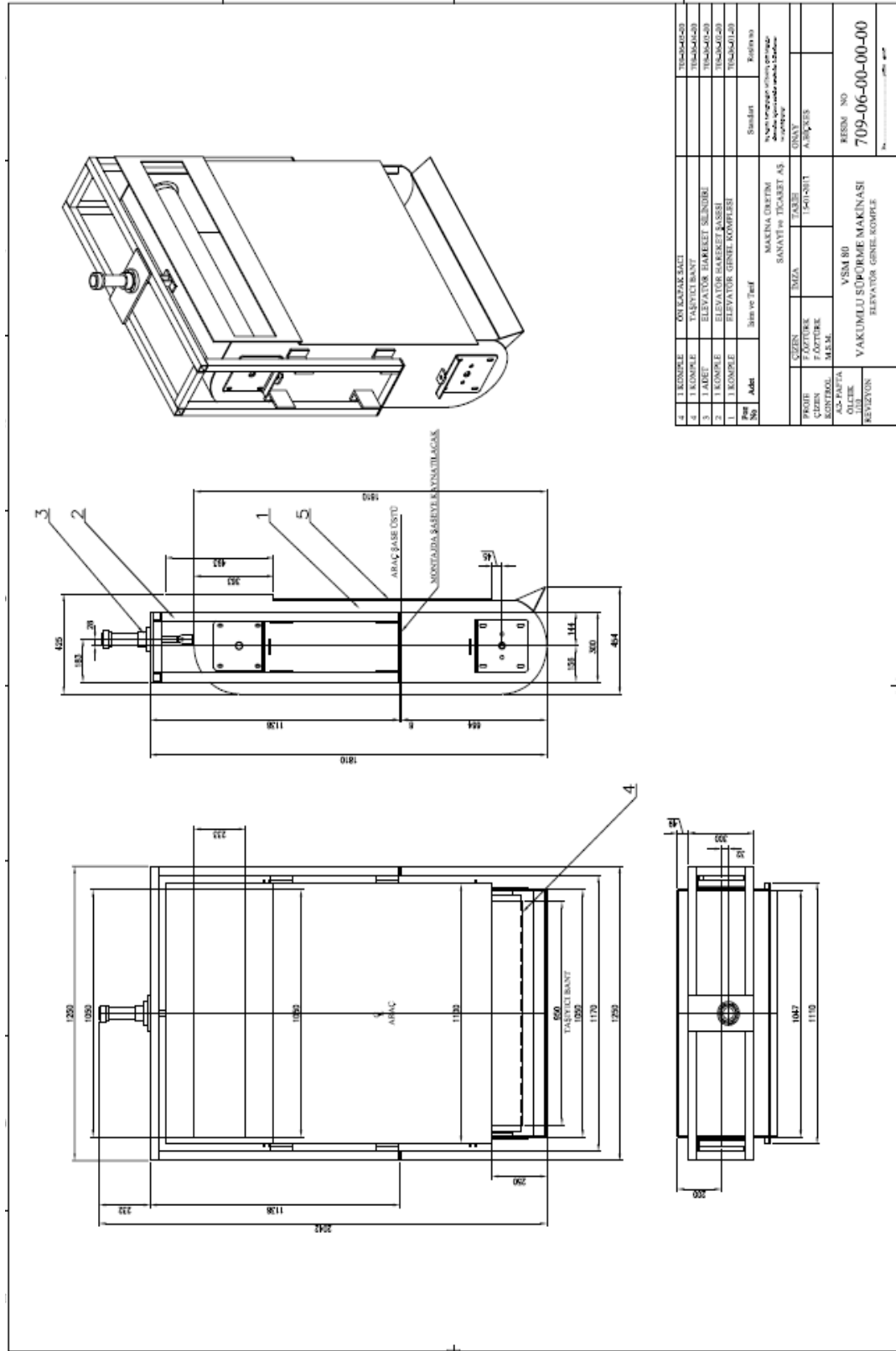


Figure 114 Elevator System of the New Müsan Sweeper (VSM 08)

