Palestine Polytechnic University

Collage of Engineering & Technology
Mechanical Engineering Department

Bachelor's Thesis

Graduation Project

Running vehicle alternator by using exhaust gas stream

Project Team

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Abstract

The emitted exhaust gas from an engine is a lost power, it can be used to generate electrical power for a vehicle, upon demand; the exhaust gas can power a turbine which in turn will drive an alternator to generate electricity for the vehicle's electrical loads, and consequently reduce the load on the engine.

The basic principle of the system is to utilize the high pressure of the exhaust gases that is directed through a nozzle which increases the velocity of these gases. This high gas velocity is impinging the turbine rotor which rotates at high speed. This rotor is connected to the alternator shaft via a V-belt which used to transmit the power of the running turbine to the alternator.

The turbine shaft and the V-belt are designed to transmit a maximum power produced from the VGT turbocharger without any problems.

The power produced from the turbine is measured at different engine speeds. This power is not enough to run the alternator and to produce needed electric current to supply all vehicle electric loads. The power produced can be used as assistant to the power source in the vehicle.

ملخص

اظهرت النتائج العملية ان حركة الشاحن التوربيني في السيارة الناتجة عن حركة تيار غازات العادم يمكن الاستفادة منها في تحريك مولد جهد، يمكن الاستفادة منه في توليد طاقة كهربائية يتم تزويدها للأحمال الكهربائية في السيارة ، بهدف تخفيف الحمل عن المحرك وبالتالي تقليل استهلاك الوقود.

يقوم المبدأ الأساسي للنظام على الاستفادة من ارتفاع ضغط غازات العادم، التي يتم توجيهها من خلال ريش موجودة داخل الشاحن التوربيني (VGT) والتي بدورها تزيد من سرعة هذه الغازات، هذه السرعة العالية لغازات العادم تعمل على تحريك التوربين الذي بدوره يتصل بمولد الجهد بواسطة قشاط من نوع (V-belt)، الامر الذي يعمل على تحريك مولد الجهد، وبالتالي توليد طاقة كهربائية.

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CHAPTER ONE

INTRODUCTION

Contents:

1.1 General outlook

The requirement of reduction in fuel consumption is needed every hour and accordingly automobiles are so designed for minimum fuel consumption in the last few decades.

If this will be done in different aspects, fuel consumption can be decreased dramatically. Therefore it is a very real need to reduce the load on the engine to decrease the fuel consumption.

In conventional automobiles and trucks, many accessories are driven by a series of belts connected to the crankshaft of the engine. Such accessories include alternator, water pump, air conditioning compressor, power steering pump, and secondary air pump. These loads present a significant drain of engine power. An alternative means of powering these devices would decrease the load on the engine and improve its performance.

One of the most important devices which can reduce the load on the engine is the alternator.

As seen in fig.1.1, around sixth of the energy lost in the exhaust gases can be recovered with a turbo generator.

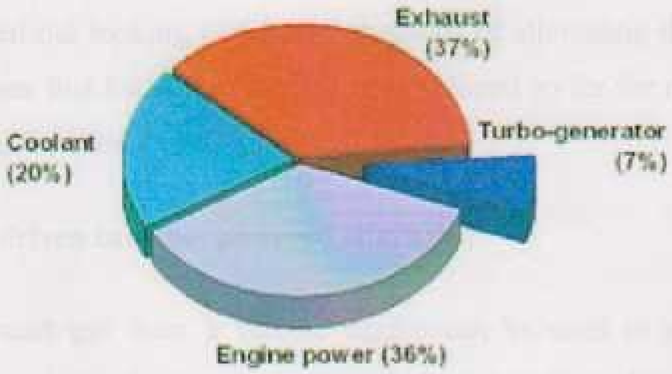


Figure (1.1) Around sixth of the energy lost in the exhaust gases can be recovered with a turbo generator.

1.2Objectives of Project

The main purpose of this project is to reduce the load on the engine in order to decrease the fuel consumption and increase the service life of the engine at the same time. There are many devices connected to the crankshaft of the engine by belts and rotate with it. One of the main devices that is running by the crankshaft of the engine is the alternator.

The basic principle of this project is to use the energy of the exhaust gas to drive a variable geometry turbocharger (VGT) and use the rotation of the turbine to drive the alternator.

The importance of this project can be summarized in the following points:

1- Reduce fuel consumption.
2- Reduce load on the engine.
3- Taking advantage of the exhaust gases.

1.3 Literature Review

The present works focusing on reducing the load on the engine by finding an alternative power source for running the alternator, which will lead to reduce the fuel consumption. This is a major issue in the world of the automobile industry. Studies have been carried out looking at different methods of alleviating this problem. The various techniques that have been studied or developed so far for reducing the load on the engine can be classified as follows:

1.3.1 Exhaust- driven turbine- powered alternator

The exhaust gas from a vehicle engine can be used to provide electrical power. The exhaust gas drives one or more turbines positioned in the exhaust line which in turn drives an alternator. A controlled bypass gate and passage flanking the turbine can selectively passes a desired portion of the exhaust gas through the turbine which will balance passes around the turbine.

An automotive exhaust system having an exhaust gas powering electrical system, the system is powered by the exhaust gases which leave the engine through an exhaust manifold. The exhaust system is one of the single-pipe exhaust variety, but it could also be a dual pipe system or some other configuration. The engine can be an internal combustion device or some other engine that produces an exhaust gas that will drive a turbine.

The manifold is connected to a bypass gate valve that can channel the exhaust gas from the manifold through a turbine input manifold to a turbine and a turbine bypass. It should be understood that anywhere from zero to 100% of the exhaust gas can be directed by the bypass gate valve through the turbine. Powered by the exhaust gas, the turbine drives an alternator and the exhaust gas exits the turbine through the turbine exhaust.

The turbine exhaust and the exhaust bypass are both connected to an exhaust pipe leading perhaps to a muffler. The turbine can be sized to maximize its output over the desired operating range of the engine.

The bypass gate valve could be controlled by a bypass controller responsive to the speed of the alternator and perhaps the load conditions. The speed can be measured as a function of the alternator shaft speed or the alternator output voltage. If the turbine speed is too high to properly run the alternator, a gear reduction unit could be inserted between the turbine and the alternator. In most application, the desired output would be direct current [1].

1.3.2 Exhaust- driven electric generator system for (ICE)

An electric generator system drivable by the energy of the exhaust gases emitted from an internal combustion engine includes two turbines drivable by the energy of exhaust gases from the internal combustion engine. The turbines having different capacities, and an electric generator mounted on a turbine shaft interconnecting the turbine impeller of the turbines. When the energy of exhaust gases emitted from the engines is larger, the turbine of the larger capacity is driven. When the exhaust energy is smaller, the turbine of the smaller capacity is driven [2].

This project uses the exhaust gases pressure for running the alternator, this pressure is being used and sent through a nozzle or set of nozzles and generates high velocity gasses at exit. This high velocity gasses is being utilized to drive a turbocharger, which will be coupled with alternator, so turbine drive the alternator instead of powering it by the crankshaft, and consequently reduce the load on the engine.

1.4 Project Scope

Chapter one: introduction

This chapter gives an introduction about the system; it clarifies the system as a practical idea and the time table.

Chapter two: system operation principle

This chapter includes the description of the project, and its operation principle.

Chapter three: shaft design

This chapter includes the design of the turbine shaft, which will use to transmit the turbine power to drive the alternator.

Chapter four: power transmission

This chapter includes the design of a belt which will be used to transmit power from turbine to the alternator, and design the required pulleys dimension.

Chapter five: power of the turbine

According this chapter we can calculate the power of the turbine by using different ways, such as dynamometer, electrical alternator, and turbine efficiency. Also it include the system main components.

Chapter six: conclusions recommendations and problems

1.5 Time table

The time schedule shows the stages of developing in our work and the process of project growth for the first semester and the second semester.

Table (1.1) project time-schedule for first semester

Week / Process	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
Collecting data and literature																
Analyzing of data																
Modeling																
Simulation																
Writing the documentation																
Final presentation																

Table (1.2) project time-schedule for second semester

Week / Process	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
Preparing the combustion engine																
Buying mechanical and electronic part																
Building experimental setup																
Chick the project part and perform the initial experiment																
Perform final experiment																
Writing the documentation																
Final presentation																

CHAPTER TWO

SYSTEM OPERATION PRINCIPLE

Contents:

2.1 Introduction

As already known, the internal combustion engine (ICE) lose approximately 40% of its energy through the exhaust. This lost energy can be used to increase the performance of the internal combustion engine by using several methods. One of the better methods of utilizing the exhaust is being achieved by this project. By using the exhaust gas pressure, this pressure is being used and sent through a nozzle or set of nozzles and generates high velocity gasses at exit. This high velocity gasses is being utilized to drive a turbine (turbine coupled with alternator by a shaft) so turbine drives the alternator instead of the crankshaft, and consequently reduces the load on the engine.

2.2 Description of Project

The engine has an exhaust manifold connected to a variable geometry turbocharger (VGT) positioned in the exhaust line which in turn drives an alternator. As the movable vanes in a VGT are adjustable, the angle of the air flow onto and against the turbine blades changes, which alters the effective area of the turbine and the turbocharger aspect ratio, besides being able to vary air pressure regardless of engine speed.

