بسم الله الرحمن الرحيم

Palestine Polytechnic University

College of Engineering & Technology

Mechanical Engineering Department

Graduation Project

Hybrid Air Conditioning in Vehicle

Project Team

Saed Hawareen **Rami Asafrah**

Project Supervisor

Dr. Zuhdi Salhab

Hebron-Palestine

2009

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According to the project supervisor and according to the agreement of the Testing committee members, this project is submitted to the Department of Mechanical Engineering at college of engineering and technology in partial fulfillment of the requirements of (B.A) the bachelor's degree in Automotive Engineering.

Supervisor Signature

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Discussion Committee Signature

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Department Head Signature

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Hebron-Palestine

2009

Dedication

To my friends…	*Who help me...*
To my parents…	*Who raised me...*
To my beloved…	*Who supported me to the end…*

To all who find their death away for the life of others.

To all who is troubled by his conscience and loyalty...

To all who violently love their homes and whom swords was broken without touching their determination…

To all the mothers who bring, raise, and present obliged us to have an ever-increasing recognition of their greatness.

To our country

To the souls of Palestine martyrs

To the freedom fighters

To whom their guidance and supports made this work possible

Acknowledgment

Our thanks go first to our advisor Dr. Zuhdi Salhab. His guidance and support made this work possible. His constant encouragement, intuitive wisdom, and resolute leadership were instrumental in completing this work.

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And, finally, our ultimate thanks go to all lecturers, doctors, engineers, and laboratories supervisors. For their efforts and their nice dealing with us improved our characters to become successful Engineers in the future.

Abstract

The basic principle of the system utilizes the high pressure of the exhaust gases that is directed through a nozzle which increases the velocity of these gases. This high velocity gas is made to impinge on the turbine rotor which rotates at high speeds. This rotor is coupled to the compressor of air conditioning system.

In this project two compressors are to be used, one which connected to the engine by belt and the other which coupled to the turbine and driven by the exhaust gases.

The power produced from the turbine will be calculated and decide if this power is enough to drive the air conditioning compressor or not.

In this project a control is being used between the two compressors, the conventional compressor is used in high demand (high load or at idle speed) but the other compressor is used in a good condition (high speed or at low load).

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CHAPTER ONE

INTRODUCTION

INTRODUCTION

1.1 General Introduction

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

If this is done by a different source, fuel consumption can decrease dramatically. Therefore it is a very real need for reduce the load on the engine which decrease the fuel consumption.

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

The most part which can reduce the load on the engine is the air conditioning compressor.

As known that IC engines lose 42% of their energy to exhaust. So a number of methods to increase the performance of the internal combustion engines have been established. A better method of utilizing the exhaust is being achieved by this project. The method uses the exhaust gases pressure from an optimal sized engine currently used. This pressure is being used and sent through a nozzle or set of nozzles and generates high velocity gases at exit. This high velocity gas is being utilized to drive a turbo compressor (turbine coupled with a compressor of air conditioning) so turbine drives the compressor of air conditioning and so lower the loads on the engine.

The main aims of this project are:

1. Taking advantage of the exhaust gases.
2. To reduce the load on the engine.
3. Optimize it to be applicable to current trend of automobiles.

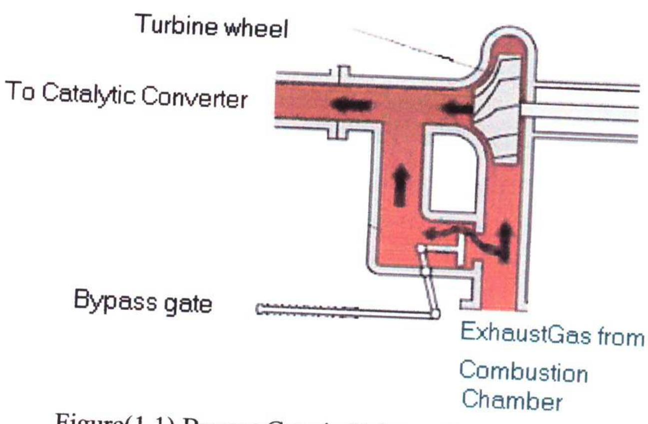
1.2 System Overview

The main purpose of this project is to reduce the load on the engine which decrease the fuel consumption and increase the life of the engine in the same time.

There are many parts connected to the crankshaft of the engine by a belt and rotate with it. The main part which considers the more one which effect on the engine is the compressor of air conditioning.

The basic principle of this project is to use the energy of the exhaust gases to drive turbine and use the rotation of the turbine to drive the compressor of air conditioning.

Bypass gate figure (1.1) will be used on the exhaust system, and controlled mechanically; the purpose of this bypass gate is to directing the exhaust gases to the turbine in some condition.



Figure(1.1) Bypass Gate in Exhaust System

So that two compressors in the same air conditioning circuit will be used. The first one is the compressor that will be used normally in the car and connected to the engine by a belt; this compressor will be used in difficult conditions such as at idle speed or on high engine loads. The second compressor (the purpose of this project) is used in the time when there are no high loads on the engine, or at high speed.

1.4 About this Report

This introductory report is divided into several chapters most of them present a subpart of the system. This report clarifies the idea of the project and explains the system design methodology.

Chapter one: introduction

This chapter gives an introductory aspect of the system, it clarifies the system as a practical idea and estimated the time needed to be done.

Chapter two : Exhaust System

This chapter shows the component of exhaust system ,the function of it ,and many phenomena occurs in this system.

Chapter three : Turbine

This chapter describe the type of turbines, the application of it, the suitable turbine for this project, and its properties.

Chapter four: Air conditioning system

This chapter talk about conventional air conditioning and describe hybrid air conditioning cycle.

Chapter five : Power transmission

This chapter describe the way of transmit the power from turbine to air conditioning compressor and the length of the belt.

Chapter six : Shaft design and Bearing lubrication

This chapter discusses the formula to design the shaft diameter and the lubrication of journal bearing.

Chapter Seven: Engine Performance

This chapter discusses the effect of turbine on the engine performance.

Chapter Eight: Test bench specification

This chapter talks about the equipment that used and test equipment fitting.

Chapter Nine: Power of turbine

This chapter describes the formula for the power input and output from turbine.

Chapter Ten: Conclusions and Recommendations

This chapter explain the conclusion of the project and the recommendation.

1.5 Project Schedule

Task/Week	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
Collecting Information about the project																
Reading																
Introduction																
Design Prototype And System selection																
Cycle analysis																
Cycle components																
Project Documentation																

Table 1.1 Project time-schedule for first semester

Task/Week	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
Installation the turbine																
Running ICE																
Pressure and temperature measure																
Installation alternator																
Writing Documentation																

Table 1.2 Project time-schedule for second semester

CHAPTER TWO

EXHAUST SYSTEM

EXHAUST SYSTEM

2.1 Introduction

Today internal combustion engines in cars, trucks, motorcycles, aircraft, construction machinery and many others, most commonly use a four-stroke cycle. The four strokes refer to intake, compression, combustion (power) and exhaust strokes that occur during two crankshaft rotations per working cycle of the Gasoline engine and Diesel engine.

The exhaust stroke is the fourth of four stages in an internal combustion engine cycle. In this stage, gases remaining in the cylinder from the fuel ignited during the compression step are removed from the cylinder through an exhaust valve at the top of the cylinder. The gases are forced up to the top of the cylinder as the piston rises and are pushed through the opening valve which then closes to allow fresh air/fuel mixture to enter into the cylinder.

2.2 Description of exhaust system

The purpose of the exhaust/emission system is to deaden the sound made by the internal combustion engine, reduces the temperature of the exhaust, and controls the emissions that come out of the vehicle.

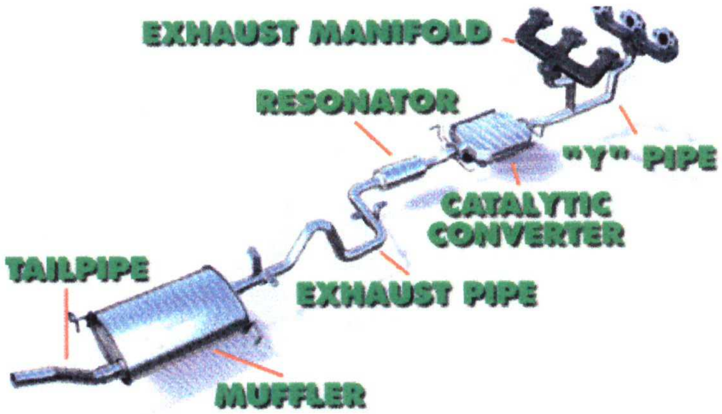
Vehicles create harmful gases during the combustion of fuel. The catalytic converter in the exhaust system turns these gases into mostly harmless ones that do much less damage to the environment.

2.3 Component of exhaust system

Figure (2.1) shows the component of exhaust system

1. Exhaust manifold.

2. Catalytic converter.
3. Muffler.
4. Resonator.
5. Exhaust pipe.
6. Tail pipe.



Figure(2.1) component of exhaust system

2.4 Exhaust flow

After combustion is completed and the resulting high-pressure gases have been used to transfer work to the crankshaft during the expansion stroke, these gases must be removed from the cylinder to make room for the air-fuel charge of the next cycle. The exhaust process that does this occurs in two steps, exhaust blowdown followed by the exhaust stroke.

2.4.1 Blowdown

Exhaust blowdown occurs when the exhaust valve starts to open towards the end of the power stroke, somewhere around 60° to 40° bBDC. At this time, pressure in the cylinder is still at about 4-5 atmospheres and the temperature is upwards of 1000 K. Pressure in the exhaust system is about one atmosphere, and when the valve is opened the resulting pressure differential causes a rapid flow of exhaust gases from the cylinder, through the valve, into the exhaust system.

2.4.2 Exhaust stroke

After exhaust blowdown, the piston passes BDC and starts towards TDC in the exhaust stroke. The exhaust valve remains open. Pressure in the cylinder resisting the piston in this motion is slightly above the atmospheric pressure of the exhaust system. The difference between cylinder pressure and exhaust pressure is the small pressure differential caused by the flow through the exhaust valves as the piston pushes the gases out of the cylinder. The exhaust valve is the greatest flow restriction in the entire exhaust system and is the location of the only appreciable pressure drop during the exhaust stroke.

2.5 Exhaust temperature

The average temperature in the exhaust system of a typical CI engine will be 650-750C^0. This is lower than SI engine exhaust because of the larger expansion cooling that occurs due to the higher compression ratios of CI engines. If the maximum temperature in a CI engine is about the same as in an SI engine, the temperature when the exhaust valve opens can be several hundred degrees less. The overall lean equivalence ratio of a CI engine also lowers all cycle temperatures from combustion on.

Exhaust temperature of an engine will go up with higher engine speed or load, with spark retardation, and/or with an increase in equivalence ratio.

2.6 Exhaust gas recirculation (EGR)

Exhaust gas recirculation (EGR) is a nitrogen oxide (NOx) emissions reduction technique used in most gasoline and diesel engines.

EGR works by recirculating a portion of an engine's exhaust gas back to the engine cylinders. Intermixing the incoming air with recirculated exhaust gas dilutes the mixture with inert gas, lowering the adiabatic flame temperature and (in diesel engines) reducing the amount of excess oxygen. The exhaust gas also increases the specific heat capacity of the mixture, lowering the peak combustion temperature. Because NOx formation progresses much faster at high temperatures, EGR serves to limit the generation of NOx. NOx is primarily formed when a mixture of nitrogen and oxygen is subjected to high temperatures.

2.7 Exhaust Backpressure

Exhaust backpressure can cause a variety of problems. A plugged catalytic converter can strangle engine breathing and cause a big drop in engine performance and fuel economy. And if the converter plugs up completely, it can make the engine stall. The same thing can happen if a muffler, resonator or double walled exhaust pipe collapses internally.

Another thing cause backpressure is turbocharger when the exhaust gases clash its blades, backpressure increases as the size of the turbo decreases and inversely, back pressure decreases as the size of the turbo increases, the turbine which uses in this project may cause backpressure.

Anything that restricts exhaust flow will create excessive backpressure in the exhaust system.

2.7.1 Exhaust Backpressure symptoms

The classic symptoms of too much backpressure include things like a lack of high speed power, poor fuel economy and even overheating. Anything that backs up exhaust pressure into the engine will also back up heat. About a third of the heat produced by combustion goes out the tailpipe as waste heat, so if the heat can't escape it can overload the cooling system and make the engine run hotter than normal, especially at highway speeds.