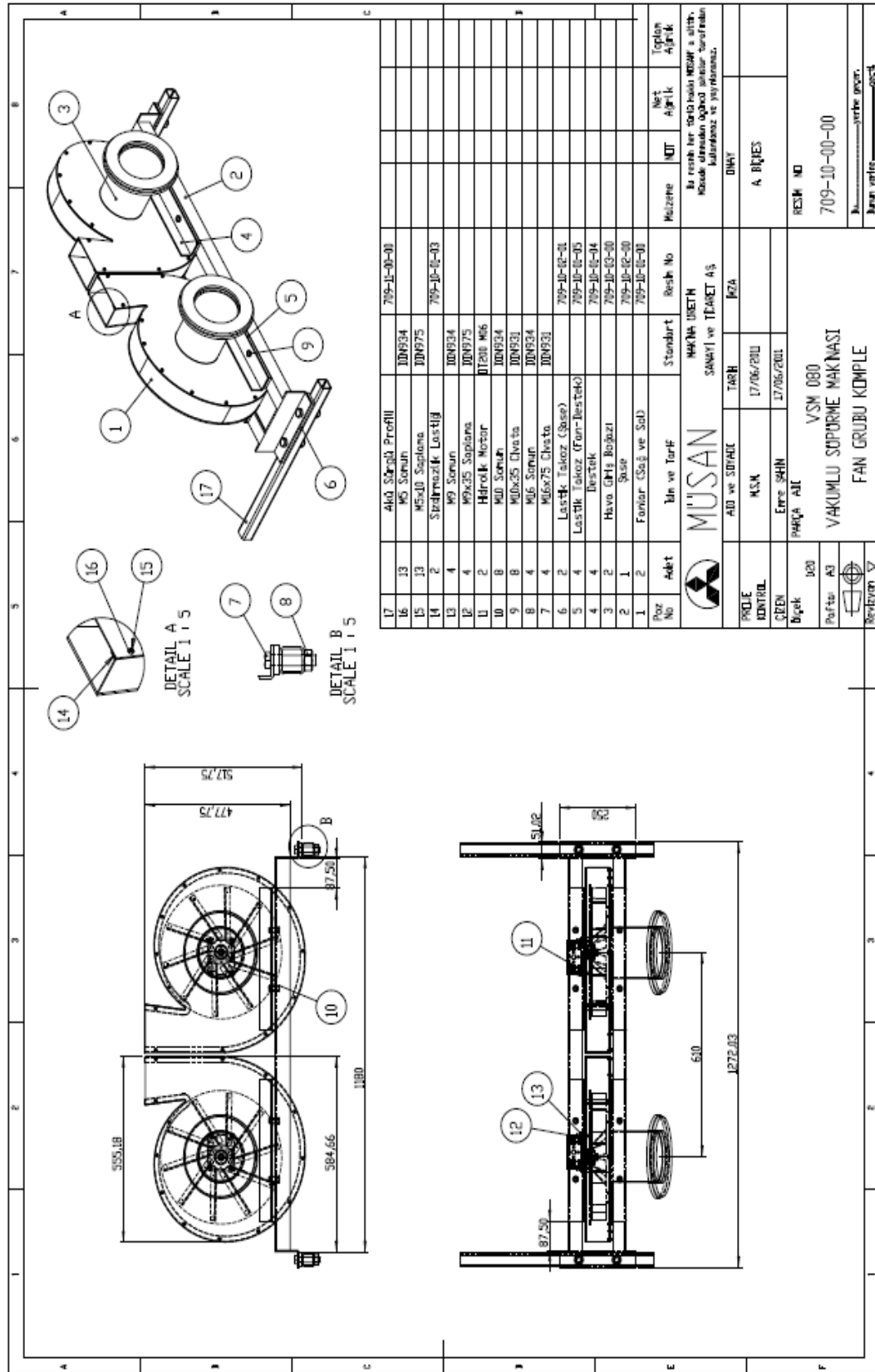


Figure 115 Fan System of the New Müsan Sweeper (VSM 08)