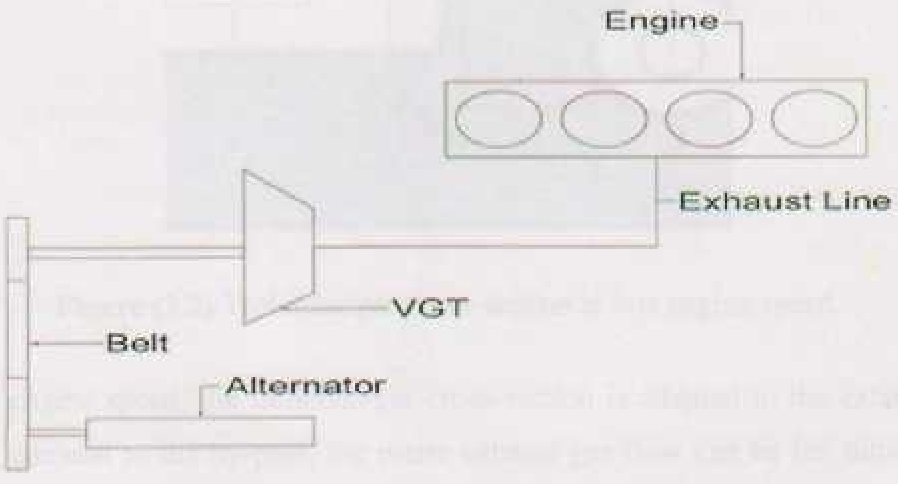


Figure (2.1) Schematic illustration of the system

The movable vanes in a VGT could be changed responsive to the speed of the alternator. The speed can be measured as a function of the alternator shaft speed and the alternator output voltage.

2.3 Operation principle

An electric alternator system drivable by the energy of the exhaust gases emitted from an internal combustion engine includes VGT drivable by the energy of exhaust gases from the internal combustion engine, and an electric alternator mounted on a turbine shaft.

At low engine speed and high charge pressure, the cross-section of the exhaust gas flow is narrowed upstream of the turbine wheel by means of vanes. Since the exhaust gas is forced to pass through the restricted cross-section more quickly, the turbine wheel rotates faster (see fig.2.2). The high turbine speed at low engine speed generates the require power to drive the alternator [3].

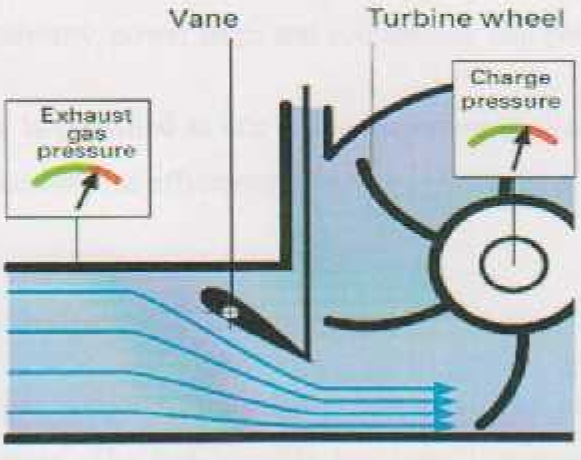


Figure (2.2) Turbocharger cross-section at low engine speed

At high engine speed, the turbocharger cross-section is adapted to the exhaust gas flow. In contrast to the by-pass, the entire exhaust gas flow can be fed through the turbine in this way. The vanes free a larger inlet cross-section, thereby ensuring that the required charge pressure is not exceeded (see fig.2.3).

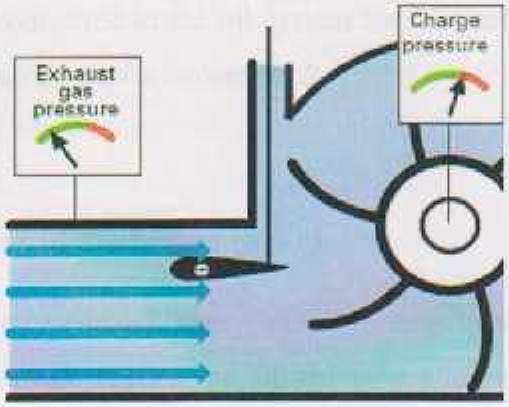


Figure (2.3) Turbocharger cross-section at high engine speed

2.4 Turbocharger

In general, a turbine is one of various rotary machines that convert the kinetic energy in a stream of fluid (gas or liquid) in to mechanical energy by passing the stream through a system of moving fans or blades. Turbines are simple but powerful machines that embody Newton's third law of motion which states that for every action there an equal and opposite reaction. Turbochargers are classified according to the driving fluid they use: steam, gas, water, and wind. Today, different types of turbines generate electricity, power ships and submarines, and propel get aircraft.

So that, turbocharger is classified as one of the important application that applied in vehicle, in order to increase the efficiency.

2.4.1 Turbocharger design

The turbocharger is made from highly heat resistance cast steel due to the exhaust gas temperature. The turbocharger has been incorporated in the cooling system to protect the shaft bearing from high temperatures.

A circulating pump ensures that the turbocharger does not overheat for up to 15 minutes after the engine has been turned off. This prevents steam bubbles forming in the cooling system.

The shaft bearing are connected to the oil system for lubrication, in order to reduce the friction, and to obtain a smooth movement.

2.4.2 Variable geometry turbocharger (VGT)

Variable geometry turbocharger (VGT) is usually designed to allow the effective aspect ratio of the turbo to be altered as condition change. This is done because optimum aspect ratio at low engine speed is very different from that at high engine speeds. If the aspect ratio is too large, the turbo will fail to create boost at low speed, if the aspect ratio is too small, the turbo will choke the engine at high speeds, leading to high exhaust manifold pressures, high pumping losses, and ultimately lower power output. By altering the geometry of the turbine housing as the engine accelerates, the turbo aspect ratio can be maintained at its optimum. Because of this, VGT have a minimal amount of lag, low boost threshold, and very efficient at higher engine speed. VGT don't require a waste gate.

Fig.2.4, shows a variable geometry turbocharger (VGT)

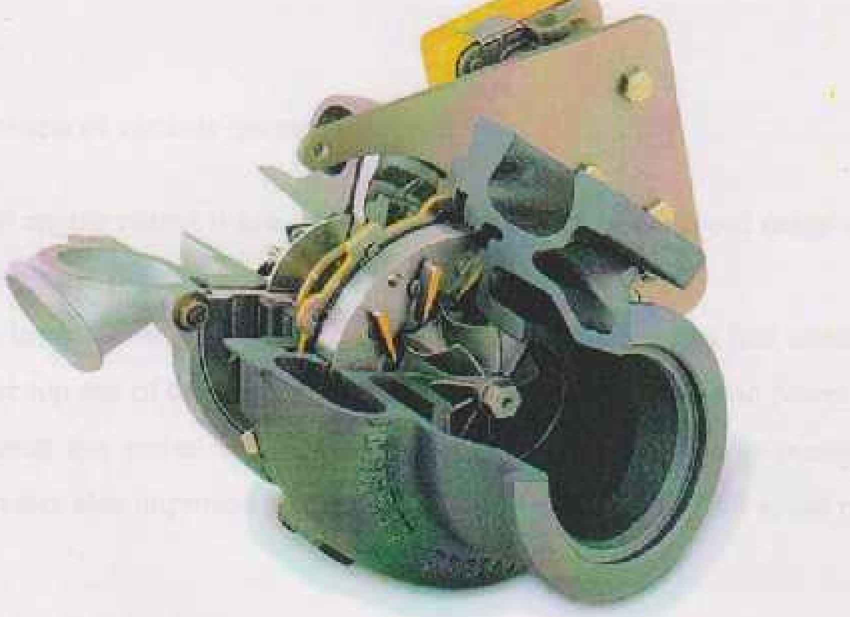


Figure (2.4) Variable geometry turbocharger (VGT)

Variable geometry turbocharger (VGT) uses a movable inlet guide vanes to regulate the speed and angle at which the exhaust gas flow strikes the turbine blades, controls the gas mass flow and its turbine power. An actuator controlled by the engine management system adjust the angle of the vanes fig2.5. Turning them to the exact position required to maintain the optimum boost pressure and air fuel ratio at every engine operating point.

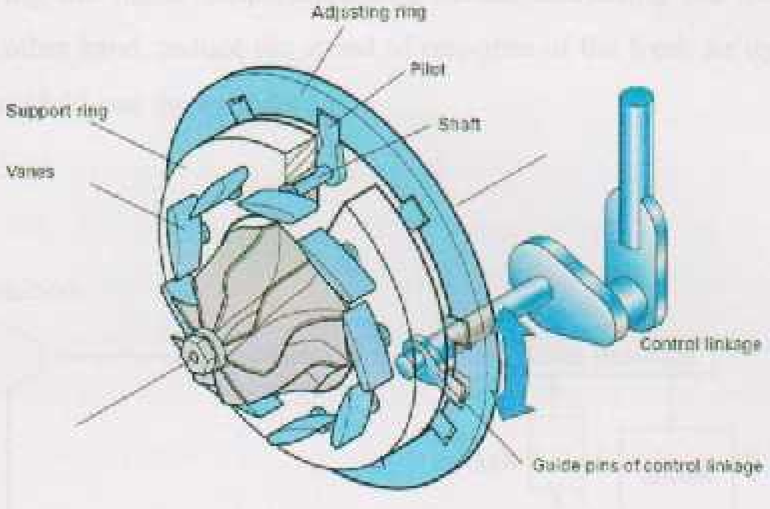


Figure (2.5) Adjusting the vanes in VGT

Main advantages of variable geometry turbocharger:

- High engine output is available at the bottom end of the speed range since the exhaust gas flow is regulated by the adjustable vanes.
- The lower exhaust gas backpressure in the turbine reduces fuel consumption at the top end of the speed range and also improves bottom end power output.
- Exhaust gas emissions decrease because an optimum charge pressure, and therefore also improved combustion, is attained across the full speed range.