If there is a complete blockage in the exhaust, the engine may start and idle fine for a minute or two, then die as backpressure builds up and strangles the engine. In some instances, backpressure may buildup to such a degree that it blow out a pipe connector or the converter shell. That makes diagnosis a lot easier, but in most cases you may not be sure if there is an exhaust restriction or not. So in this project the backpressure must be measured and must be avoided.

2.8 The advantages of exhaust gases

There is significant effect of emission resulting from car exhaust gases to human life because of its production of gases such as carbon monoxide (CO), hydrocarbons, nitrogen oxides (NOx), partly unburnt fuel, and particulate matter. Which causes eye, Throat, lung irritation and breathing difficulties for human, and cause acid rain, destroy ozone at the stratosphere and produce photochemical smog.

In the same time although of this effects to human health and environment **there are some advantages of the exhaust gases such as.**

1. **Generate electricity**

   Using the temperature and pressure of the exhaust gas to drive a turbine which coupled to generator and create electricity by a different ways.

2. **Turbocharger**

A turbocharger, or turbo, is an air compressor used for forced-induction of an internal combustion engine. Like a supercharger, the purpose of a turbocharger is to increase the mass of air entering the engine to create more power. However, a turbocharger differs in that the compressor is powered by a turbine driven by the engine's own exhaust gases.

The other advantage of the exhaust gases uses its energy to drive a turbine which drives the compressor of the air conditioning, and this advantage is the goal of this project.

CHAPTER THREE

TURBINE

TURBINE

3.1 Introduction

A turbine is any of various rotary machines that convert the kinetic energy in a stream of fluid (gas or liquid) into mechanical energy by passing the stream through a system of fixed and 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 is an equal and opposite reaction. They 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 jet aircraft.

3.2 Uses of turbines

Almost all electrical power on Earth is produced with a turbine of some type, the exceptions being solar panels and fuel cells. All jet engines rely on turbines to supply mechanical work from their fuel, as do all nuclear warships and power plants.

Turbines are often part of a larger machine. A Gas turbine, for example, may refer to an internal combustion machine that contains a turbine, compressor, combustor and alternator.

Piston engines, especially for aircraft, can use a turbine powered by their exhaust to drive an intake compressor, a configuration known as a turbocharger (turbine supercharger) or colloquially as a "turbo".

Turbines can have incredible power density (with respect to volume and weight). This is because of their ability to operate at very high speeds. The Space Shuttle fuel pump turbine, for example, is slightly larger than an automobile engine and produces 25,000 hp (19 MW).

3.3 Gas turbine

A gas turbine, also called a combustion turbine, is a rotary engine that extracts energy from a flow of combustion gas. It has an upstream compressor coupled to a downstream turbine, and a combustion chamber in-between. (Gas turbine may also refer to just the turbine element).

Energy is added to the gas stream in the combustor, where air is mixed with fuel and ignited. Combustion increases the temperature, velocity and volume of the gas flow. This is directed through a (nozzle) over the turbine's blades, spinning the turbine and powering the compressor.

Energy is extracted in the form of shaft power, compressed air and thrust, in any combination, and used to power aircraft, trains, ships, generators, and even tanks.

3.3.1The application of gas turbine

The gas turbine is used in many applications, and the application determines in most parts the type of gas turbine best suited. The three major types of applications are aircraft propulsion, power generation, and mechanical drives.

3.3.1.1 Aircraft propulsion

The aircraft propulsion gas turbines can be subdivided into two major categories, the jet propulsion and turboprop engines. The jet engine consists of a gasifier section and a propulsive thrust section . The gasifier section is the section of the turbine, which produces high pressure and temperature gas for the power turbine. This comprises a compressor section and a turbine section. The sole job of the gasifier turbine section is to drive the gas turbine compressor. This section has one or two shafts.

The two-shaft gasifier section usually exists in the new high-pressure type gas turbine where the compressor produces a very high pressure ratio, and has two different sections. Each section comprises many stages. The two different compressor sections consist of the low-pressure compressor section, followed by a high-pressure section. Each section may have between 10 to 15 stages. The jet engine has a nozzle following the gasifier turbine, which produces the thrust for the engine. In the newer jet turbines the compressor also has a fan section ahead of the turbine and a large amount of the air from the fan section bypasses the rest of the compressor and produces thrust. The thrust from the fan amounts to more than the thrust from the exhaust.

The jet engine has led the field of gas turbines in firing temperatures. Pressure ratio of 40:1 with firing temperatures reaching 1370 C^0 is now the mode of operation of these engines.

3.3.1.2 Power generation

The power generation turbines can be further divided into three categories:

1. Small standby power turbines less than 2-MW. The smaller sized of these turbines in many cases have centrifugal compressors driven by radial inflow turbines, the larger units in this range are usually axial-flow compressors sometimes combined with a centrifugal compressor at the last stage, and are operated by axial turbines.
2. Medium-sized gas turbines between 5–50 MW are a combination of aeroderivative and frame type turbines. These gas turbines have axial-flow compressors and axial-flow turbines.
3. Large power turbines over 50–480 MW, these are frame type turbines. The new large turbines are operating at very high firing temperatures of about 2400 ◦ F (1315C) with cooling provided by steam, at pressure ratios approaching 35:1.

3.3.1.3 Mechanical drives

Mechanical drive gas turbines are widely used to drive pumps and compressors. Their application is widely used by offshore and petro- chemical industrial complexes. These turbines must be operated at various speeds and thus usually have a gasifier section and a power section. These units in most cases are aero-derivative turbines, which were originally designed for aircraft application. There are some smaller frame type units, which have been converted to mechanical drive units with a gasifier and power turbine.

3.3.2 Parts of gas turbine

1. The Air Compressor.
2. The Combustion Chamber.
3. The Turbine.

This gas turbine can uses in the vehicle put in this project only one part of gas turbine will be used its the turbine part.

3.4 Turbine

The turbine, which consists of a turbine wheel and a turbine housing, converts the engine exhaust gas which drive the turbine as shown in figure (3.1) into mechanical energy to drive the compressor of the air conditioning.

The gas, which is restricted by the turbine's flow cross-sectional area, results in a pressure and temperature drop between the inlet and outlet. This pressure drop is converted by the turbine into kinetic energy to drive the turbine wheel.

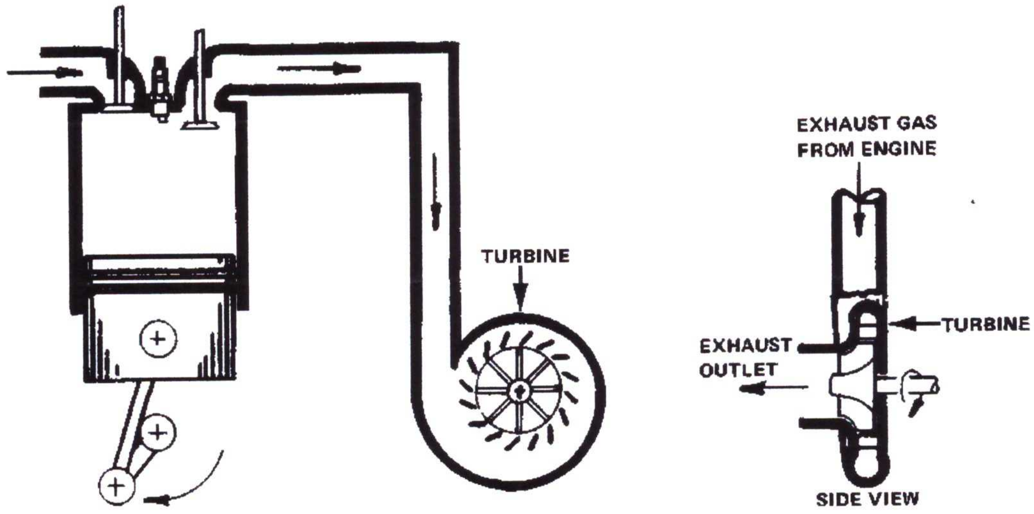


Figure (3.1)-Engine driving gas turbine

There are two main types of turbine: axial and radial turbine:-

3.4.1 Axial Turbine

The turbine extracts kinetic energy from the expanding gases that flow from the exhaust, converting this energy into shaft horsepower.

In the axial-flow type turbine flow through the wheel is only in the axial direction. and it is made up of stationary nozzles (vanes or diaphragms) and rotating blades (buckets) attached to a turbine wheel (disc). Turbines are divided into three types: "impulse," "reaction," and a combination of the two designs called "impulse-reaction."

The energy drop to each stage is a function of the nozzle area and airfoil configuration. Turbine nozzle area is a critical part of the design: too small and the nozzles will have a tendency to "choke" under maximum flow conditions, too large and the turbine will not operate at its best efficiency.

3.4.1.1 Impulse

In the impulse type turbine there is no net change in pressure between rotor inlet and rotor exit. Therefore, the blades Relative Discharge Velocity will be the same as its relative Inlet Velocity. The nozzle guide vanes are shaped to form passages, which increase the velocity and reduce the pressure of the escaping gases.

3.4.1.2 Reaction

In the reaction turbine the nozzle guide vanes only alter the direction of flow. The decrease in pressure and increase in velocity of the gas is accomplished by the convergent shape of the passage between the rotor blades.

Note that in the impulse turbine no pressure drop or expansion occurs across the moving rows. While in the reaction turbine the fixed nozzles perform the same function as in the impulse turbine.

Turbines may be either single- or multiple-stage. When the turbine has more than one stage, stationary vanes are located upstream of each rotor wheel. Therefore, each set of stationary vanes forms a nozzle vane assembly for the turbine wheel that follows. The rotor wheels may or may not operate independently of each other, depending upon the power requirements of the turbine.

The turbine wheel, or turbine blade and disc assembly, consists of the turbine blades and the turbine disc. The blades are attached to the disc using a "fir-tree" design. While almost all blades employ the fir-tree root design, the blades themselves may be solid or hollow (to provide for blade cooling), and with or without tip shrouds. Some blades, primarily in the lower pressure stages where the blade length can be very long, also include a damping wire through sets of blades.

3.5 The radial-inflow turbine

The radial-inflow turbine has been in use for many years. Its first appeared as a practical power-producing unit in the hydraulic turbine field. Basically a centrifugal compressor with reversed flow and opposite rotation, the radial-inflow turbine was the first used in jet engine flight in the late 1930s. It was considered the natural combination for a centrifugal compressor used in the same engine.

Designers thought it easier to match the thrust from the two rotors and that the turbine would have a higher efficiency than the compressor for the same rotor because of the accelerating nature of the flow.

The performance of the radial-inflow turbine is now being investigated with more interest by the transportation and chemical industries: In transportation, this turbine is used in turbochargers for both spark ignition and diesel engines; in aviation, the radial-inflow turbine is used as an expander in environmental control systems; and in the petrochemical industry, it is used in expander designs, gas liquefaction expanders, and other cryogenic systems. Radial-inflow turbines are also used in various small gas turbines to power helicopters and as standby generating units.

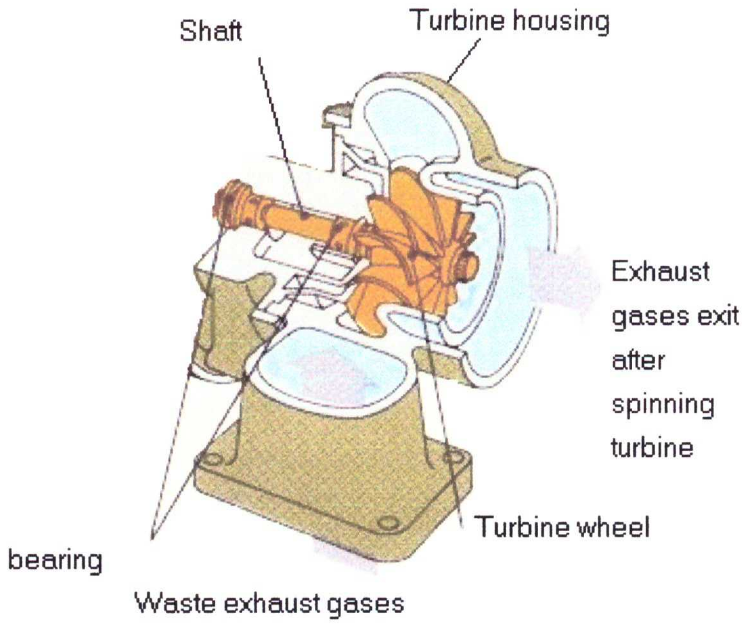
The radial-inflow turbine's greatest advantage is that the work produced by a single stage is equivalent to that of two or more stages in an axial turbine.