APPENDIX B

HYDRAULIC SYSTEM SCHEMATIC

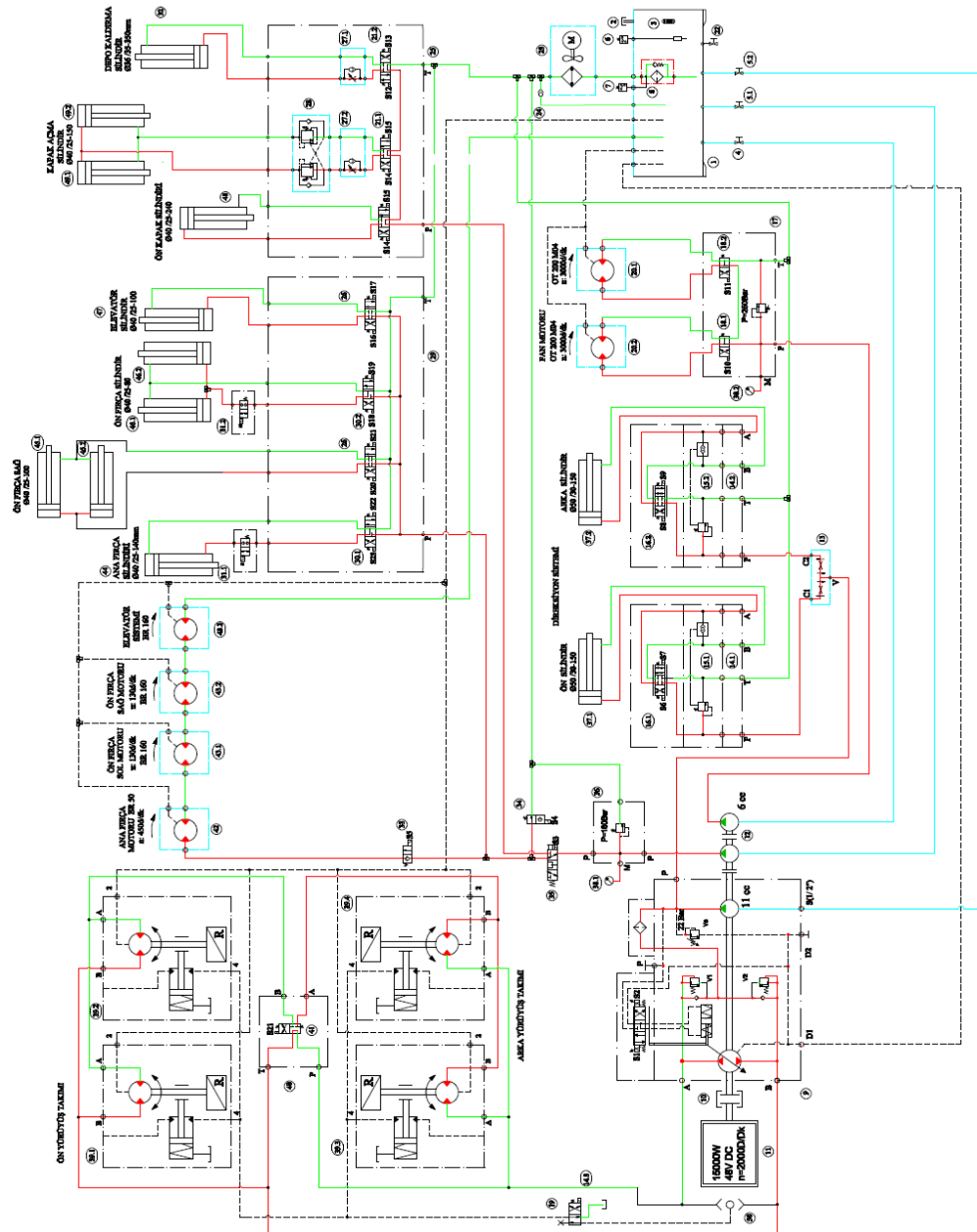


Figure 116 Hydraulic System of the New Müsan Sweeper (VSM 80)