2.4.3 VGT control for reduce emission

To reduce the emission of harmful oxides of nitrogen (NO_x), a partition of the exhaust gas can be diverted back to the intake manifold to dilute the air supplied by the compressor. This process is referred to as exhaust gas recirculation (EGR). It is, typically, accomplished with an EGR valve that connects the intake manifold and the exhaust manifold (see fig.2.6). In the cylinder, the combustion products act as inert gas thus lowering the flame temperature and, hence, decreasing the formation of (NO_x). On the other hand, reduce the speed of response of the fresh air dynamics in to the intake manifold and the cylinder [4].

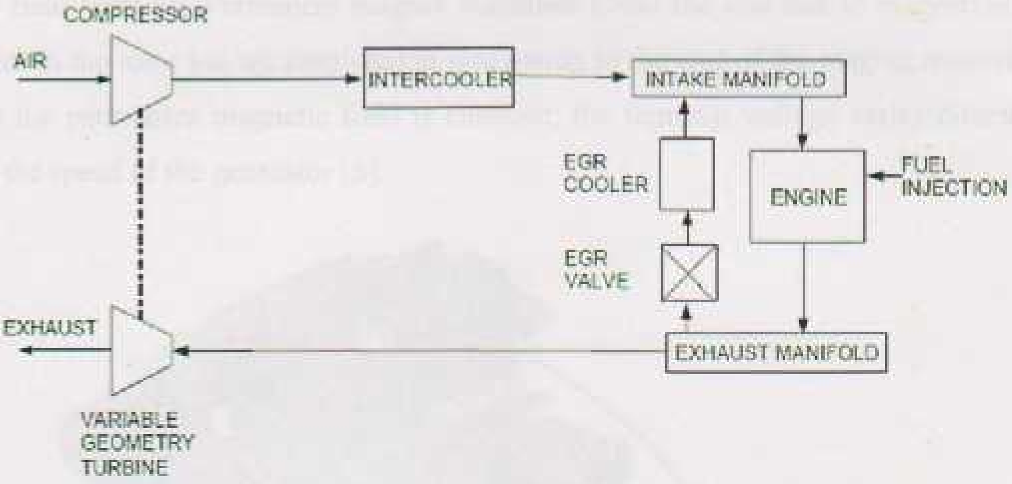


Figure (2.6) Schematic representation of the diesel engine

This behavior is caused by the two distinct mechanisms by which EGR interacts with the fresh air flow in to the cylinders. First, burned gas fraction from the EGR valve displaces fresh air from the intake manifolds and consequently decreases the speed with which fresh air mass flow in to the cylinders. Secondly, a friction of the exhaust gas that can be used by the turbine is diverted through the EGR valve to the intake manifold, reducing the turbine power and consequently the flow delivered to the intake manifold through the compressor.

2.5 Alternator principles

Alternators generate electricity by the same principle as DC generators. When magnetic field lines cut across a conductor, a current is induced in the conductor. In general, an alternator has a stationary part (stator) and a rotating part (rotor). The stator contains winding of conductors and the rotor contains a moving magnetic field. The field cut across the conductors, generating an electrical current, as the mechanical input causes the rotor to turn.

The rotor magnetic field maybe produced by induction (in a brushes generator), by permanent magnets, or by a rotor winding energized with direct current through slip rings and brushes. Automotive alternators in variably use brushes and slip rings, which allow control of the alternator generated voltage by varying the current in the rotor field winding. Permanent magnet machines avoid the loss due to magnetizing current in the rotor but are restricted in size owing to the cost of the magnet material, since the permanent magnetic field is constant; the terminal voltage varies directly with the speed of the generator [5].

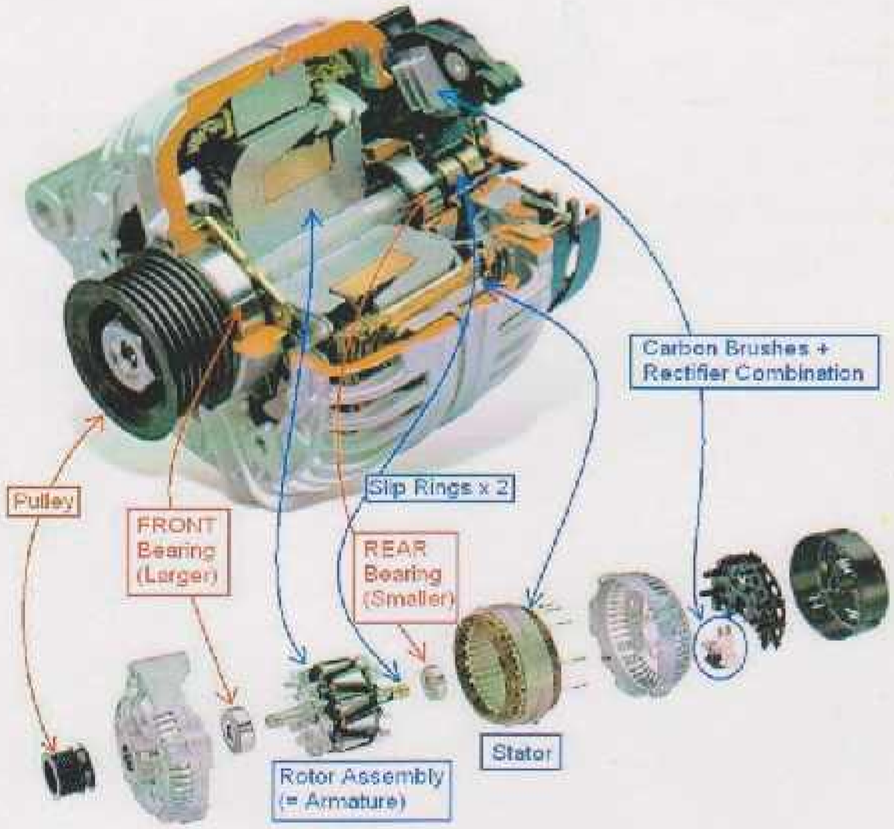


Figure (2.7) Alternator

CHAPTER THREE

SHAFT DESIGN

Contents:

3.1 Introduction

A shaft is a rotating member, usually of circular cross section, used to transmit power or motion. It provides the axis of rotation, or oscillation, of element such as gears, pulleys, flywheels, cranks sprockets…etc, and controls the geometry of their motion. An axle is a non rotating member that carries no torque and is used to support rotating wheels, pulleys, and the like. A non-rotating axle can readily be designed and analyzed as a static beam.

The shaft should be designed to transmit a maximum power that will produce from the turbine without any problems, the maximum power produced doesn't exceed 3000 watt. The purpose of the design is to find the diameter of the shaft by using the endurance limit fatigue strength, and endurance limit modifying factors.

Properties of the shaft locally depend on its history, cold work, cold forming, rolling of fillet features, heat treatment, including quenching medium, agitation, and tempering regimen.

3.2 Design formula

The purpose of this chapter is to find the diameter of the shaft using the endurance limit fatigue strength, and endurance limit modifying factors.

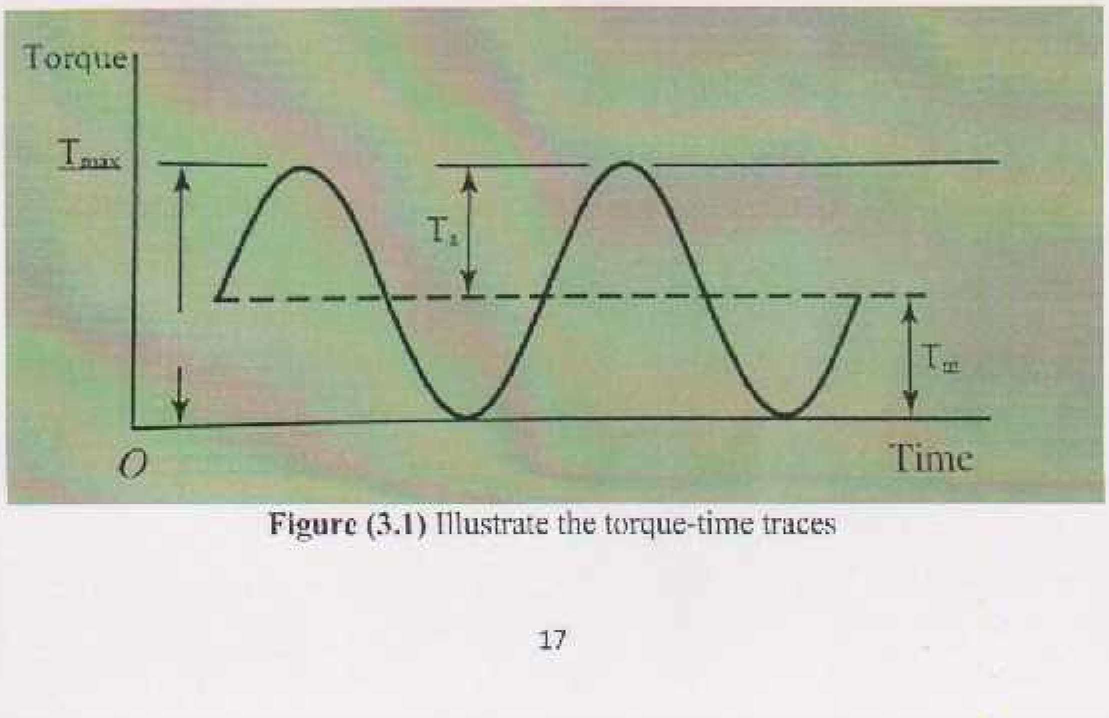


Figure (3.1) Illustrate the torque-time traces

The components of torque-time traces are:-

T_m = midrange component

T_{max} = maximum torque

T_{min} = minimum torque = zero

T_a = amplitude component

$$T_a = \frac{T_{max} - T_{min}}{2} = \frac{T_{max} - 0}{2} = 0.5\ T_{max} \qquad 3.1$$

$$T_m = \frac{T_{max} + T_{min}}{2} = \frac{T_{max} + 0}{2} = 0.5\ T_{max} \qquad 3.2$$

$$T_{max} = P_{max}/\omega \qquad 3.3$$

Where:

P_{max}: maximum output power (watt)

ω: number of turbine revolution (rev/min)

$$\tau_a = \frac{16 T_a}{\pi d^3} \qquad 3.4$$

$$\tau_m = \frac{16 T_m}{\pi d^3} \qquad 3.5$$

Where:

d : the diameter of the shaft (m)

τ_m : midrange torsion (psi)

τ_a : amplitude torsion (psi)

Fatigue factor of safety:

by mode – goodman: $$\frac{\tau_a}{S_e} + \frac{\tau_m}{S_{su}} = \frac{1}{n_f} \tag{3.6}$$

but: $$S_{ut} = 0.67S_{ut} \tag{3.7}$$

Where:

n_f : fatigue factor of safty and the value must be more than 3

S_{ut} : is the minimum tensile strengh (Kpsi)

and:

$$S_e = K_a K_b K_c K_d K_e K_f S_e' \tag{3.8}$$

Where:

S_e': rotary – beam test specimen endurance limit

And $$S_e' = \begin{cases} 0.5S_{ut} & S_{ut} \leq 200\ Kpsi\ (1400\ MPa) \\ 100\ \ Kpi & S_{ut} > 200\ Kpsi \\ 700\ MPa & S_{ut} > 1400\ MPa \end{cases} \tag{3.9}$$

S_e: endurance limit at the critical location of amachine part in the geometry and condition of use (Kpsi)

K_a: surface condition modification factor

K_b: size modification factor

K_c: load modification factor

K_d: temperature modification factor

K_e: reliability factor

K_f: miscellaneous effects modification factor

When endurance tests of parts are not available, estimation are made by applying Marin factor to the endurance limit.

Surface factor K_a

$$K_a = aS_{ut}^{b} \qquad 3.10$$

Where:

S_{ut}: is the minimum tensile strengh and a & *b are to be found in table* (3.1)

Surface Finish	Factor a S_{ut}, kpsi	Factor a S_{ut}, MPa	Exponent b
Ground	1.34	1.58	−0.085
Machined or cold-drawn	2.70	4.51	−0.265
Hot-rolled	14.4	57.7	−0.718
As-forged	39.9	272.	−0.995

Table (3.1) parameters for Marin surface modification factor

Size factor K_b

The size factor has been evaluated using 133 sets of data points. The results for bending and torsion may be expressed as:

$$K_b = \begin{cases} (d/0.3)^{-0.107} = 0.879d^{-0.107} & 0.11 \le d \le 2 \text{ in} \\ 0.91d^{-0.157} & 2 < d \le 10 \text{ in} \\ (d/7.62)^{-0.107} = 1.24d^{-0.107} & 2.79 \le d \le 51 \text{ mm} \\ 1.51d^{-0.157} & 51 < d \le 254 \text{ mm} \end{cases} \qquad 3.11$$

For axial loading there is no size effect, so $K_b = 1$

Loading factor K_c

When fatigue tests are carried out with rotating bending, axial (push-pull), and torsional loading, the endurance limits differ with S_{ut}. The average value of the load factor is:

$$K_C = \begin{cases} 1 & bending \\ 0.85 & axial \\ 0.59 & torsion \end{cases} \qquad 3.12$$

Temperature factor K_d

If the rotating beam endurance limit is known at room temperature, then use

$$K_d = \frac{S_T}{S_{RT}} \qquad 3.13$$

Temperature, °C	S_T/S_{RT}	Temperature, °F	S_T/S_{RT}
20	1.000	70	1.000
50	1.010	100	1.008
100	1.020	200	1.020
150	1.025	300	1.024
200	1.020	400	1.018
250	1.000	500	0.995
300	0.975	600	0.963
350	0.943	700	0.927
400	0.900	800	0.872
450	0.843	900	0.797
500	0.768	1000	0.698
550	0.672	1100	0.567
600	0.549		

Table (3.2) effect of operating temperature on the tensile strength of steel

Where:

S_T: tensile strengh at operating temperature

S_{RT}: tensile strengh at room temperature

Reliability factor K_e

Table (3.3) shows the reliability factors K_e corresponding to 8 percent standard deviation of the endurance limit.

$$K_e = 1 - 0.08z_a \tag{3.14}$$

Reliability, %	Transformation Variate z_a	Reliability Factor k_e
50	0	1.000
90	1.288	0.897
95	1.645	0.868
99	2.326	0.814
99.9	3.091	0.753
99.99	3.719	0.702
99.999	4.265	0.659
99.9999	4.753	0.620

Table (3.3) reliability modification factor

Miscellaneous-Effects factor K_f

Through the factor K_f is intended to account for the reduction in endurance limit due to all other effects, it is really intended as a reminder that these must be accounted for because actual values of K_f are not always available. But the assuming value is

$$K_f = 1$$

Yield factor of safety

The shear yield strength predicted by the distortion-energy theory is

$$S_{sy} = 0.577S_y \tag{3.15}$$

And

$$n_y = \frac{S_{sy}}{\tau_a + \tau_m} \tag{3.16}$$

Where:

n_y: yeild factor of safty and the value must be more than 3

S_{sy}: the shear yeild strengh (Kpsi)

When the value of fatigue factor of safety more than 3, and the value of yield factor of safety more than 3 that is mean **no fatigue and no yield (good design).**

3.3 Data analysis

$$T_{max} = P_{max}/\omega = \frac{3000\ \text{watt}}{\frac{10000\left(\frac{rev}{min}\right)*3.14*2}{60\ min}} = 2.866\ \text{N.m}$$

$$T_a = \frac{T_{max} - 0}{2} = 0.5\ T_{max} = 0.5 * 2.866 = 1.433\ \text{N.m}$$

$$T_m = \frac{T_{max} + 0}{2} = 0.5\ T_{max} = 0.5 * 2.866 = 1.433\text{N.m}$$

Assume the diameter of the shaft (10 mm) then:

$$\tau_a = \frac{16\ T_a}{\pi\ d^3} = \frac{16 * 1.433}{3.14 * (10 * 10^{-3})^3} = 7.3 * 10^6\ \text{N/m}^2 = 1.059\ \text{Kpsi}$$

$$\tau_m = \frac{16\ T_m}{\pi\ d^3} = \frac{16 * 1.433}{3.14 * (10 * 10^{-3})^3} = 7.3 * 10^6\ \text{N/m}^2 = 1.059\ \text{Kpsi}$$

The type of steel is 1050 HR, and from appendix, the value of S_{ut} is 90 Kpsi

Then : $S_{su} = 0.67\ S_{ut} = 0.67 * 90 = 60.3\ \text{Kpsi}$

From equation (3.9), $S_e' = 0.5 * S_{ut} = 0.5 * 90 = 45\ \text{Kpsi}$

and : $S_e = K_a K_b K_c K_d K_e K_f S_e'$

Where:

$K_a = aS_{ut}^b = 14.4 * (90)^{-0.718} = 0.57$, a&b *from table* (3.1)

$K_b = 1.24d^{-0.107} = 1.24 * (10)^{-0.107} = 0.97$

$K_c = 0.59$, from equatiom (3.12)

$K_d = 0.768$, from table (3.2)

$K_e = 1$, when reliability (50%), from table (3.3)

$K_f = 1$

The endurance limit:-

$S_e = 0.57 * 0.97 * 0.59 * 0.768 * 1 * 1 * 45 = 11.27$ Kpsi

Fatigue factor of safety:-

$$\frac{\tau_a}{S_e} + \frac{\tau_m}{\tau_{su}} = \frac{1}{n_f} \rightarrow \frac{1.059}{11.27} + \frac{1.059}{60.3} = \frac{1}{n_f} \rightarrow n_f = 8.9 > 3$$

Yield factor of safety:-

$S_{sy} = 0.577 S_y = 0.577 * 49.5 = 28.56$ Kpsi , S_y from appendix

And : $$n_y = \frac{S_{sy}}{\tau_a + \tau_m} = \frac{28.56}{1.059 + 1.059} = 13.4 > 3$$

Since the fatigue factor of safety and yield factor of safety are more than 3, this means that the design is good (because no fatigue and no yield).

CHAPTER FOUR

POWER TRANSMISSION

Contents:

4.1 Introduction

Belts, ropes, chains, and other similar elastic or flexible elements are used in conveying systems and in the transmission of power over comparatively long distance. In this project a V-belt will be used to transmit power from the turbocharger to the alternator.

4.2 Belts

The four principal types of belts are shown, with some of their characteristics in table (4-1). Crowned pulleys are used for flat belts, and grooved pulleys, or sheaves, for round and V belts. Timing belts require toothed wheels, or sprockets. In all cases, the pulley axes must be separated by certain minimum distance, depending upon the belt type and size, to operate properly. Other characteristic of belts are:

- They may be used for long center distances.
- Except for timing belts, there are some slip and creep, and so the angular velocity ratio between the driving and driven shaft neither constant nor exactly equal to the ratio of the pulley diameters.
- In some cases and idler or tension pulley can be used to avoid adjustments in center distance that are ordinary necessitated by age or the installation of new belts.

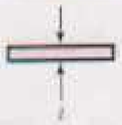
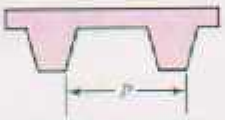
Belt Type	Figure	Joint	Size Range	Center Distance
Flat		Yes	$t = \begin{cases} 0.03 \text{ to } 0.20 \text{ in} \\ 0.75 \text{ to } 5 \text{ mm} \end{cases}$	No upper limit
Round		Yes	$d = \frac{1}{8}$ to $\frac{3}{4}$ in	No upper limit
V		None	$b = \begin{cases} 0.31 \text{ to } 0.91 \text{ in} \\ 8 \text{ to } 19 \text{ mm} \end{cases}$	Limited
Timing		None	$p = 2$ mm and up	Limited

Table (4.1) Characteristics of some common belt types

The belt which used to transmit the power between turbine and alternator is V belt, because of the following advantages.