This phenomenon occurs because a radial-inflow turbine usually has a higher tip speed than an axial turbine. Since the power output is a function of the square of the tip speed ($P \propto U^2$) for a given flow rate, the work is greater than in a single-stage axial-flow turbine.

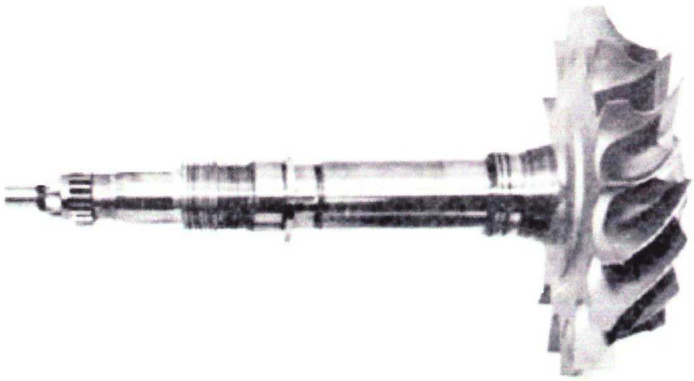
The radial-inflow turbine has another advantage: Its cost is much lower than that of a single or multistage axial-flow turbine. The radial-inflow turbine has a lower turbine efficiency than the axial-flow turbine; however, lower initial costs may be an incentive to choosing a radial-inflow turbine.

The choice of turbine depends on the application, though it is not always clear that any one type is superior. For small mass flows, the radial machine can be made more efficient than the axial one. The radial turbine is capable of a high-pressure ratio per stage than the axial one.

In inward flow radial turbine, gas enters in the radial direction and leaves axially at outlet. The rotor, which is usually manufactured of cast nickel alloy, has blades that are curved to change the flow from the radial to the axial direction as shown in figure (3.2), and figures (3.3) show photographs of the radial turbine.



Figure(3.2) direction of gas flow



Figure(3.3)-Radial Turbine

3.6 Radial flow turbines versus axial flow turbines

An axial flow turbine has the following advantages.

- It can be designed for a very large range of loadings with a large variation in size, speed and efficiency depending on the requirements.
- For a highly loaded design weight is lower.
- If the required expansion ratio is such that more than one radial turbine is required then the inter-turbine duct is complex. This leads to multi-stage radial turbines rarely being considered. There are no such problems with axial flow turbines.

The radial turbine has the following advantages.

- Radial turbines are capable of up to 8 :1 expansion ratio in a single stage. For an axial flow turbine this will require at least two stages.
- Its cost is much lower than that of a single or multistage axial-flow turbine.
- It has a shorter length than two axial stages, but similar to one.

- The work produced by a single stage is equivalent to that of two or more stages in an axial turbine.
- It's used in small size.

In summary, from the advantages of the radial turbine its seen that it's the most suitable and popular type for automotive application and for this project.

3.7 The turbocharger turbine

The turbocharger turbine is radial turbine and it's very similar to the specifications of the radial turbine which will be used in this project.

In the volute of such radial or centripetal turbines, exhaust gas pressure is converted into kinetic energy and the exhaust gas at the wheel circumference is directed at constant velocity to the turbine wheel. Energy transfer from kinetic energy into shaft power takes place in the turbine wheel, which is designed so that nearly all the kinetic energy is converted by the time the gas reaches the wheel outlet.

3.7.1 Operating characteristics

The turbine performance increases as the pressure drop between the inlet and outlet increases, i.e. when more exhaust gas is dammed upstream of the turbine as a result of a higher engine speed, or in the case of an exhaust gas temperature rise due to higher exhaust gas energy. Figure (3.3) is a performance map that shows the effect of turbine inlet temperature and pressure, while power is dependent on the efficiency of the unit, the flow rate, and the available energy (turbine inlet temperature).

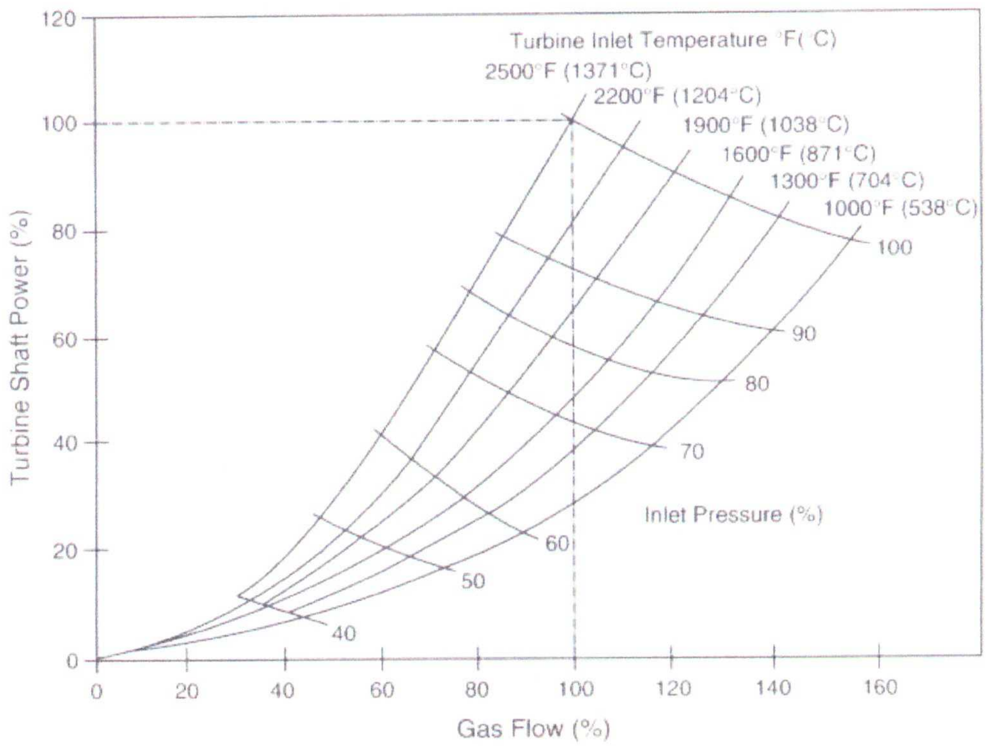


Figure (3.3)-Turbine performance

The turbine's characteristic behavior is determined by the specific flow cross-section, the throat cross-section, in the transition area of the inlet channel to the volute. By reducing this throat cross-section, more exhaust gas is dammed upstream of the turbine and the turbine performance increases as a result of the higher pressure ratio. The turbine's flow cross-sectional area can be easily varied by changing the turbine housing.

3.8 Turbine Placement

3.8.1 Heat

The turbine mustn't come near anything that will be affected by heat and there must be plenty of room around them.

3.8.2 Heat Retention

The power used to power a turbine is exhaust gas velocity. When temperature drops in an exhaust system, so does it's velocity. This is important when considering placement of a turbine because the farther the turbine is from the engine, the greater the temperature drop will be. In this respect, putting the turbine as close to the engine as practical will give best turbine performance.

3.8.3 Turbine lubrication

The turbine designed to use the engine lubricating oil as in the turbocharger. The actual type and viscosity will be dictated by the engine but, in general it will operate well in any oil that will work in the engine.

It's not necessary to install a special oil filter in the turbine oil line if the engine is equipped with an oil filter. If an engine is not equipped with a full-flow oil filter, then a filter is definitely recommended in the oil-inlet line and it should be of the type with a built-in bypass so oil will still get to the turbine even if the filter is clogged with dirt. It's not uncommon for a turbine to fail from lack of oil because the oil filter has not serviced. On the other hand, the life of turbine running on dirty oil can be measured in hours.

3.8.4 Turbine cooling

The other using of oil its used for cooling by the circulation of the oil from the engine to the turbine. When the engine is shutting down then the oil and coolant stop flowing. If the engine shutting down when the turbine is hot, the oil can burn and build up in the unit (known as "coking") and eventually cause it to leak oil (this is the most common turbine problem). It is a good idea to let the engine idle for at least 2 minutes. This will cool the turbine down and help prevent coking.

CHAPTER FOUR

VEHICLE AIR CONDITIONING SYSTEM

VEHICLE AIR CONDITIONING SYSTEM

4.1 Introduction

Air-conditioning systems are common on modern motor vehicles. Over half of new vehicles are fitted with them as standard and this proportion is on the increase.

They work in much the same way as a refrigerator. An evaporator converts the refrigerant from a liquid into a gas and there is a condenser for reversing the process. The evaporator is normally located just behind the dashboard so that it can absorb heat from the interior of the vehicle, while the condenser is at the front, to ensure a free flow of air over it to help cool it down.

These effects are used to cool down the interior of a vehicle. The system is entirely enclosed and nobody should be exposed to the refrigerant chemical during normal operation.

4.2 Conventional air conditioning system

As shown in figure (4.1) The main components of an air-conditioning system are:-

1. **Cabin / Pollen Filter**

Passenger compartment filters (Pollen or Cabin filter) improve the incoming air into the vehicle interior.

The filter is located in the vehicle fan air intake duct where it filters out dust, contamination, pollen and bacteria. If a filter element with a layer of activated charcoal is used, even gases with unpleasant odours can be eliminated from the vehicle interior.

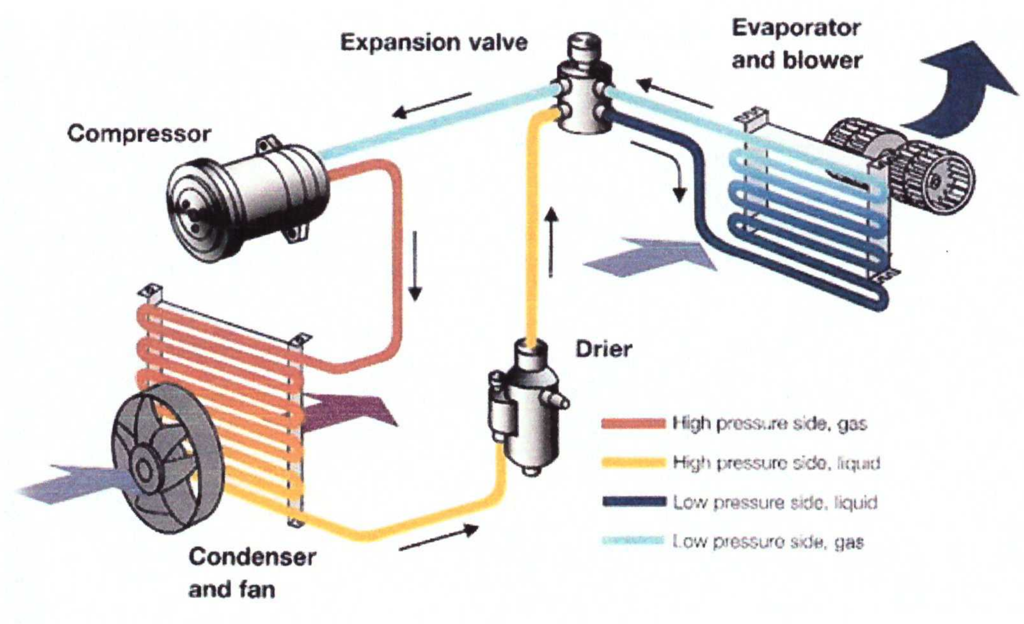


Figure (4.1) The main components of an air-conditioning system

2. Air-Conditioning Compressors

The AC compressor is driven by the engine via a poly-V-belt or multiple V-belts and compresses or pumps the refrigerant in the system. There are different types of compressors.

The refrigerant coming from the evaporator is sucked into the compressor in the gaseous state at low temperature and low pressure, compressed and then transferred to the condenser in the gaseous state at high temperature and high pressure.

3. Condensers

The condenser is required to cool down the refrigerant heated up by being compressed.

The hot refrigerant gas flows into the condenser at the top transferring heat to the surroundings through the piping and fins. After cooling the refrigerant exits the condenser at the lower connection in a liquid state.

4. Receiver Drier

The purpose of the filter/drier is to remove foreign particles as well as moisture from the refrigerant. In addition it also serves as a refrigerant reservoir. It has a compensation chamber and in some cases an observation glass.

The liquid refrigerant enters the filter/drier through the inlet connection, flows through a hydroscopic filter and exits the component through the outlet line.

The upper section of the filter/drier serves as a compensation chamber and the bottom section as a refrigerant reservoir and filter chamber.