APPENDIX C

HYDRAULIC SYSTEM COMPONENTS

Table 35 Equipment List to be used in the Hydraulic System of VSM 08

FİRMA :MÜSAN		PARÇA ADI KONTROL
TARİH :01/04/2011		
PROJE : UĞUR TOSUN		
ÇİZEN : MUSTAFA ERKEN		
1	ÖZEL YAĞ TANKI	1
2	MINTOR TCAR 4080 DEPO KAPAĞI	1
3	MINTOR FLT-2 SEVİYE GÖSTERGESİ	1
4	PAKKENS 3/ 4" EMİŞ VANASI	1
5	PAKKENS 1" EMİŞ VANASI	2
6	FOX SEVİYE ŞALTERİ	1
7	FOX K 4S NA 1/ 8" KIRILIK ŞALT.	1
8	OMT 11/ 2" DÖNÜŞ FİLTRESİ	1
9	SAMHYDRAULİK MD10V 28 HER KON	1
10	KAMPANA - KAPLIN	1
11	15000W m2000rpm 48V DC ANA TAH.	1
12	OT 200 P11+6 TANDEM DİŞLİ POM.	1
13	LUEN VDF/ 38-C AKIŞ BÖLÜCÜ	1
14	NG 6 BAS. EMNİYETSİZ ÇELİK PLETT	3
15	ARON NG 6 AM3H3VP1 HYDROSTAT	2
16	ARON NG 6 XDP3C01 24V DC VALF	2
17	OMT NG 6 2" Lİ BAS. EMN. SERİ BLOK	1
18	ARON NG 6 TB HM 24V DC SEL. VALF	2
19	ARON NG 6 TB DÜZ ÇAPRAZ 24 V	1
20	OT 200 M04 HİDROMOTOR	2
21	ARON NG 6 AM 24V DC SEL. VALF	2
22	PAKKENS 3/ 4" BOŞALTIMA VANASI	1
23	OMT SS40 24 VDC SOĞUTUCU (MÜŞTERİDEN)	1
24	BLEFFE 1" 6 BAR ÇEKVALF	1
25	OMT NG 6 3" Lİ BAS. EMNİYETSİZ SERİ	1
26	ARON NG 6 KM 24V DC SEL. VALF	1
27	ARON NG 6 MOD.A HATTI ÇEKİLİ HIZ.	2
28	LUEN NG 6 MOD.İKİZ OVC	1
29	OMT NG 6 4" LÜ BAS. EMN.SİZ SERİ BL.	1
30	ARON NG 6 ABT 24 V DC SEL. VALF	2
31	ÖZEL YÜZME BLOĞU	2
32	Ø63 /35-350mm DEPO KALDIRMA SİL.	1
33	LUEN 3/ 8" NK 24V DC POPET VALF	1
34	LUEN 3/ 8" NA 24V DC POPET VALF	1
35	WALVOIL3/ 8" DFB052/ 3 24V DC 3 YOLLU ELEKTRİK Lİ VALF	1
36	OILCOMP 3/ 8" BAS. EMNİYET VALFİ	1
37	Ø50/ 30- 150 DİREKSİYON SİLİNDİRİ	2
38	PAKKENS Ø63*315 ALT GLİS. MAN.	2
39	SAM. HRL130 CN 320 YÜR. RED.	4
40	NG10 BASINÇ EMNİYETSİZ PLETT	1
41	ARON NG 10 TB AM 24V DC SEL. VALF	1
42	SAMHYDRAULİK HR 50 HİDROMOT.	1
43	SAMHYDRAULİK HR 160 HİDROMOT.	3
44	Ø40 /25-140mm ANA FIRÇA SİLİNDİRİ	1
45	Ø40 /25-100mm ÖN FIRÇA SAĞ	2
46	Ø40 /25-80mm ÖN FIRÇA SİLİNDİRİ	2
47	Ø40 /25-100mm ELEVATÖR SİLİNDİRİ	1
48	Ø40/ 25-240mm ÖN KAPAK SİLİNDİRİ	1
49	Ø40/ 25-150mm KAPAK AÇMA SİLİNDİRİ	1
50	FERRO 1/ 4" VEYA VALFİ	1