4.3 V Belts

4.3.1 Advantage and disadvantages of (V belt)

Advantages:

1. The force of friction between the surface of the belt and pulley is high due to wedge action. This increase the pulling capacity of the belt and consequently results in increase the power transmission capacity.
2. V-belts have short center distance, which results in compact construction.
3. They permit high speed reduction even up to 7 to 1.
4. Smooth and quite operation even at high speed.
5. Positive drive, the slip is negligible due to wedge action.

Disadvantages:

1. The efficiency of the V belt is lower than that of flat belt.
2. The construction of V grooved pulley is complicated and costly.

4.3.2 V belt design

Design a V-belt drive for the following data:

The faster shaft runs at n=2000 rpm, the slower shaft runs at 1000 rpm, and the required power transmission is 2 hp.

$$SR = \frac{\text{rpm of faster shaft}}{\text{rpm of slower shaft}} = \frac{D}{d} = \frac{2000}{1000} = 2:1$$

$$H_d = k_s n_d H_{nom} = 1.2 * 1 * 2 = 2.4 \text{ hp}$$

Where:

SR: Speed ratio

H_d: Design power that is expected to be transmitted from the power source to the load through the belts.

K_s: The service factor, from appendix

n_d: The design factor

H_{nom}: Is the nominal transmitted power between the driver and the driven machine

Compute the driving sheaving-size limits that would produce a belt speed.

$$5\,m/s \le \frac{\pi d n}{60} \le 25\,m/s \rightarrow \frac{300}{\pi n} \le d \le \frac{1500}{\pi n} \rightarrow \frac{300}{\pi(2000)} \le d \le \frac{1500}{\pi(2000)}$$

$$0.048m \le d \le 0.238m \rightarrow 48mm \le d \le 238mm$$

For AX section belt, we may select: d = 54mm, D = 108mm

According to the selected standard driving sheave diameter d, D, and the driving rpm (n), we can compute the related power (H_r) of the selected belt section.

$$H_r = H_{basic} + H_{add} = 1.28 + 0.51 = 1.79 \text{ KW}$$

Where:

H_{basic}: basic power (KW)rating of section (AX), from appendix

H_{add}: additional power (KW)per belt for speed ratio, from appendix

To specify a trial center distance (C):

$D < C < 3(D + D) \rightarrow 108 < C < 3(54 + 108) \rightarrow 108mm < C < 486mm$

Therefore, let use consider, C = 400mm

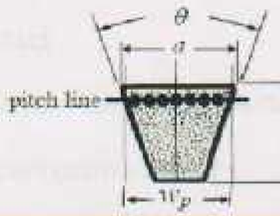
To compute the required belt pitch-length(L_p):

$$L_p = 2C + \frac{\pi}{2}(D + d) + \frac{(D - d)^2}{4C} = 2 * 400 + \frac{\pi}{2}(108 + 54) + \frac{(108 - 54)^2}{4 * 400}$$

$$= 1051.8mm$$

Then, the inside length of the belt according to table (4.2) is:

$L_i = L_P - 36 = 1051.8 - 36 = 1015.8mm$



Section	Dimension		Angle	Pitch width	Belt Length Factor		
	a mm	b mm	θ Deg	w_p mm	Lp to La mm	Li to Lp mm	Li to La mm
A & AX	13	8	40	11.0	14	36	50
B & BX	17	11	40	14.0	26	43	69
C & CX	22	14	40	19.0	32	56	88
D	32	19	40	27.0	40	79	119
E	38	23	40	32.0	53	92	145
3V & 3VX	9.7	8	40	8.9	13	37	50
5V & 5VX	16	14	40	15.2	25	60	85
8V	25	23	40	25.4	53	92	145

Table (4.2) standard V-belt section

From appendix (table 17-2(i)), select AX-40 with $L_i = 1016mm$

This yeild $L_P = 1016 + 36 = 1052mm$

Now, the exact center distance becomes

$$C = 0.25\left\{\left(L_P - \frac{\pi}{2}(D + d)\right) + \sqrt{\left(L_P - \frac{\pi}{2}(D + d)\right)^2 - 2(D - d)^2}\right\}$$

$$= 0.25\left\{\left(1052 - \frac{\pi}{2}(108 + 54)\right) + \sqrt{\left(1052 - \frac{\pi}{2}(108 + 54)\right)^2 - 2(108 - 54)^2}\right\}$$

$$= 398mm$$

$(D - d)/C = (108 - 54)/398 = 0.135$

From appendix (table 17-8): $\theta = 172.25 = 3$ rad, $K_1 = 0.983$

From appendix (table 17-9): $K_2 = 0.9$

Where:

θ: The angle of contact

K_1: The angle of contact correction factor

K_2: The belt legth correction factor

The allowable power per belt:

$H_a = K_1K_2H_r = 0.983 * 0.9 * 1.79 = 1.584$ KW $= 2.13$ hp

Then, the number of belt required to carry the design power:

$N_b = \frac{H_d}{H_a} = \frac{2.4}{2.13} = 1.12$ Therefore, select $N_b = 2$belts

$$T_1 = \frac{e^{\mu\theta/\sin\alpha}(H_a/v)}{e^{\mu\theta/\sin\alpha} - 1}$$

Assume $\mu = 0.3, 2\alpha = 40^{o}, v = \pi dn/60 = \pi * 0.3 * \frac{2000}{60} = 31.4\ m/s$

$$T_1 = \frac{e^{0.3(3)/\sin 20^{0}}(1584/31.4)}{e^{0.3(3)/\sin 20^{0}} - 1} = 54.37\ N$$

$$T_2 = \frac{(H_a/v)}{e^{\mu\theta/\sin\alpha} - 1} = \frac{(1584/31.4)}{e^{0.3(3)/\sin 20^{0}} - 1} = 3.91\ N$$

$$T_i = \frac{T_1 + T_2}{2} = 29.14\ N$$

Where:

T_1, T_2: The working belt tension loads

The minimum allowance, x and y for adjusting drive canter distance for the belts, from appendix (Table 17-3(a))

$$x = 20mm, y = 15mm$$

$$F_a = \sqrt{T_1^{\ 2} + T_2^{\ 2} + 2T_1T_2\cos(\pi - \theta)}$$

$$= \sqrt{54.37^2 + 3.91^2 + 2(54.37)3.91\cos(\pi - 3)} = 58.3\ N$$

Where:

F_a: The driving shaft load

4.3.3 Summary of the design

Input: 2 hp @ 2000 rpm

Service factor: 1.2

Design power: 2.4 hp

Belt: AX-40, 2belt

Sheaves: Driver, 54mm pitch diameter

Driven, 108mm pitch diameter

Center distance: 398 mm

Belt tension: $T_1 = 54.37$ N, $T_2 = 3.91$ N, $T_i = 29.14$ N

Minimum allowance: x = 20 mm, y =15 mm

Driving shaft load: $F_a = 58.3$ N

CHAPTER FIVE

POWER OF THE TURBOCHARGER

Contents:

5.1 Power input to the turbine

The power input to the turbine increases as the pressure and volume flow rate of the exhaust gases increase. The following equation expresses the value of power input to the turbine.

$$\text{Power}_{in} = \frac{Q * P_{in}}{17.1}\left[\left(\frac{P_{in}}{P_{out}}\right)^{\alpha} - 1\right] \qquad 5.1$$

$$\alpha = \frac{K-1}{K} \qquad 5.2$$

Where:

Power_{in}: Power input (KW)

Q : Volume flow rate (m^3/Sec)

P_{in} : Exhaust gas pressure before the turbine (KPa)

P_{out} : Exhaust gas pressure after the turbine (KPa)

K : Ratio of specific heats (C_p/C_V)

$$Q = \frac{rpm}{60}\frac{V_{cc}}{2(for\ two\ revolution)} \qquad 5.3$$

Where:

V_{cc} : Combustion chamber volume in cubic centimeters (cm^3)

Note:

1- The volume of the engine is 1900 cc = $1900*10^{-6}$ m^3

2- K for air = 1.4

5.2 Power output from turbine

One of the most important ways to find the turbine output power is:

5.2.1 Dynamometer

Dynamometer is used to measure the torque and angular velocity of the turbine then the power is expected as follows:-

$$P_{out} = T * \omega \qquad 5.4$$

$$\omega = \frac{N * 2\pi}{60} \qquad 5.5$$

Where:

P_{out} : turbine output power

T : Torque (N.m)

ω : Angular velocity

Fig.5.1, and fig.5.2, Shows some of the turbine speed readings at a certain engine speeds.