5. Evaporator

The evaporator is located in the evaporator housing and operates like a heat exchanger, it consists of numerous cooling coils connected by fins. As the high pressure liquid refrigerant, metered by the expansion valve or the fixed orifice tube, is injected into the evaporator, it is no longer under pressure. The evaporating temperature of the liquid refrigerant at this pressure is below that of the air passing across the evaporator. Therefore, the refrigerant absorbs heat from the air and evaporates into a vapour, thus cooling the air.

6. Expansion Valve

The expansion valve is the separation point between the high and low pressure stages in the refrigerant circuit and is installed in front of the evaporator. In order to achieve optimum refrigeration capacity in the evaporator the refrigerant flow is regulated by the expansion valve depending on the temperature. Expansion valves are available in different versions.

The liquid refrigerant coming through the filter from the condenser flows through the valve body and expands into the outlet line to the evaporator. This high reduction in pressure results in a decrease in the temperature. In order to achieve optimum refrigeration capacity in the evaporator the refrigerant flow is regulated by the expansion valve depending on the temperature.

If the temperature of the refrigerant increases entering the evaporator, the thermostat in the expansion valve opens and with it the valve, increasing the flow of refrigerant to the evaporator.

If the temperature of the refrigerant drops entering the evaporator the thermostat closes and with it the valve, decreasing the flow to the evaporator

7. Condenser Fans

Condenser fans are used in vehicles with air-conditioning. It is also possible to replace the present fan with a fan with greater capacity.

Condenser fans are designed as suction-type fans (behind condenser toward rear of vehicle) or as pressure-type fans (in front of condenser toward front of vehicle). They serve refrigerant returning to liquid in the condenser and engine cooling in all vehicle operating states.

8. Fittings

There are different variants of fittings. It should be noted that connections for flexible refrigerant pipes have definite mechanical characteristics at one end:

- A cylindrical straight connecting pipe whose diameter on the outside fits the diameter on the inside of the pipe. The connecting pipe has radial grooves so that the pipe does not slip.
- A connecting sleeve which is pressed by a press or by pliers and which, due to this connects the connecting pipe with the connecting hose tightly.

The other end has, depending on the manufacturer, different connection possibilities.

The function of the fittings is to connect the single components which carry the refrigerant of the air-conditioning by hoses or by aluminium pipes.

9. Pressure Switches

Pressure switches serve to protect the air-conditioning from damage, resulting from excessively high or low pressures.

Different types are:

- Low pressure switches
- High pressure switches
- Trinary switches.

Trinary switches contain the high and low pressure switches as well as an additional switching contact for the condenser fan.

As a rule the pressure switch (pressure monitor) is installed on the high pressure side of the air-conditioning. It switches off the power to the compressor clutch when the pressure is too high (approx. 26-33 bar) and switches it back on when the pressure decreases (approx. 5 bar).

When the pressure is too low (2 bar) the power supply is also interrupted in order to avoid compressor damage resulting from insufficient lubrication and switched back on when the pressure increases (2.2 bar). The third switching contact in the trinary switch controls the electric condenser fan to ensure optimum condensation of the refrigerant in the condenser.

4.3 Hybrid air conditioning in vehicle

4.3.1 Introduction

In this project two compressor will be used in the same air conditioning cycle, the first one is the conventional one, and the other is the compressor which driven by a turbine and this is the principle of this project.

4.3.2 Air conditioning cycle

air conditioning cycle have two compressor connecting in parallel as shown in figure (4.2), one of them(compressor 1) connecting to the engine by a belt and this one is used in a difficult conditioning(at high engine load or at idle speed),the other one(compressor 2) is connecting to the turbine and used at good conditioning(at low engine load or at high speed).

The other component of this cycle is as the conventional air conditioning cycle except Check valve and Solenoid valve are used to control the operation of the two compressors.

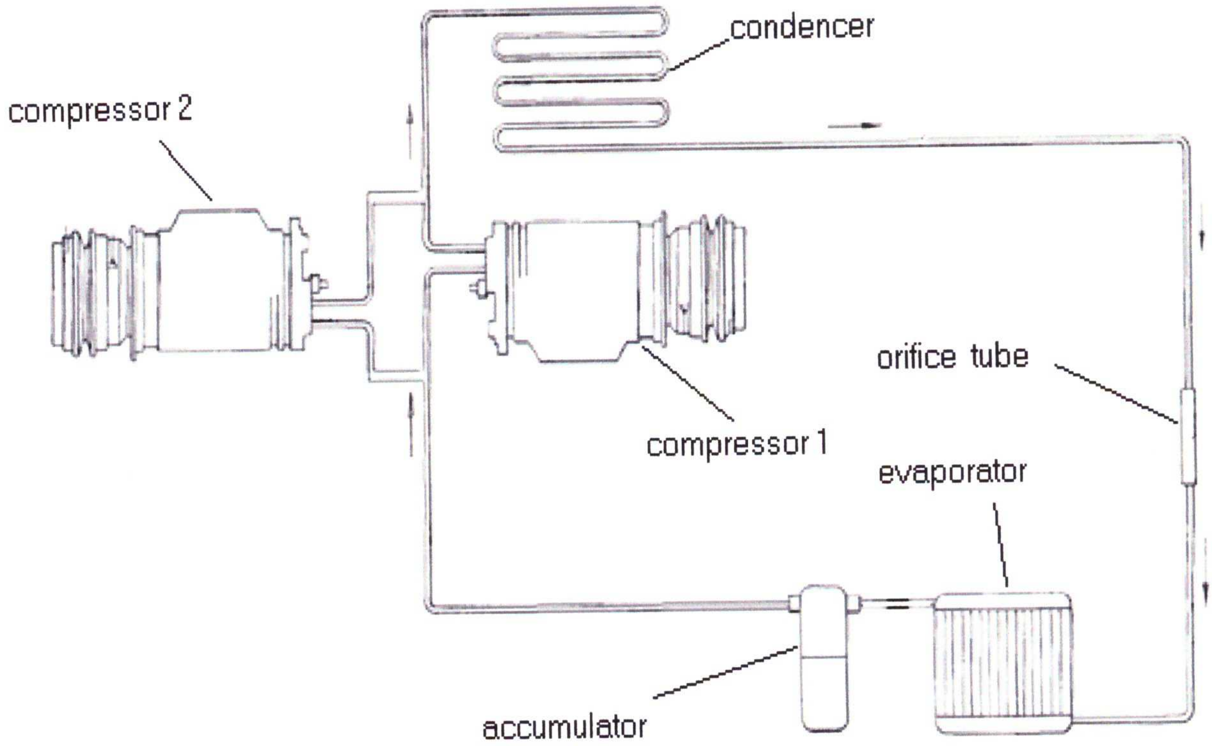


Figure (4.2)Hybrid air condition cycle

CHAPTER FIVE

POWER TRANSMISSION

POWER TRANSMISSION

5.1 Introduction

Belts, ropes, chains, and other similar elastic or flexible machine elements are used in Conveying systems and in the transmission of power over comparatively long distances.

It often happens that these elements can be used as a replacement for gears, shafts, bearings, and other relatively rigid power-transmission devices. In many cases their use simplifies the design of a machine and substantially reduces the cost.

In addition, since these elements are elastic and usually quite long, they play an important part in absorbing shock loads and in damping out and isolating the effects of vibration. This is an important advantage as far as machine life is concerned.
Most flexible elements do not have an infinite life when they are used, it is important to establish an inspection schedule to guard against wear, aging, and loss of elasticity.
The elements should be replaced at the first sign of deterioration.

5.2 Belts

The four principal types of belts are shown, with some of their characteristics, in Table 5–1. 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 a certain minimum distance, depending upon the belt type and size, to operate properly. Other characteristics of belts are:

- They may be used for long center distances.

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

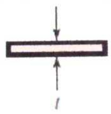
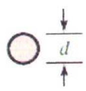
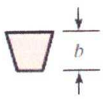
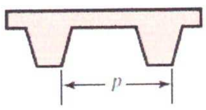
Belt Type	Figure	Joint	Size Range	Center Distance
Flat		Yes	$t = \begin{cases} 0.03 \text{ to } 0.20 \text{ in} \\ 0.75 \text{ to } 5 \text{ mm} \end{cases}$	No upper limit
Round		Yes	$d = \frac{1}{8}$ to $\frac{3}{4}$ in	No upper limit
V		None	$b = \begin{cases} 0.31 \text{ to } 0.91 \text{ in} \\ 8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

Table (5.1) Characteristics of Some Common Belt Types

The belt that used to transmission the power between turbine and air conditioning compressor is V belt.

5.3 V Belts

5.3.1 Advantages 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 is increase the pulling capacity of the belt and consequently result in increase the power transmitting capacity.

2- V-belt have short center distance, which results in compact construction .

3-They permit high speed reduction even up to 7 to 1.

4-Smooth and quiet operation even at high speed.

5-Postive drive, the slip is negligible due to wedge action.

6-They can operate in any position even when the belt is vertical.

Disadvantages

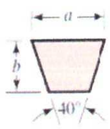
1- The efficiency of the V-belt is lower than that of flay belt.

2-The construction of V grooved pulley is complicated and costly.

5.3.2 V belt design

The cross-sectional dimensions of V belts have been standardized by manufacturers, with each section designated by a letter of the alphabet for sizes in inch dimensions. Metric sizes are designated in numbers.

Dimensions, minimum sheave diameters, and the horsepower range for each of the lettered sections are listed in Table 5-2



Belt Section	Width a, in	Thickness b, in	Minimum Sheave Diameter, in	hp Range, One or More Belts
A	$\frac{1}{2}$	$\frac{11}{32}$	3.0	$\frac{1}{4}$–10
B	$\frac{21}{32}$	$\frac{7}{16}$	5.4	1–25
C	$\frac{7}{8}$	$\frac{17}{32}$	9.0	15–100
D	$1\frac{1}{4}$	$\frac{3}{4}$	13.0	50–250
E	$1\frac{1}{2}$	1	21.6	100 and up

Table (5-2) Standard V-Belt Sections

Section	Circumference, in
A	26, 31, 33, 35, 38, 42, 46, 48, 51, 53, 55, 57, 60, 62, 64, 66, 68, 71, 75, 78, 80, 85, 90, 96, 105, 112, 120, 128
B	35, 38, 42, 46, 48, 51, 53, 55, 57, 60, 62, 64, 65, 66, 68, 71, 75, 78, 79, 81, 83, 85, 90, 93, 97, 100, 103, 105, 112, 120, 128, 131, 136, 144, 158, 173, 180, 195, 210, 240, 270, 300
C	51, 60, 68, 75, 81, 85, 90, 96, 105, 112, 120, 128, 136, 144, 158, 162,173, 180, 195, 210, 240, 270, 300, 330, 360, 390, 420
D	120, 128, 144, 158, 162, 173, 180, 195, 210, 240, 270, 300, 330, 360,390, 420, 480, 540, 600, 660
E	180, 195, 210, 240, 270, 300, 330, 360, 390, 420, 480, 540, 600, 660

Table (5-3) Inside Circumferences of Standard V Belts

Belt section	A	B	C	D	E
Quantity to be added	1.3	1.8	2.9	3.3	4.5

Table (5-4) Length Conversion Dimensions

To specify a V belt, give the belt-section letter, followed by the inside circumference in inches (standard circumferences are listed in Table 5-3). For example, B75 is a B-section belt having an inside circumference of 75 in.

Calculations involving the belt length are usually based on the pitch length. For any given belt section, the pitch length is obtained by adding a quantity to the inside circumference (Tables 5–3 and 5-4). For example, a B75 belt has a pitch length of 76.8 in. Similarly, calculations of the velocity ratios are made using the pitch diameters of the sheaves, and for this reason the stated diameters are usually understood to be the pitch diameters even though they are not always so specified.

The minimum sheave diameters have been listed in Table 5-2

In the case of flat belts, there is virtually no limit to the center-to-center distance.

Long center-to-center distances are not recommended for V belts because the excessive vibration of the slack side will shorten the belt life materially. In general, the center -to- center distance should be no greater than 3 times the sum of the sheave diameters and no less than the diameter of the larger sheave. Link-type V belts have less vibration, because of better balance, and hence may be used with longer center-to- center distances.

The nominal speed ratio :

$$\mathbf{D/d=W_{1}/W_{2}} \text{ ....}(5.1)$$

Where:

D = pitch diameter of the large sheave.

d = pitch diameter of the small sheave.

W_1 = 20000 RPM

W_2 = 3000 RPM and Select (d) = 3cm.