APPENDIX D

ELECTRIC SYSTEM (PLC) SCHEMATICS

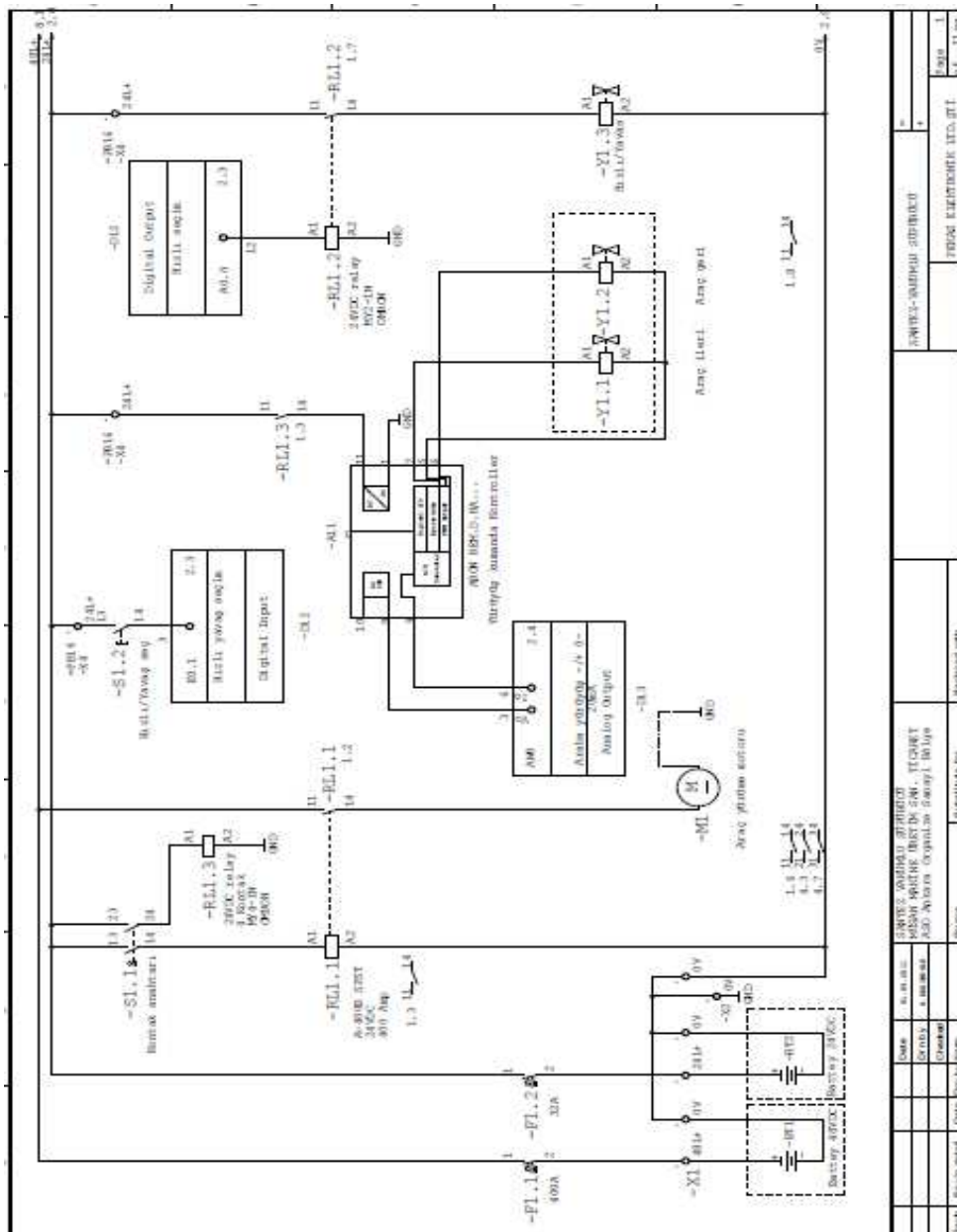


Figure 1.17 Electric System of the New Müsan Sweeper (VSM 08)