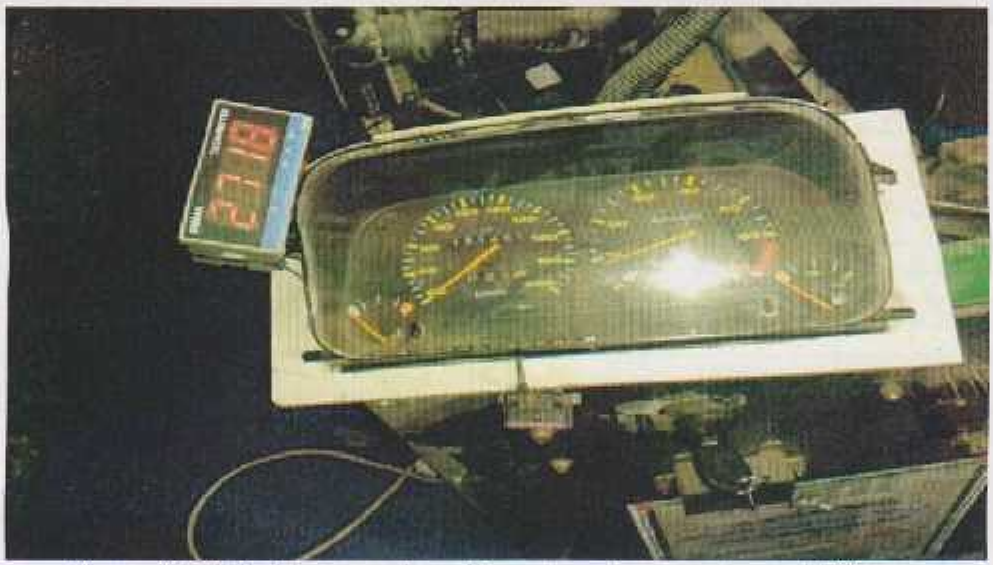


Figure (5.1) Turbine speed readings at engine speed equal 800 rpm

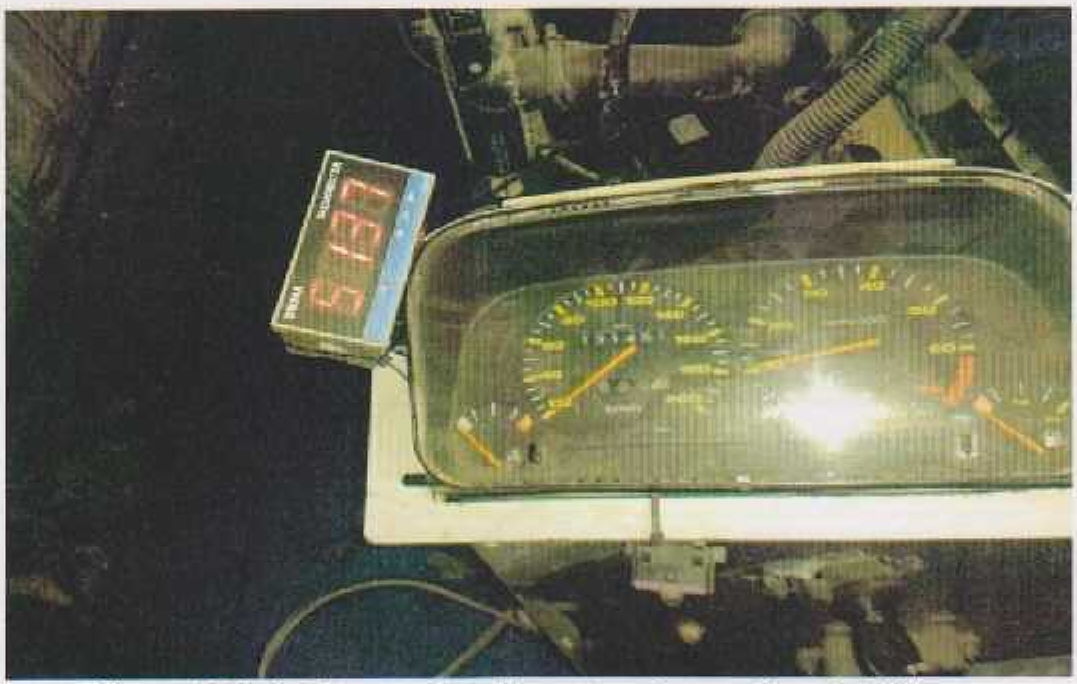


Figure (5.2) Turbine speed readings at engine speed equal 1000 rpm

Table (5.1), Shows the turbine output power calculations at a certain engine speeds, and fig.5.3, Shows the relation between engine speed and turbine output power.

Engine speed (rpm)	Radius (r) (m)	Mass (Kg)	Turbine speed (rpm)	Output power (KW)
800	0.025	0.08	2718	0.0057
1000	0.025	0.08	5137	0.0108
1500	0.025	0.25	2812	0.0184
2000	0.025	0.55	2640	0.0524
3000	0.025	2.5	2200	0.1440
4000	0.025	3.5	2350	0.2150
5000	0.025	4.2	3476	0.382

Table (5.1) Turbine output power at a certain engine speeds

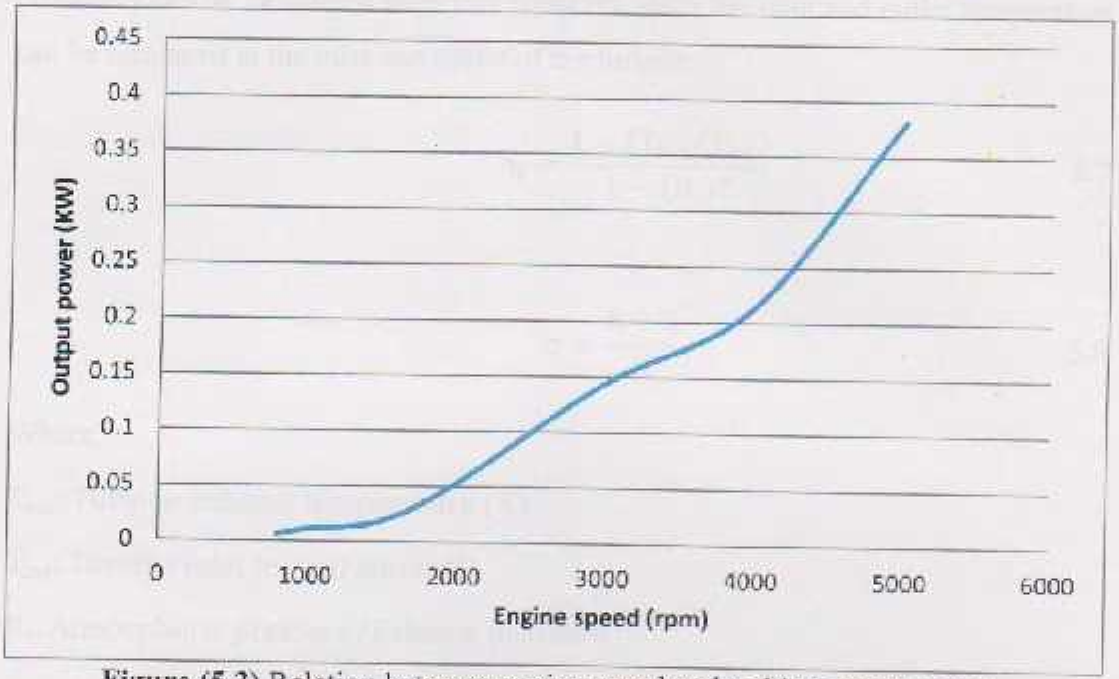


Figure (5.3) Relation between engine speed and turbine output power

Notes:

- The power output that's calculated here is the max. Power without load on the engine.
- The power output with load on the engine wasn't calculated, because there is no way to do that in the mechanical work shop.

5.3 Turbine efficiency

The second way to determine the turbine input power is using the turbine efficiency equations.

$$\eta_t = \frac{\text{power output}}{\text{power input}} \qquad 5.6$$

The output power can be calculated from equation 5.4 and 5.5, and the turbine efficiency can be calculated from this equation, since the inlet and outlet temperature can be measured at the inlet and outlet of the turbine.

$$\eta_t = \frac{1 - (T_{ext}/T_{int})}{1 - (R_t)^{\sigma}} \qquad 5.7$$

$$\sigma = \frac{K - 1}{K} \qquad 5.8$$

Where:

T_{ext}: Turbine exhaust temperature (K)

T_{int}: Turbine inlet temperature (K)

R_t: Atmospheric pressure/Exhaust pressure

K: Ratio of specific heat (C_p/C_v)

C_p: Specific heat at constant pressure (KJ/Kg. K)

C_v: Specific heat at constant volume (KJ/Kg. K)

In this case, the efficiency of turbine is selected, which is selected to be about 80%, because the efficiency of turbine is between (70%-80%).

The power input is:

$$\eta_t = \frac{\text{power output}}{\text{power input}} \rightarrow 0.8 = \frac{0.0057}{\text{power input}} \qquad 5.9$$

Power input = 0.00712 KW

Table (5.2) shows the turbine input power at a certain engine speeds, and fig.5.4, shows the relation between engine speed and turbine input power.

Engine speed (rpm)	input power (KW)
800	0.0071
1000	0.0135
1500	0.023
2000	0.0655
3000	0.18
4000	0.269
5000	0.478

Table (5.2) Turbine input power at a certain engine speeds

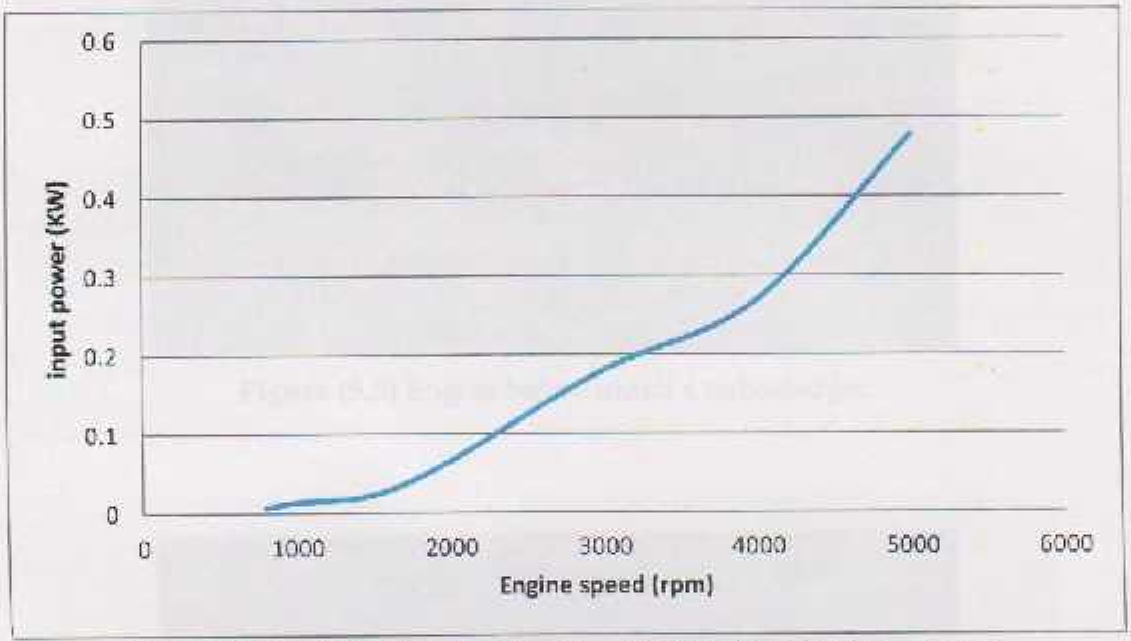


Figure (5.4) Relation between engine speed and turbine input power

As a result of this, we concluded that the power produced from the turbine is not enough to run the alternator and produce needed electric current to supply all vehicle electric loads.