Then : D/3=20000/3000

D=20 cm

Specify a trail center distance:

$$\mathbf{D < C < 3*(D+d)} \text{ ....}(5.2)$$

Then : 20 < C < 3*(20+3) → 20<C<69

Select C = 50cm

The pitch length Lp:

$$\mathbf{Lp=2C+\pi*(D+d)/2+(D-d)^2/4C} \text{ ....}(5.3)$$

$$Lp = 2*50+3.14*(20+3)/2+(20-3)^2/4*50 = 137.25 \text{ cm}$$

Then : Select the length of the belt 137cm.

CHAPTER SIX

SHAFT DESIGN AND BEARING LUBRICATION

SHAFT DESIGN AND BEARING LUBRICATION

Shaft design

6.1Introduction

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 elements such as gears, pulleys, flywheels, cranks, sprockets, and the like and controls the geometry of their motion. An axle is a nonrotating 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.

Because of the ubiquity of the shaft in so many machine design applications, there is some advantage in giving the shaft and its design a closer inspection. A complete shaft design has much interdependence on the design of the components. The design of the machine itself will dictate that certain gears, pulleys, bearings, and other elements will have at least been partially analyzed and their size and spacing tentatively determined.

6.2 Shaft Materials

Deflection is not affected by strength, but rather by stiffness as represented by the modulus of elasticity, which is essentially constant for all steels. For that reason, rigidity cannot be controlled by material decisions, but only by geometric decisions.

Necessary strength to resist loading stresses affects the choice of materials and their treatments. Many shafts are made from low carbon, cold-drawn or hot-rolled steel, such as ANSI 1020-1050 steels.

Significant strengthening from heat treatment and high alloy content are often not

6.3 Providing for Torque Transmission

Most shafts serve to transmit torque from an input gear or pulley, through the shaft, to an output gear or pulley. Of course, the shaft itself must be sized to support the torsional stress and torsional deflection. It is also necessary to provide a means of transmitting the torque between the shaft and the gears. Common torque-transfer elements are:

- Keys
- Splines
- Setscrews
- Pins
- Press or shrink fits
- Tapered fits

6.4 Design the shaft diameter

6.4.1 Introduction

In most testing of those properties of materials that relate to the stress-strain diagram, the load is applied gradually, to give sufficient time for the strain to fully develop. Furthermore, the specimen is tested to destruction, and so the stresses are applied only once. Testing of this kind is applicable, to what are known as static conditions; such conditions closely approximate the actual conditions to which many structural and machine members are subjected.

The condition frequently arises, however, in which the stresses vary with time or they fluctuate between different levels. For example, a particular fiber on the surface of a rotating shaft subjected to the action of bending loads undergoes both tension and compression for each revolution of the shaft. If the shaft is part of an electric motor rotating at 1725 rev/min, the fiber is stressed in tension and compression 1725 times each minute. If, in addition, the shaft is also axially loaded (as it would be, for example, by a helical or worm gear), an axial component of stress is superposed upon the bending component. In this case, some stress is always present in any one fiber, but now the level of stress is Fluctuating. These and other kinds of loading occurring in machine members produce stresses that are called variable, repeated, alternating, or fluctuating stresses.

Often, machine members are found to have failed under the action of repeated or fluctuating stresses; yet the most careful analysis reveals that the actual maximum stresses were well below the ultimate strength of the material, and quite frequently even below the yield strength. The most distinguishing characteristic of these failures is that the stresses have been repeated a very large number of times. Hence the failure is called a fatigue failure.

When machine parts fail statically, they usually develop a very large deflection, because the stress has exceeded the yield strength, and the part is replaced before fracture actually occurs. Thus many static failures give visible warning in advance. But a fatigue failure gives no warning! It is sudden and total, and hence dangerous. It is relatively simple to design against a static failure, because our knowledge is comprehensive. Fatigue is a much more complicated phenomenon, only partially understood, and the engineer seeking competence must acquire as much knowledge of the subject as possible.

6.4.2 Design formula

The purpose of the section is to find the diameter of the shaft using the Endurance, Limit Fatigue Strength, and Endurance Limit Modifying Factors.

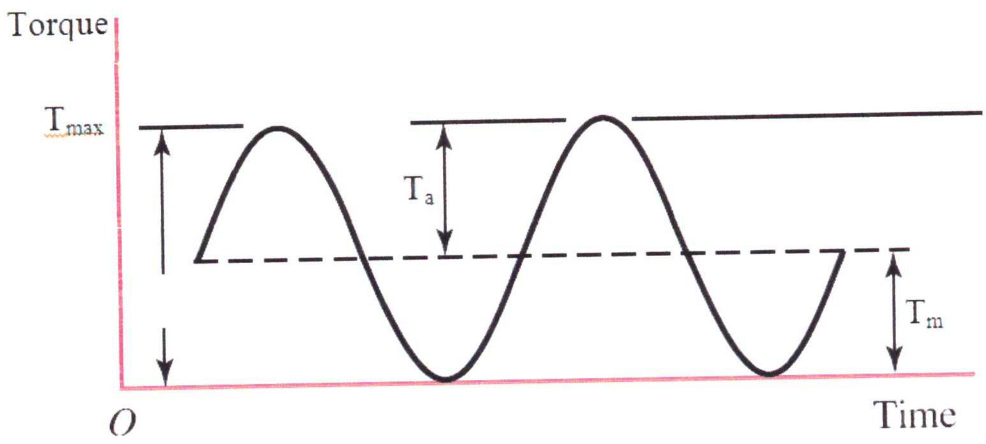


Figure (6.1) illustrates the torque-time traces

This Figure (6.1) illustrates the torque-time traces that occur. The components of torque are :-

Tm = midrange component

$Tmax$ = maximum torque

T_{min} = minimum torque = zero

Ta = amplitude component

$$Ta = \frac{Tmax - Tmin}{2} = \frac{Tmax - 0}{2} = 0.5\ T_{max} \text{ ....(6.1)}$$

$$T_m = \frac{Tmax + Tmin}{2} = \frac{Tmax + 0}{2} = 0.5\ T_{max} \text{ ....(6.2)}$$

$$T_{max} = p_{max} / \omega \text{ ....(6.3)}$$

Where:

p_{max} : maximum output power (watt)

ω : number of engine revolution(rev/min)

$$\tau_a = \frac{16\,Ta}{\pi d^3} \quad \text{..................(6.4)}$$

$$\tau_m = \frac{16\,Tm}{\pi d^3} \quad \text{..................(6.5)}$$

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 :- $$\frac{\tau_a}{S_e} + \frac{\tau_m}{S_{su}} = \frac{1}{n_f} \quad \text{..................(6.6)}$$

but : $S_{su} = 0.67\,S_{ut}$ ....(6.7)

where :

n_f : Fatigue factor of safety and the value must be more than 3

Sut : is the minimum tensile strength (kpsi)

and :

$$S_e = k_a k_b k_c k_d k_e k_f\, S'e \quad \text{..................(6.8)}$$

where :

S′e = rotary-beam test specimen endurance limit

And $$S'_e = \begin{cases} 0.5 S_{ut} & S_{ut} \le 200 \text{ kpsi } (1400 \text{ MPa}) \\ 100 \text{ kpsi} & S_{ut} > 200 \text{ kpsi} \\ 700 \text{ MPa} & S_{ut} > 1400 \text{ MPa} \end{cases} \quad \text{........(6.9)}$$

Se = endurance limit at the critical location of a machine 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, estimations are made by applying Marin factors to the endurance limit.

Surface Factor k_a

$$k_a = aS_{ut}^{b} \quad \text{..............(6.10)}$$

Where : Sut is the minimum tensile strength and a and b are to be found in Table(6.1)

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 (6.1)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:

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$$k_b = \begin{cases} (d/0.3)^{-0.107} = 0.879d^{-0.107} & 0.11 \le d \le 2 \text{ in} \\ 0.91d^{-0.157} & 2 < d \le 10 \text{ in} \\ (d/7.62)^{-0.107} = 1.24d^{-0.107} & 2.79 \le d \le 51 \text{ mm} \\ 1.51d^{-0.157} & 51 < d \le 254 \text{ mm} \end{cases} \quad \text{.................(6.11)}$$

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

Loading Factor kc

When fatigue tests are carried out with rotating bending, axial (push-pull), and torsional loading, the endurance limits differ with Sut. the average values of the load factor as:

$$k_c = \begin{cases} 1 & \text{bending} \\ 0.85 & \text{axial} \\ 0.59 & \text{torsion} \end{cases} \quad \text{................(6.12)}$$

Temperature Factor k_d

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

$$k_d = \frac{S_T}{S_{RT}} \quad \text{...............(6.13)}$$

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 (6.2) Effect of Operating Temperature on the Tensile Strength of Steel.

Where :

S_T = tensile strength at operating temperature;

S_{RT} = tensile strength at room temperature

Reliability Factor k_e

the reliability modification factor table(6.3) to account for this can be written as:

$$k_e = 1 - 0.08\, z_a \quad \text{.................}(6.14)$$

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(6.3) reliability modification factor

Miscellaneous-Effects Factor k_f

Though 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

Ssy = 0.577Sy ....(6.15)

And :

$$n_y = \frac{S_{sy}}{\tau_a + \tau_m} \quad \text{..................(6.16)}$$

Where :

n_y : Yield factor of safety and the value must be more than 3

S_{sy} : the shear yield strength (kpsi)

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

6.4.3 Data analysis

$$T_{max} = p_{max}/\omega = \frac{800\ watt}{\frac{20000(\frac{rev}{min})*3.14*2}{60\ min}} = 0.382\ N.m$$

$$Ta = \frac{Tmax-0}{2} = 0.5\ T_{max} = 0.5*0.382 = 0.191\ N.m$$

$$T_m = \frac{Tmax+0}{2} = 0.5\ T_{max} = 0.5*0.382 = 0.191\ N.m$$

Assume the diameter of the shaft (4 mm) then :

$$\tau_a = \frac{16\ Ta}{\pi d^3} = \frac{16*0.191}{3.14\ (4*10^{-3})^3} = 15.2*10^6\ N/m^2 = 2.205\ kpsi$$

$$\tau_m = \frac{16\ Tm}{\pi\ d^3} = \frac{16*0.191}{3.14\ (4*10^{-3})^3} = 15.2*10^6\ N/m^2 = 2.205\ kpsi$$

Table (6.4) Typical applications for plain steels (based on the SAE-AISI numbers)

	Number	Properties	Applications
Low Carbon	1006-12	soft and plastic	Sheets, stripping, tubes, welding
	1015-22	soft and tough	rivets, screws, wire, structural shapes
	1023-32	medium	pipes, gears, shafts, bars, structural shapes
Medium Carbon	1035-40		large section parts: forged parts, shafts, axles, rods, gears
	1041-50		heat treated parts: shafts, axles, gears, spring wire
	1052-55		heavy duty machine parts: gears, forgings
High Carbon	1060-70	shock resistant	
	1074-80	tough and hard	dies, rails, set screws
	1084-95		shear blades, hammers, wrenches, chisels, cable wire
			cutting tools: dies, milling cutters, drills, taps, etc.

From table (6.4) , the type of steel is 1050 HR, and from appendix , the value of S_{ut} is 90 kpsi

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

From table , $S'e = 0.5*S_{ut} = 0.5*90 = 45$ kpsi

$$and: \quad S_e = k_a k_b k_c k_d k_e k_f\ S'e$$

where:

$k_a = aS_{ut}^{b} = 14.4*(90)^{-0.107} = 0.57$, a & b from table (6.1)

$k_b = 1.24d^{-0.107} = 1.24*(5)^{-0.107} = 1.04$

$k_c = 0.59$, from equation (6.12)

$k_d = 0.768$, from table (6.2)

$k_e = 1$, when reliability (50%) table (6.3)

$k_f = 1$

then endurance limit : -

$$S_e = 0.57 * 1.04 * 0.59 * 0.768 * 1 * 1 * 45 = 12.08 \text{ kpsi}$$

Fatigue factor of safety : -

$$\frac{\tau_a}{S_e} + \frac{\tau_m}{S_{su}} = \frac{1}{n_f} \rightarrow \frac{2.205}{12.08} + \frac{2.205}{60.3} = \frac{1}{n_f} \rightarrow n_f = 4.56 > 3$$

Yield factor of safety: -

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

And : $$n_y = \frac{S_{sy}}{\tau_a + \tau_m} = \frac{28.56}{1.88 + 1.88} = 6.4 > 3$$

So that the design is good because no fatigue and no yield.