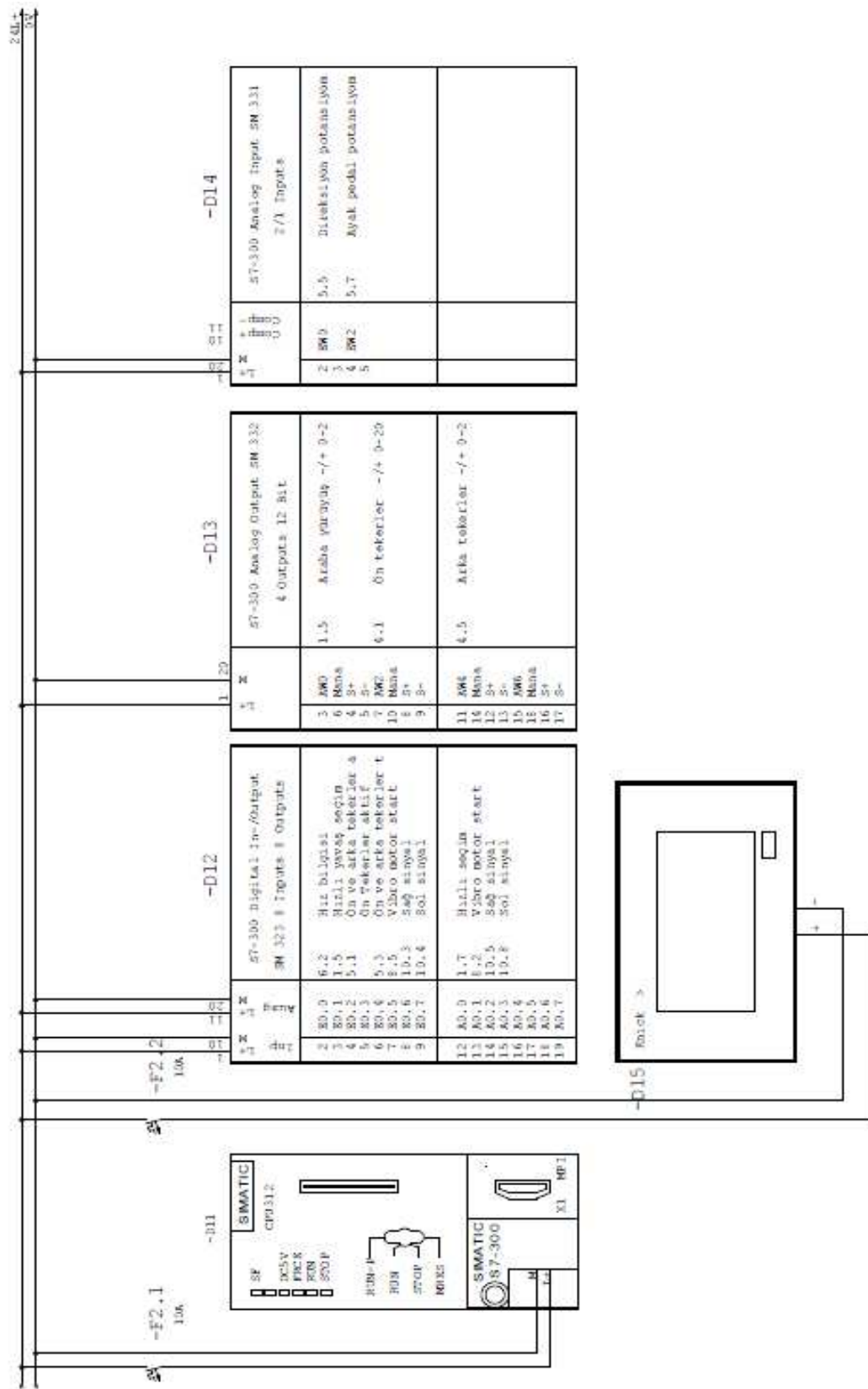


Figure 118 Electric System Inputs/Outputs and LCD Screen

APPENDIX E

MANUFACTURING PHOTOS OF THE NEW SWEEPER



Figure 119 Prototype Vehicle (VSM 080)



Figure 120 VSM 080 Sweeper before the External Cladding and Accessories