5.4 Project main components

5.4.1 Internal combustion engine

The practical part of the project will be done using (VW, 1900cc, 99model) engine available in the mechanical engineering department workshop, four strokes, four cylinder. The engine was a naturally separated engine, and we install a turbocharger in this engine as shown in fig.5.5, and fig.5.6.

Figure (5.5) Engine before install a turbocharger

Figure (5.6) Engine after install a turbocharger

5.4.2 Turbocharger

A variable geometry turbocharger will be used the velocity of the exhaust gasses to drive the alternator, as shown in fig.5.7.

Figure (5.7) Turbocharger

5.4.3 Alternator

The alternator will be used to convert the mechanical energy that produced from the turbine in to electrical energy. Fig.5.8 shows an alternator connected with the turbocharger by a V-belt.

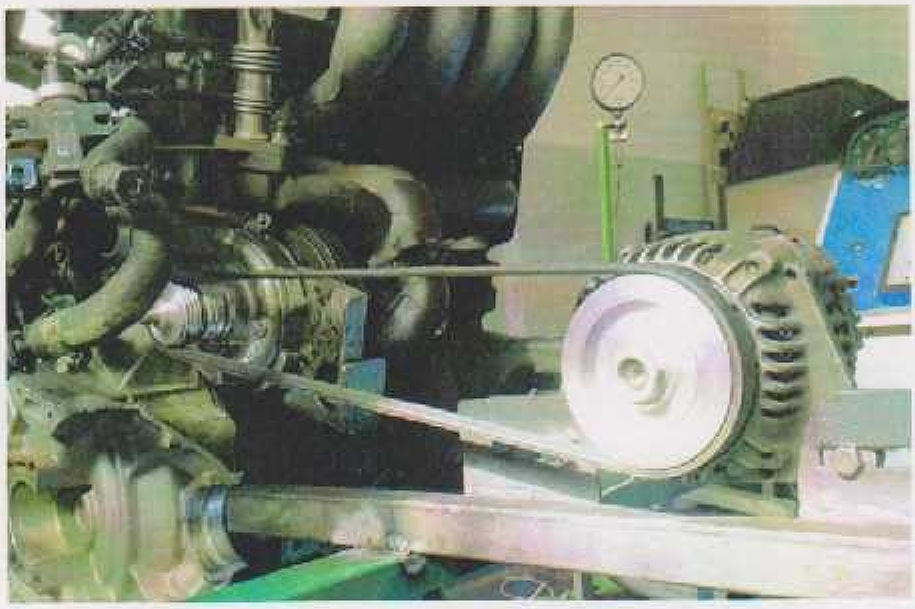


Figure (5.8) Alternator connected with the turbocharger

5.4.4 RPM sensor

RPM sensor is used to measure the number of revolutions of the turbocharger at specified value of weight, for the purpose of measuring the turbine output power. As seen in fig.5.9.

Figure (5.9) RPM sensor

CHAPTER SIX

CONCLUSIONS RECOMMENDATIONS AND PROBLEMS

Contents:

6.1 Conclusions

The maximum power produced from the turbine is about 382 W, and this is not enough to run the alternator and produce needed electric current to supply all vehicle electric loads.

6.2 Recommendations

1. Utilization the power produced from exhaust gases to operate some vehicle devices that require low level of power.
2. The power produced can be used as assistant to the power source in the vehicle.

6.3 Problems

The problem that interfaces this project is installation of the dynamometer on the engine to find the effect of turbine on the engine power, because there were difficulties to the install the dynamometer on the engine which required calibration.

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Table A–20

Deterministic ASTM Minimum Tensile and Yield Strengths for Some Hot-Rolled (HR) and Cold-Drawn (CD) Steels [The strengths listed are estimated ASTM minimum values in the size range 18 to 32 mm ($\frac{3}{4}$ to $1\frac{1}{4}$ in). These strengths are suitable for use with the design factor defined in Sec. 1–10, provided the materials conform to ASTM A6 or A568 requirements or are required in the purchase specifications. Remember that a numbering system is not a specification.] *Source:* 1986 SAE Handbook, p. 2.15.

1 UNS No.	2 SAE and/or AISI No.	3 Proces-sing	4 Tensile Strength, MPa (kpsi)	5 Yield Strength, MPa (kpsi)	6 Elongation in 2 in, %	7 Reduction in Area, %	8 Brinell Hardness
G10060	1006	HR	300 (43)	170 (24)	30	55	86
		CD	330 (48)	280 (41)	20	45	95
G10100	1010	HR	320 (47)	180 (26)	28	50	95
		CD	370 (53)	300 (44)	20	40	105
G10150	1015	HR	340 (50)	190 (27.5)	28	50	101
		CD	390 (56)	320 (47)	18	40	111
G10180	1018	HR	400 (58)	220 (32)	25	50	116
		CD	440 (64)	370 (54)	15	40	126
G10200	1020	HR	380 (55)	210 (30)	25	50	111
		CD	470 (68)	390 (57)	15	40	131
G10300	1030	HR	470 (68)	260 (37.5)	20	42	137
		CD	520 (76)	440 (64)	12	35	149
G10350	1035	HR	500 (72)	270 (39.5)	18	40	143
		CD	550 (80)	460 (67)	12	35	163
G10400	1040	HR	520 (76)	290 (42)	18	40	149
		CD	590 (85)	490 (71)	12	35	170
G10450	1045	HR	570 (82)	310 (45)	16	40	163
		CD	630 (91)	530 (77)	12	35	179
G10500	1050	HR	620 (90)	340 (49.5)	15	35	179
		CD	690 (100)	580 (84)	10	30	197
G10600	1060	HR	680 (98)	370 (54)	12	30	201
G10800	1080	HR	770 (112)	420 (61.5)	10	25	229
G10950	1095	HR	830 (120)	460 (66)	10	25	248

Table 17-1(b): Recommended minimum pulley diameter

Classic belt section	Minimum pulley pitch diam. [mm]	Narrow belt section	Minimum pulley pitch diam. [mm]
A	71	3V	63
AX	63	3VX	50
B	106	5V	125
BX	90	5VX	100
C	170	8V	315
CX	140	f1.tif	
D	315		
E	400		

Table 17-2(*i*): Standard length of classical section, AX

20

Belt Reference	Inside Length (mm)	Belt Reference	Inside Length (mm)	Belt Reference	Inside Length (mm)	Belt Reference	Inside Length (mm)
AX 22	559	AX 70	1778	AX 120	3048	AX 170	4318
AX 25	635	AX 75	1905	AX 125	3175	AX 175	4445
AX 30	762	AX 80	2032	AX 130	3302	AX 180	4572
AX 35	889	AX 85	2159	AX 135	3429	AX 185	4699
AX 40	1016	AX 90	2286	AX 140	3556	AX 190	4826
AX 45	1143	AX 95	2413	AX 145	3683	AX 195	4953
AX 50	1270	AX 100	2540	AX 150	3810	AX 200	5080
AX 55	1397	AX 105	2667	AX 155	3937		
AX 60	1524	AX 110	2794	AX 160	4064		
AX 65	1651	AX 115	2921	AX 165	4191		

Table 17-3(*a*) Minimum allowance; x and y for adjusting drive center distance for classical belts

Belt length (mm)	Minimum allowance x (mm) - for tensioning	Minimum allowance y (mm) - for fitting				
		A/AX	B/BX	C/CX	D	E
≤ 200	5	—	—	—	—	—
> 200 ≤ 250	5	—	—	—	—	—
> 250 ≤ 315	5	—	—	—	—	—
> 315 ≤ 670	10	10	10	—	—	—
> 670 ≤ 1 000	15	15	15	—	—	—
> 1 000 ≤ 1 250	20	15	15	20	—	—
> 1 250 ≤ 1 800	25	20	20	25	—	—
> 1 800 ≤ 2 240	25	20	20	25	35	—
> 2 240 ≤ 3 000	35	20	20	30	35	40
> 3 000 ≤ 4 000	45	20	20	30	35	40
> 4 000 ≤ 5 000	55	20	20	30	35	40
> 5 000 ≤ 6 300	70	20	25	35	40	45
> 6 300 ≤ 8 000	85	20	25	40	45	50
> 8 000 ≤ 10 000	110	25	25	45	45	50
> 10 000 ≤ 12 500	135	—	30	45	50	55
> 12 500 ≤ 15 000	150	—	40	55	60	65
> 15 000 ≤ 18 000	190	—	40	55	60	65