6.5 Journal Bearings and their lubrication

6.5.1 Journal Bearings

A journal or sleeve bearing figure (6.2) and figure (6.4) consists of a cylindrical housing supporting a rotating shaft. The term "journal" refers to the portion of a shaft contained within a bearing; "sleeve" refers to the bearing configuration. If the bearing is a full-cylindrical, 360-degree design, it is called a bushing. A shaft that is loaded in a single direction can be supported by a journal bearing in the form of a partial cylinder. Such a bearing supports the shaft in the load zone only. For example, cranes, earth-moving equipment and railroad journals use partial-cylinder bearings to support loads directed against the top portion of an axle.

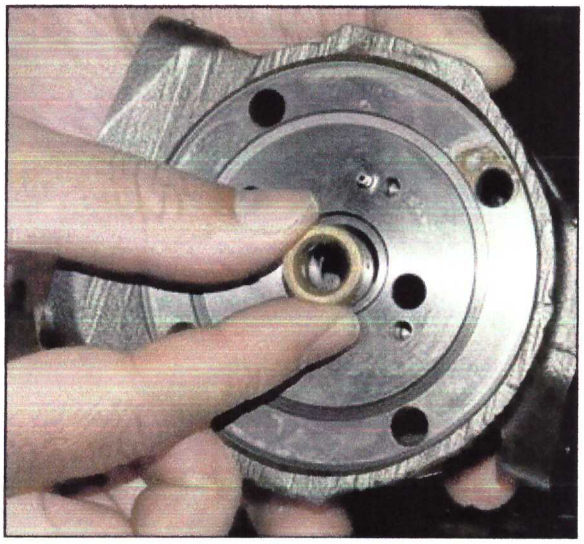


Figure (6.2) journal bearing

Journal bearings frequently contain two or more parts to facilitate removal or replacement. Automotive engine main bearings, for instance, contain two half-round sleeves which hold the crankshaft journals.

6.5.2 Thrust Bearings

Thrust bearings figure (6.3) and figure (6.4) accommodate the axial movement of a rotating shaft. They are usually used in conjunction with journal bearings and are lubricated by grease, which leaks from the ends of the journal housings.

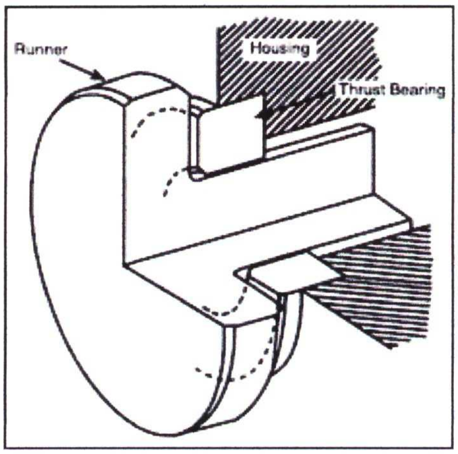


figure (6.3) Thrust bearings

1- Journal bearing
2- Thrust bearing

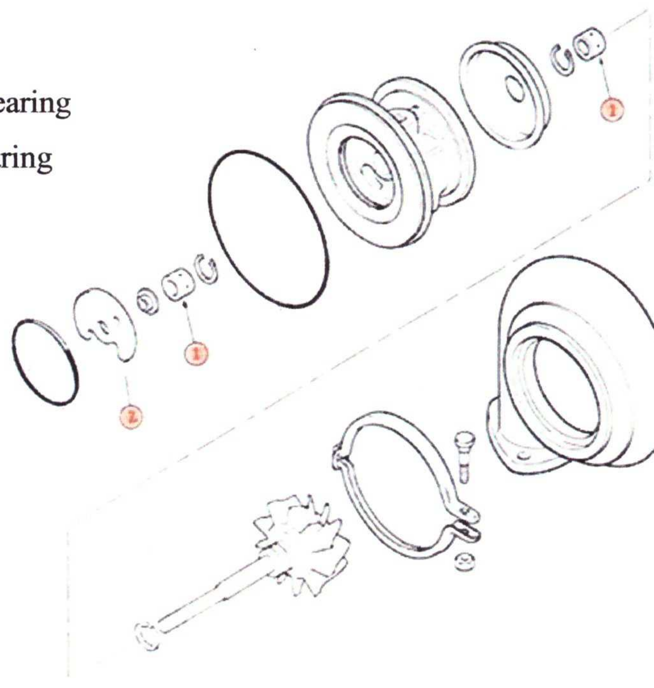


Figure (6.4) journal and Thrust bearing in the turbo

6.5.3 Lubrication of Plain Bearings

The mode of lubrication of a plain bearing depends on the conditions that affect the bearing's ability to develop a load-carrying fluid film to separate the journal and bearing surfaces. If such a film is not produced (or before it is produced), the lubrication mode is termed boundary or mixed-film - lubrication where the surfaces are not completely separated and some metal-to-metal contact occurs. If a lubricating film is formed with sufficient pressure to separate the journal and bearing surfaces, the lubrication mode is termed hydrodynamic or full fluid-film lubrication.

1. **Boundary Lubrication**

When the shaft is at rest or at low speeds (typically at start up) or under high loads, asperities on the surface of the bearing and journal are in contact. Lubrication under these conditions as shown in figure (6.5) depends on the nature of the contacting surfaces, lubricant decomposition products that may be present, or surface-active additives which form a thin, soft film on the metal surfaces and prevent metal-junction adhesion to reduce friction.

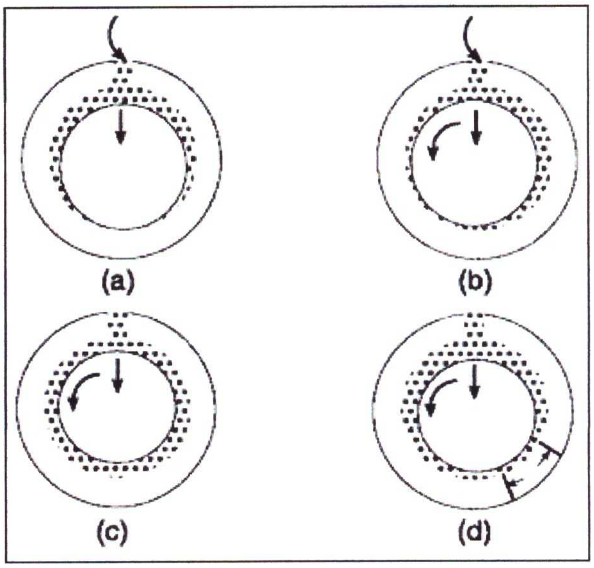


Figure (6.5) Lubrication conditions

Because of the generation of relatively high levels of friction and heat, and the resulting high rate of surface wear, boundary lubrication is not the most desirable mode of operation. However, at times, it is completely unavoidable. As the journal starts to rotate, it climbs the bearing surface in a direction opposite to rotation. A layer of grease (oil in an engine) clings to the journal and rotates with it. This layer is carried into the converging space between the journal and the bearing and begins to form a thin fluid film. The journal rotates with the film until sufficient fluid has been carried into the converging space to separate the surfaces further. A grease layer clings to the journal and rotates with it; another layer adheres to the bearing surface and remains stationary. Layers of grease within the film slide between the outside layers; those closest to the journal move most while those layers closest to the bearing move least.

2. **Hydrodynamic Lubrication**

As speed increases, the wedging action of the lubricant moves in the direction of rotation, and pressure within the film becomes greater so the journal is now riding on a full fluid-film and hydrodynamic lubrication is reached. If loading on the bearings is increased sufficiently, the hydrodynamic film may collapse and the bearing will revert to the boundary-lubrication mode.

Grease should be introduced to the bearing where fluid pressure is least - at the point of maximum clearance within the bearing. Grooves are often incorporated in the interior surface of a journal bearing to relieve pressure and to store reserve lubricant. When loading is in one direction, axial grooves running lengthwise on the bearing surface and located in areas of low pressure will not disturb the lubricating film and can relieve pressure. When the direction of loading is variable, the location of pressure extremes within the bearing is also variable. Under these conditions, well-spaced annular or circumferential grooves will relieve pressure without substantially interrupting lubricating films. Axial grooves should be bevelled so lubricating grease is more easily swept from the groove by the rotating shaft.

6.5.3.1 Choice of lubricant

The choice between oil or grease lubrication depends on the relationship of journal speed to viscosity. Slower journal speeds have higher viscosity requirements while high speeds call for a light-bodied oil. Bearings designed for low speed operation usually have a relatively large clearance between shaft and housing, while high-speed bearings usually have a much smaller clearance.

CHAPTER SEVEN

ENGINE PREFORMANCE

absorption or passive dynamometer. A dynamometer that can either drive or absorb is called a universal or active dynamometer.

7.3.1 Principles of operation

An absorbing dynamometer acts as a load that is driven by the prime mover that is under test. The dynamometer must be able to operate at any speed, and load the prime mover to any level of torque that the test requires. A dynamometer is usually equipped with some means of measuring the operating torque and speed.

7.3.2 Detailed dynamometer description

A dynamometer figure (7.1) consists of an absorption (or absorber/driver) unit, and usually includes a means for measuring torque and rotational speed. An absorption unit consists of some type of rotor in a housing. The rotor is coupled to the engine or other equipment under test and is free to rotate at whatever speed is required for the test. Some means is provided to develop a braking torque between dynamometer's rotor and housing. The means for developing torque can be frictional, hydraulic, electromagnetic etc. according to the type of absorption/driver unit.

One means for measuring torque is to mount the dynamometer housing so that it is free to turn except that it is restrained by a torque arm. The housing can be made free to rotate by using trunnions connected to each end of the housing to support the dynamometer in pedestal mounted trunnion bearings. The torque arm is connected to the dynamometer housing and a weighing scales is positioned so that it measures the force exerted by the dynamometer housing in attempting to rotate. The torque is the force indicated by the scales multiplied by the length of the torque arm measured from the center of the dynamometer.

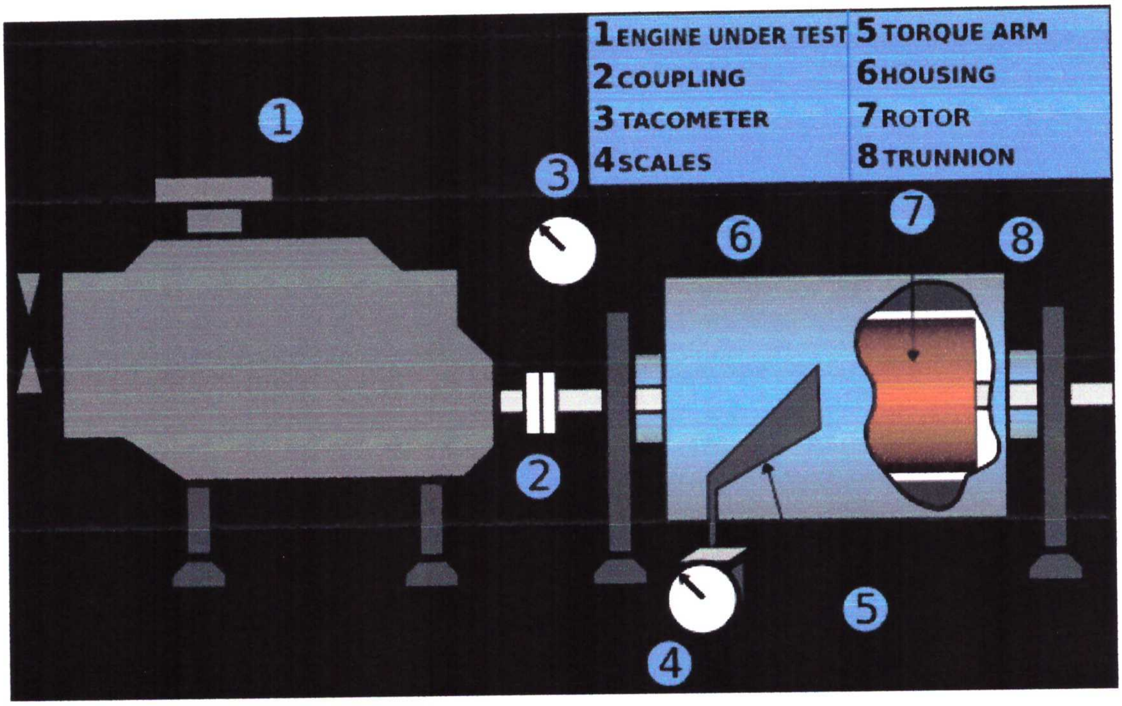


Figure (7.1) dynamometer

Another means for measuring torque is to connect the engine to the dynamometer through a torque sensing coupling or torque transducer. A torque transducer provides an electrical signal that is proportional to torque.

With electrical absorption units, it is possible to determine torque by measuring the current drawn (or generated) by the absorber/driver. This is generally a less accurate method and not much practiced in modern time, but it may be adequate for some purposes.

A dynamometer that can measure torque and power delivered by the power train of a vehicle directly from the drive wheel or wheels (without removing the engine from the frame of the vehicle), is known as a chassis dynamometer.

7.3.3 Dynamometer equations

In most dynamometers power (P) is not measured directly; it must be calculated from torque (T) and angular velocity (ω) values or force (F) and radius of moment arm of the dynamometer (R):

T= F . R......................(7.1)

Where:

T : torque (N.m)

R : radius of moment arm of the dynamometer (m)

F : restraining force required to stop the moment arm rotating (the effective load on the dynamometer) (N)

The power P delivered by the engine turning at a angular speed w and absorbed by the dynamometer can he calculated from

P = ω . T.....................(7.2)

Where:

ω : is the angular speed of dynamometer shaft (rev/ s)

$$\mathbf{P(kw)=\frac{T*w}{9549}} \quad ....(7.3)$$

Note:-

There were difficulties to the install the dynamometer on the engine which required calibration, so the effect turbine using was neglected .

CHAPTER EIGHT

TEST BENCH SPECIFICATION

TEST BENCH SPECIFICATION

8.1 Introduction

This chapter contains the project part properties, fitting steps, and focus on the technical specification.

8.2 Project parts properties

8.2.1. Internal combustion engine

The practical part of the project will be done using (VW, 2.40 liter, 82 model) engine available in the mechanical engineering department workshop, four strokes, and six cylinders as shown in figure (8.1).

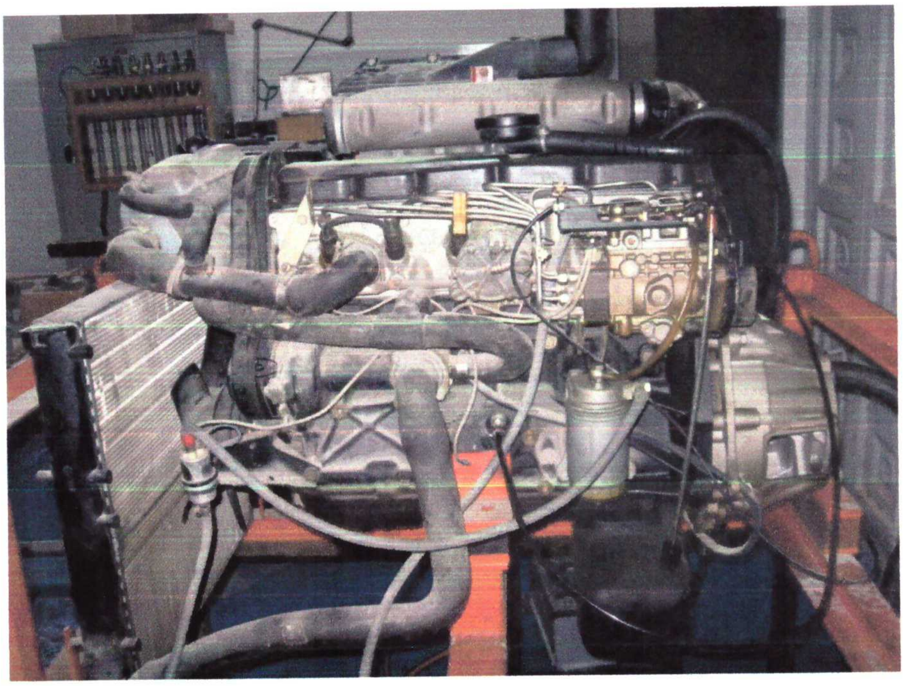


Figure (8.1) VW engine

8.2.2 Air conditioning compressor

Displacement: 10-161 cc /r

Number of cylinder: 5

Diameter of cylinder: 36.1

Stroke: 30.6 mm

Max. speed: 7000 rev/ min

Refrigerant: R134a

Oil: PAG

Voltage: 12V

Number of groove: A-131mm

OEM: 1854056/1854057

figure (8.2)Air conditioning compressor

8.2.3 Test equipment

8.2.3.1 pressure gauge

It's a device for measuring the pressure of a exhaust gases before and after the turbine .

Figure (8.3)

8.2.3.2 Thermocouple

A thermocouple figure (8.4) is a junction between two different metals that produces a voltage related to a temperature difference. Thermocouples are a widely used type of temperature sensor. They are cheap and interchangeable, have standard connectors, and can measure a wide range of temperatures. The main limitation is accuracy System errors of less than one Kelvin (K) can be difficult to achieve.

Figure(8.4) Thermocouple

Its used to for measuring the temperature of a exhaust gases before and after the turbine.

Description of thermocouple: Thermocouples provide an economic means of measuring temperature with many practical advantages for the user. Type K is nickel doped with aluminum and nickel doped with chromium.

Type	:	K
Temperature Range	:	-200 to 1100C^0
Units	:	°C, °F, K, Rankine.

8.2.3.3 Temperature controller

The temperature controller figure (8.5) takes an input from a temperature sensor and has an output that is connected to a control element .

How do Temperature Controllers work?

To accurately control process temperature without extensive operator involvement, a temperature control system relies upon a controller, which accepts a temperature sensor such as a thermocouple or RTD as input. It compares the actual temperature to the desired control temperature, or set point, and provides an output to a control element. The controller is one part of the entire control system, and the whole system should be analyzed in selecting the proper controller.

Temperature controller figure(8.5)

8.2.3.4 DEUMO

for measuring the speed of turbine in RPM.

Description of Deumo

Three measuring range on each model.
Direct RPM reading.
Very wide speed range.
Clear reading-very long scale dial .
self powered – no batteries required.
Speed range
40 – 500 RPM
400 – 5000 RPM
4000 – 50000 RP

Figure (8.6) DEUMO

8.3 Project parts fitting

8.3.1 Pressure gauge fitting

The pressure gauge will be fitted before and after the turbine as shown in figure (8.7) to measure the pressure of exhaust gases before and after the turbine.

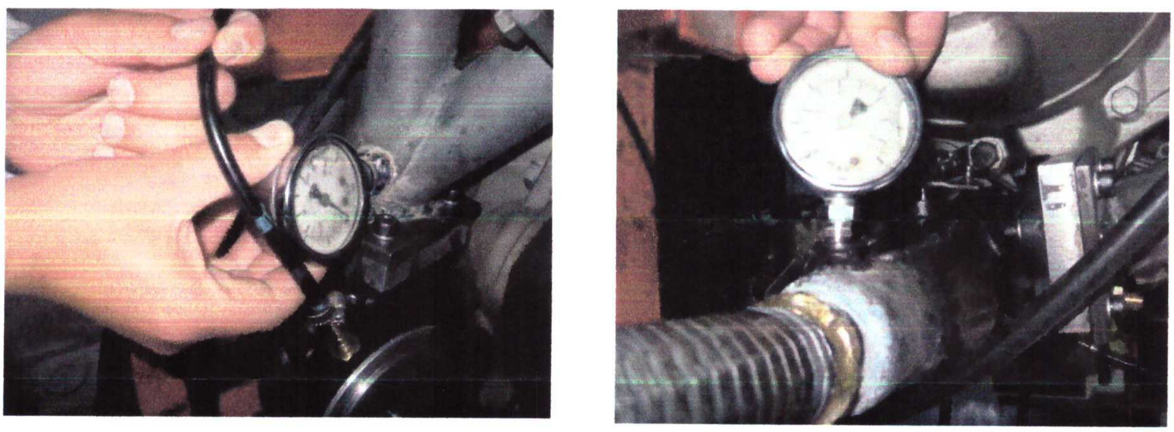


Figure (8.7) Pressure gauge fitting

CHAPTER NINE

POWER OF THE TURBINE

POWER OF THE TURBINE

9.1 Power input to the turbine

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

$$\textbf{Power}_{\text{in}} = \frac{\mathbf{Q} * \mathbf{P}_{in}}{\mathbf{17.1}} *[(\frac{\mathbf{P_{in}}}{\mathbf{P_{out}}})^{\alpha} \ - \ \mathbf{1}] \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots..(9.1)$$

Then : $\alpha = \frac{k-1}{K}$

Where:-

Power $_{in}$: Power input (KW).
Q : Volume flow rate (m^3/Sec).
P : Exhaust gas pressure (Kpa).
k : Ratio of specific heats, C_P / C_v

$$\textbf{Volume flow rate (Q)} = \frac{rpm}{60} * \frac{Vcc}{2\ (\,for\ two\ revolution\,)} \quad \ldots\ldots\ldots..(9.2)$$

Where:

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

Using the pressure gauge as shown in figure (9.1), the pressure input to turbine is measured and its change with engine speed as shown in table (9.1)

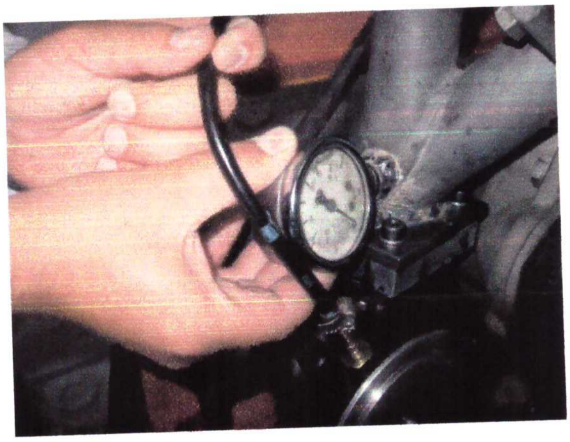


Figure (9.1) Pressure gauge

Engine speed (RPM)	Pressure (Kpa)
1500	130
2000	160
3000	190
4000	200
5000	250
6000	300

Table (9.1) relation between pressure and engine speed

Note :

1- The volume of the engine is 2400 cc = $2400*10^{-6}$ m^3

2- K for air =1.4

Then :

$$\textbf{Volume flow rate (Q)} == \frac{1500}{60} * \frac{2400*10^{-6}}{2\ (\textit{for two revolution})} = 0.03\ m^3/s$$

$$\alpha = \frac{1.4-1}{1.4} = 0.286$$

$$\text{Power}_{\text{in}} = \frac{0.03*1.3*10^2}{17.1} * \left[\left(\frac{1.3*10^2}{1*10^2}\right)^{0.286} - 1\right]$$

$$= 0.0178\ \text{KW}$$

Table(9.2) and figure(9.2) show the relation between power and exhaust pressure

Engine speed (RPM)	Exhaust pressure (Kpa)	Volume flow rate(m^3/s)	Input power (KW)
1500	130	0.03	0.0178
2000	160	0.04	0.0538
3000	190	0.06	0.1343
4000	200	0.08	0.205
5000	250	0.1	0.438
6000	300	0.12	0.777

Table(9.2) relation between power and exhaust pressure

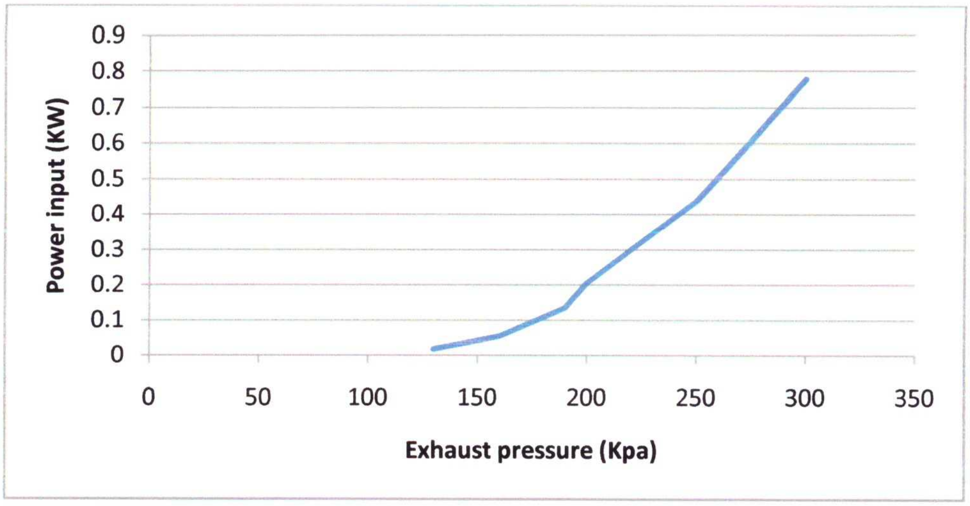


Figure (9.2) relation between power and exhaust pressure

9.2 Power output from turbine

There are three ways to find the outlet power of the turbine.