Table 17-6: V-belt service factors, K_s

		Types of driver					
		Soft starts			Heavy starts		
		AC motors: Normal torque[a] DC motors: Shunt-wound Engines: Multiple-cylinder			AC motors: High torque[b] DC motors: Series-wound, compound-wound Engines: 4-cylinder or less		
Load Type	Driven machine type	<6 h per day	6–15 h per day	>15 h per day	<6 h per day	6–15 h per day	>15 h per day
Smooth	Agitators, blowers, fans, centrifugal pumps, light conveyors	1.0	1.1	1.2	1.1	1.2	1.3
Light Shock	Generators, machine tools, mixers, gravel conveyors	1.1	1.2	1.3	1.2	1.3	1.4
Medium Shock	Bucket elevators, textile machines, hammer mills, heavy conveyors	1.2	1.3	1.4	1.4	1.5	1.6
High Shock	Crushers, ball mills, hoists, rubber extruders	1.3	1.4	1.5	1.5	1.6	1.8
Heavy Shock	Any machine that can choke	2.0	2.0	2.0	2.0	2.0	2.0

Table 17-7(l): Basic power (kW) rating of section "AX" /part1

N RPM	Pitch diameter of the smaller pulley (mm)																Additional Power (kW) per belt for speed ratio			
	63	71	80	90	95	100	106	112	118	125	132	140	150	160	180	200	1.01 to 1.05	1.06 to 1.26	1.27 to 1.57	For > 1.57
700	0.70	0.90	1.11	1.35	1.47	1.59	1.72	1.86	1.98	2.14	2.30	2.47	2.68	2.89	3.31	3.78	0.02	0.08	0.12	0.18
950	0.88	1.11	1.39	1.69	1.84	1.98	2.16	2.33	2.51	2.71	2.90	3.12	3.39	3.65	4.18	4.81	0.03	0.11	0.18	0.24
1450	1.09	1.45	1.85	2.27	2.48	2.69	2.93	3.17	3.40	3.66	3.95	4.25	4.60	4.98	5.68	6.84	0.04	0.17	0.24	0.37
2850	1.46	2.08	2.70	3.29	3.73	4.05	4.43	4.81	5.17	5.59	5.96	6.39	6.90	7.38	8.23	10.92	0.09	0.33	0.47	0.73
100	0.16	0.20	0.24	0.28	0.30	0.33	0.36	0.38	0.41	0.43	0.46	0.49	0.54	0.58	0.66	0.75	0.00	0.01	0.02	0.03
200	0.28	0.35	0.42	0.50	0.55	0.59	0.63	0.68	0.72	0.78	0.83	0.89	0.97	1.04	1.19	1.35	0.01	0.02	0.03	0.05
300	0.39	0.48	0.59	0.70	0.76	0.82	0.88	0.95	1.02	1.09	1.17	1.25	1.35	1.46	1.67	1.92	0.01	0.03	0.05	0.08
400	0.47	0.60	0.74	0.88	0.96	1.03	1.11	1.20	1.28	1.38	1.47	1.59	1.72	1.85	2.11	2.96	0.01	0.05	0.07	0.10
500	0.56	0.70	0.87	1.05	1.13	1.22	1.32	1.42	1.52	1.63	1.75	1.89	2.06	2.22	2.53	2.98	0.02	0.06	0.08	0.13
600	0.63	0.81	1.00	1.21	1.30	1.41	1.52	1.65	1.75	1.90	2.04	2.18	2.37	2.56	2.93	3.69	0.02	0.07	0.10	0.15
700	0.70	0.90	1.11	1.35	1.47	1.59	1.72	1.86	1.96	2.14	2.30	2.47	2.68	2.89	3.31	3.78	0.02	0.08	0.12	0.19
800	0.77	0.99	1.23	1.49	1.69	1.75	1.90	2.05	2.21	2.31	2.54	2.74	2.97	3.21	3.66	4.30	0.02	0.09	0.13	0.21
900	0.83	1.07	1.33	1.63	1.76	1.97	2.08	2.25	2.40	2.59	2.78	2.99	3.26	3.51	4.01	4.70	0.03	0.10	0.15	0.23
1000	0.88	1.14	1.44	1.75	1.91	2.06	2.25	2.43	2.60	2.81	3.01	3.24	3.53	3.80	4.35	5.10	0.03	0.11	0.17	0.26
1100	0.93	1.22	1.53	1.88	2.05	2.21	2.40	2.60	2.79	3.01	3.23	3.48	3.78	4.08	4.66	5.50	0.03	0.13	0.18	0.29
1200	0.99	1.29	1.62	2.00	2.17	2.35	2.56	2.77	2.92	3.21	3.44	3.71	4.03	4.35	4.97	5.90	0.04	0.14	0.20	0.31
1300	1.03	1.35	1.72	2.11	2.30	2.49	2.71	2.93	3.15	3.40	3.65	3.93	4.27	4.61	5.26	6.25	0.04	0.15	0.22	0.33
1400	1.07	1.42	1.81	2.22	2.42	2.63	2.86	3.09	3.32	3.59	3.85	4.15	4.50	4.86	5.54	6.60	0.04	0.16	0.23	0.36
1500	1.11	1.48	1.89	2.32	2.53	2.75	2.99	3.24	3.49	3.77	4.04	4.35	4.72	5.09	5.81	6.95	0.05	0.17	0.25	0.38
1600	1.16	1.54	1.96	2.43	2.65	2.87	3.13	3.39	3.64	3.94	4.22	4.55	4.95	5.32	6.07	7.30	0.05	0.18	0.26	0.41
1700	1.19	1.60	2.04	2.52	2.75	2.98	3.27	3.53	3.80	4.11	4.40	4.74	5.15	5.54	6.31	7.64	0.05	0.20	0.28	0.44
1800	1.22	1.65	2.11	2.61	2.86	3.10	3.38	3.66	3.95	4.26	4.57	4.92	5.34	5.75	6.54	7.98	0.05	0.21	0.30	0.46
1900	1.25	1.70	2.18	2.70	2.96	3.21	3.51	3.80	4.08	4.41	4.74	5.09	5.53	5.95	6.76	8.31	0.06	0.22	0.31	0.49
2000	1.28	1.74	2.25	2.79	3.06	3.32	3.62	3.93	4.22	4.56	4.89	5.26	5.71	6.14	6.97	8.65	0.06	0.23	0.33	0.51
2100	1.31	1.79	2.31	2.88	3.15	3.41	3.74	4.04	4.35	4.70	5.04	5.49	5.88	6.33	7.17	8.98	0.06	0.24	0.35	0.54
2200	1.33	1.83	2.37	2.96	3.23	3.52	3.84	4.16	4.47	4.84	5.19	5.58	6.05	6.50	7.35	9.05	0.07	0.25	0.36	0.56
2300	1.35	1.87	2.43	3.02	3.32	3.61	3.95	4.27	4.60	4.97	5.32	5.72	6.21	6.67	7.52	9.55	0.07	0.26	0.38	0.59
2400	1.38	1.91	2.49	3.10	3.40	3.70	4.04	4.38	4.71	5.09	5.46	5.86	6.35	6.81	7.68	9.85	0.07	0.26	0.40	0.62
2500	1.40	1.94	2.54	3.17	3.48	3.78	4.14	4.48	4.82	5.21	5.58	6.00	6.49	6.96	7.82	10.08	0.08	0.29	0.41	0.64
2600	1.42	1.97	2.58	3.23	3.56	3.86	4.23	4.58	4.92	5.32	5.70	6.12	6.62	7.09	7.96	10.30	0.08	0.30	0.43	0.67
2700	1.44	2.01	2.64	3.31	3.62	3.95	4.32	4.67	5.03	5.43	5.81	6.24	6.74	7.21	8.07	10.53	0.08	0.31	0.45	0.69

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Table 17-8:
Angle of contact correction factor, K_1

(D-d)/C	Arc of Contact on Small Sheave θ [deg.]	Correction Factor K_1
0.00	180	1.00
0.10	174	0.99
0.20	169	0.97
0.30	163	0.96
0.40	157	0.94
0.50	151	0.93
0.60	145	0.91
0.70	139	0.89
0.80	133	0.87
0.90	127	0.85
1.00	120	0.82
1.10	113	0.80
1.20	106	0.77
1.30	99	0.73
1.40	91	0.70
1.50	83	0.65

Table 17-9: Belt length correction factor, K_2

Length Factor K_2	Belt Pitch Length, Lp [mm]							
	A,AX	B,BX	C,CX	D	E	3V,3VX	5V,5VX	8V
0.80	630							
0.81		930						
0.82	700		1550	2740				
0.83		1000				630		
0.84	790		1760					
0.85		1100				710		
0.86	890			3130				
0.87		1210	1950	3330		800	1400	2540
0.88	990							
0.89						900	1600	3000
0.90	1100	1370	2190	3730				
0.91			2340				1800	3350
0.92		1560	2490	4080		1000		
0.93	1250						2090	
0.94			2720	4620	5334	1120		4060
0.95		1760	2800				2240	
0.96	1430		3080		6045	1250	2500	
0.97		1950		5400				5080
0.98	1550		3310			1400	2800	
0.99	1640	2180	3520		6807			6000
1.00	1750	2300		6100		1600	3150	
1.02	1940	2500	4060		7569	1800	3550	7100
1.03				6840	8331			8000
1.04	2050	2700				2000	4090	
1.05	2200	2850	4600	7620	9093			9000
1.06	2300						4500	
1.07				8410	9855	2240		10160
1.08	2480	3200	5380				5000	
1.09	2570			9140	10617	2500	5600	11430
1.10	2700	3600						12700
1.11			6100			2800	6300	
1.12	2910			10700	12141			
1.13	3080	4060				3150	7100	
1.14	3290		6860		13665			
1.15		4430				3550	7880	
1.16	3540	4820	7600	12200	15189			
1.18		5000		13700				
1.19		5370			16713			
1.20		6070		15200				
1.21			9100					
1.24			10700					