9.2.1By dynamometer

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

$$\mathbf{P = T * \omega} \quad \text{......................}(9.3)$$

where :

$$\omega = \frac{rpm * 2\pi}{60} \quad \text{..................}(9.4)$$

where :

T : Torque(N.m)

ω : Number of revolution(rev/minute).

But using dynamometer was not applicable because there is no suitable dynamometer to do that.

9.2.2 Electrical Alternator.

The second way to find the turbine output power (by using alternator connected with the turbine shaft as show in figure(9.3)) by using a volt ampere tester as show in figure(9.4)

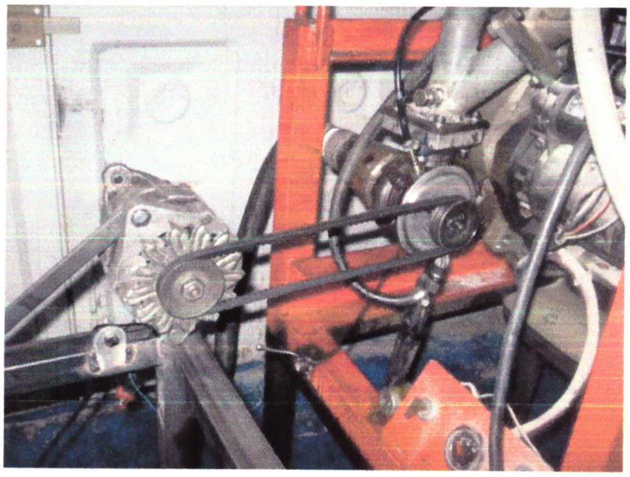


Figure (9.3) alternator connected with turbine

Figure (9.4) a volt ampere tester

This device is used to measure the maximum voltage and the maximum current.

Then :

the max. electric output power = (max. voltage) * (max. current).

P = V * I ....(9.5)

Where

P: The output power (KW)

V: Voltage (volt)

I: Current(ampere)

The max. mechanical power of turbine = electric power / turbine efficiency

$$\mathbf{P} = (\boldsymbol{V} * \boldsymbol{I})/\boldsymbol{\eta_t} \quad \text{............... (9.6)}$$

Where :

η_t :is the turbine efficiency.

This way to find the turbine output power by alternator is not completed because a problem occurred in the turbine in the final work.

9.2.3 Turbine Efficiency

The third way to determine the turbine output power is using the turbine efficiency equations.

$$\eta_t = \frac{\textbf{\textit{Power output}}}{\textbf{\textit{Power input}}} \quad \text{............(9.7)}$$

The input power can be calculated from the previous equation, and the turbine efficiency can be calculated from this equation, since the inlet and outlet temperature can be measured by thermocouple and the inlet and outlet pressure can be measured by pressure gauge at the inlet and outlet of turbine.

$$\eta_t = \frac{1-(T_{EXT}/TIT)}{1-(R_t)^{\sigma}} \text{..................(9.8)}$$

But :

$$\sigma = \frac{k-1}{k} \text{.....................(9.10)}$$

Where:-

T_{EXH} = Turbine total exhaust temperature, K

TIT = Turbine total inlet temperature, K

R_t = Atmospheric pressure / Exhaust pressure

k = Ratio of specific heats, C_P / C_v

C_P = Specific heat at constant pressure, kilojoules/kilogram. K

C_v= Specific heat at constant volume, kilojoules/kilogram. K

But this way to find power outlet of turbine is not completed because a problem occurred in the turbine in the final work of the project. So that to solve this problem the efficiency of turbine is selected, which is selected to be about 80%, Because the efficiency of turbine is between (70% - 80%) .

Then : the power output is

$$\eta_t = \frac{\textit{Power output}}{\textit{Power input}} \rightarrow \mathbf{0.8} = \frac{\textit{Power output}}{0.0178}$$

Power output = 0.0142 KW

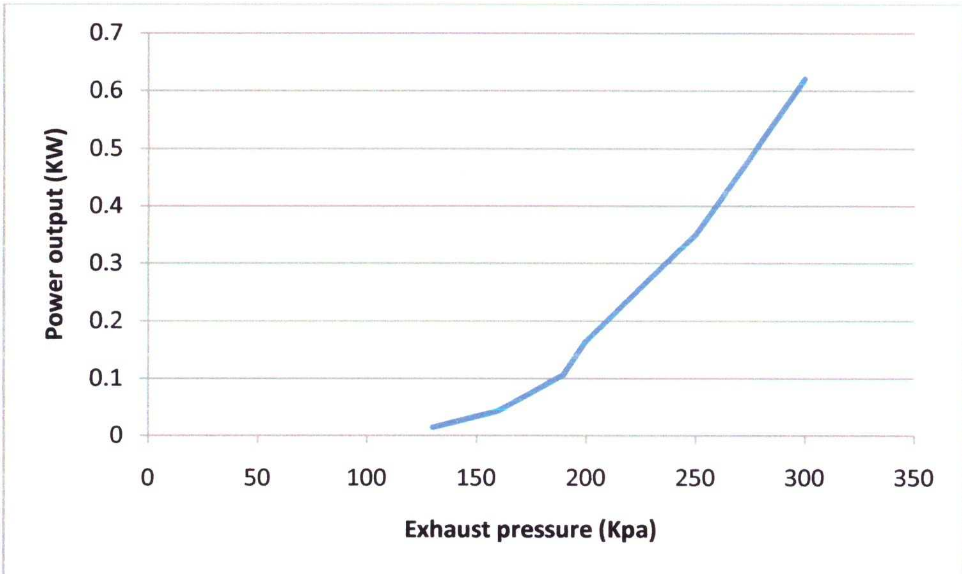
Table(9.3) and figure(9.5) show the relation between power, exhaust pressure and engine speed.

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.

Engine speed (RPM)	Exhaust pressure (Kpa)	Volume flow rate(m^3/s)	Output power (KW)
1500	130	0.03	0.0142
2000	160	0.04	0.043
3000	190	0.06	0.107
4000	200	0.08	0.164
5000	250	0.1	0.350
6000	300	0.12	0.621

Table(9.3)Relation between power output and exhaust pressure



Figure(9.5)relation between power and exhaust pressure

CHAPTER TEN

CONCLUSION RECOMMENDATIONS AND PROBLEMS

CONCLUSION RECOMMENDATIONS AND PROBLEMS

10.1 Conclusions

1. The max. power produced from turbine is about 650 W, and this is not enough to drive air conditioning compressor.
2. The power produced can be used for different way in the vehicle.
3. The engine performance must be calculated before and after turbine installation to know the effect of turbine on the power of the engine.

10.2 Recommendations

1. Utilization the power produced from exhaust gases to operate some devices in the vehicle.
2. The possibility of using a compressor of air conditioning with low requirement of power to operate a small refrigerant in vehicle.

10.3 Problems

1. The first problem that interfaces this project is the 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.
2. The second problem was a problem occurred in the turbine because:
   A. The efficiency of turbine is not so high when it's installed.
   B. The work on the turbine and the installation of a pulley on it caused problem in shaft balance.

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Appendix

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 Processing	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 A-21

Mean Mechanical Properties of Some Heat-Treated Steels

[These are typical properties for materials normalized and annealed. The properties for quenched and tempered (Q&T) steels are from a single heat. Because of the many variables, the properties listed are global averages. In all cases, data were obtained from specimens of diameter 0.505 in, machined from 1-in rounds, and of gauge length 2 in. unless noted, all specimens were oil-quenched.] Source: ASM Metals Reference Book, 2d ed., American Society for Metals, Metals Park, Ohio, 1983.

1 AISI No.	2 Treatment	3 Temperature °C (°F)	4 Tensile Strength MPa (kpsi)	5 Yield Strength, MPa (kpsi)	6 Elongation, %	7 Reduction in Area, %	8 Brinell Hardness
1030	Q&T*	205 (400)	848 (123)	648 (94)	17	47	495
	Q&T*	315 (600)	800 (116)	621 (90)	19	53	401
	Q&T*	425 (800)	731 (106)	579 (84)	23	60	302
	Q&T*	540 (1000)	669 (97)	517 (75)	28	65	255
	Q&T*	650 (1200)	586 (85)	441 (64)	32	70	207
	Normalized	925 (1700)	521 (75)	345 (50)	32	61	149
	Annealed	870 (1600)	430 (62)	317 (46)	35	64	137
1040	Q&T	205 (400)	779 (113)	593 (86)	19	48	262
	Q&T	425 (800)	758 (110)	552 (80)	21	54	241
	Q&T	650 (1200)	634 (92)	434 (63)	29	65	192
	Normalized	900 (1650)	590 (86)	374 (54)	28	55	170
	Annealed	790 (1450)	519 (75)	353 (51)	30	57	149
1050	Q&T*	205 (400)	1120 (163)	807 (117)	9	27	514
	Q&T*	425 (800)	1090 (158)	793 (115)	13	36	444
	Q&T*	650 (1200)	717 (104)	538 (78)	28	65	235
	Normalized	900 (1650)	748 (108)	427 (62)	20	39	217
	Annealed	790 (1450)	636 (92)	365 (53)	24	40	187
1060	Q&T	425 (800)	1080 (156)	765 (111)	14	41	311
	Q&T	540 (1000)	965 (140)	669 (97)	17	45	277
	Q&T	650 (1200)	800 (116)	524 (76)	23	54	229
	Normalized	900 (1650)	776 (112)	421 (61)	18	37	229
	Annealed	790 (1450)	626 (91)	372 (54)	22	38	179
1095	Q&T	315 (600)	1260 (183)	813 (118)	10	30	375
	Q&T	425 (800)	1210 (176)	772 (112)	12	32	363
	Q&T	540 (1000)	1090 (158)	676 (98)	15	37	321
	Q&T	650 (1200)	896 (130)	552 (80)	21	47	269
	Normalized	900 (1650)	1010 (147)	500 (72)	9	13	293
	Annealed	790 (1450)	658 (95)	380 (55)	13	21	192
1141	Q&T	315 (600)	1460 (212)	1280 (186)	9	32	415
	Q&T	540 (1000)	896 (130)	765 (111)	18	57	262

(continued)

Table A-21 *(Continued)*

Mean Mechanical Properties of Some Heat-Treated Steels

[These are typical properties for materials normalized and annealed. The properties for quenched and tempered (Q&T) steels are from a single heat. Because of the many variables, the properties listed are global averages. In all cases, data were obtained from specimens of diameter 0.505 in, machined from 1-in rounds, and of gauge length 2 in. Unless noted, all specimens were oil-quenched.] *Source: ASM Metals Reference Book, 2d ed., American Society for Metals, Metals Park, Ohio, 1983.*

1 AISI No.	2 Treatment	3 Temperature °C (°F)	4 Tensile Strength MPa (kpsi)	5 Yield Strength, MPa (kpsi)	6 Elongation, %	7 Reduction in Area, %	8 Brinell Hardness
4130	Q&T*	205 (400)	1630 (236)	1460 (212)	10	41	467
	Q&T*	315 (600)	1500 (217)	1380 (200)	11	43	435
	Q&T*	425 (800)	1280 (186)	1190 (173)	13	49	380
	Q&T*	540 (1000)	1030 (150)	910 (132)	17	57	315
	Q&T*	650 (1200)	814 (118)	703 (102)	22	64	245
	Normalized	870 (1600)	670 (97)	436 (63)	25	59	197
	Annealed	865 (1585)	560 (81)	361 (52)	28	56	156
4140	Q&T	205 (400)	1770 (257)	1640 (238)	8	38	510
	Q&T	315 (600)	1550 (225)	1430 (208)	9	43	445
	Q&T	425 (800)	1250 (181)	1140 (165)	13	49	370
	Q&T	540 (1000)	951 (138)	834 (121)	18	58	285
	Q&T	650 (1200)	758 (110)	655 (95)	22	63	230
	Normalized	870 (1600)	1020 (148)	655 (95)	18	47	302
	Annealed	815 (1500)	655 (95)	417 (61)	26	57	197
4340	Q&T	315 (600)	1720 (250)	1590 (230)	10	40	486
	Q&T	425 (800)	1470 (213)	1360 (198)	10	44	430
	Q&T	540 (1000)	1170 (170)	1080 (156)	13	51	360
	Q&T	650 (1200)	965 (140)	855 (124)	19	60	280

*Water-quenched