ADDIS ABABA UNIVERSITY

ADDIS ABABA INSTITUTE OF TECHNOLOGY

SCHOOL OF MECHANICAL AND INDUSTRIAL ENGINEERING

Investigation of compressed Air Engine starting system for Light Diesel-Electrical Passenger locomotive

By

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Abstract

This Thesis Investigates Compressed Air Engine starting system for light Diesel Electrical Passenger Locomotive. Most Internal combustion engines are usually started by means of Electrical starting arrangements. The heavy diesel engines are starting with compressed air, this compressed Air Engine starting system coupled to the engine flywheel, or to inject the compressed air directly into the engine cylinders through a slashing valve, but our Thesis consider on the compressed air Engine starting system directly inject into Engine cylinder. compressed Air Engine starting systems are an additional starting system mounted on locomotive when an Electrical starting system is failed compressed air starting system comes into operation.

The main advantage related to compressed air engine starting for light Diesel Electrical Locomotive fueled engines, is to avoid traffic delay when battery system or electrical starting system not working properly.

Light Diesel-Electrical passenger Locomotive Engines during compressed air starting the capacity or the amount of compressed air regulated by pressure control valve and pressure reduced valve. In this Thesis, analytical method, modeling of the system and simulation method by software of compressed air Engine starting system for light Diesel Electrical Locomotive (third stage air compressor, air filter, air cylinder, pressure control valve, pressure reducing valve, electro pneumatic valve, Air distributer and air starting valve) are developed and validated against other application area of the system.

The objective of our research for Investigating compressed air Engine starting system for light Diesel Electrical Locomotive, by method of numerical Analysis, modeling(CATIA and AutoCAD) and computer simulations(ANSYS workbench), the system existing on other area but can be possible to adopt Railway Industry.

Keywords: ANSYS workbench; light Diesel Electrical Locomotive; Engine starting; CATIA and AutoCAD; air distributor.

CHAPTER ONE: INTRODUCTION

1.1 Background

All combustion engines rely on some sort of external power system to crank the engine and produce sufficient cylinder pressure for ignition of the fuel. In small automotive engines, electric starter motors are normally used. For larger engines, like the ones used in Locomotive, the power demand is so vast that an electrical starting system powerful enough to accelerate the system would be unpractical large. Instead alternative starting systems with better power-to-size ratio is used.

1.2 Starting Systems

Engine starting system converts electrical, hydraulic or compressed Air energy from the main source into mechanical energy to start the engine[1]. Types of Engine starting system:-

- Electric Engine starting system
- Hydraulic Engine starting system
- Air motor Engine starting system and
- Compressed air admission Engine starting system.

1.2.1 Electric Engine Starting System

The purpose of the starter motor and related circuitry is to convert the electrical energy stored in the battery (put there by the alternator) into mechanical energy in order to rotate the engine to allow it to be started. An electric motor is virtually a mirror image of an alternator i.e. a conductor is suspended in a magnetic field, current is passed through this conductor and the interaction between the subsequent magnetic fields creates movement (as opposed to the mechanical movement of a conductor in the magnetic field generating current flow in the conductor)[2].

The main components of the electric starting system are a storage battery, starting motor, and associated control and protective devices.

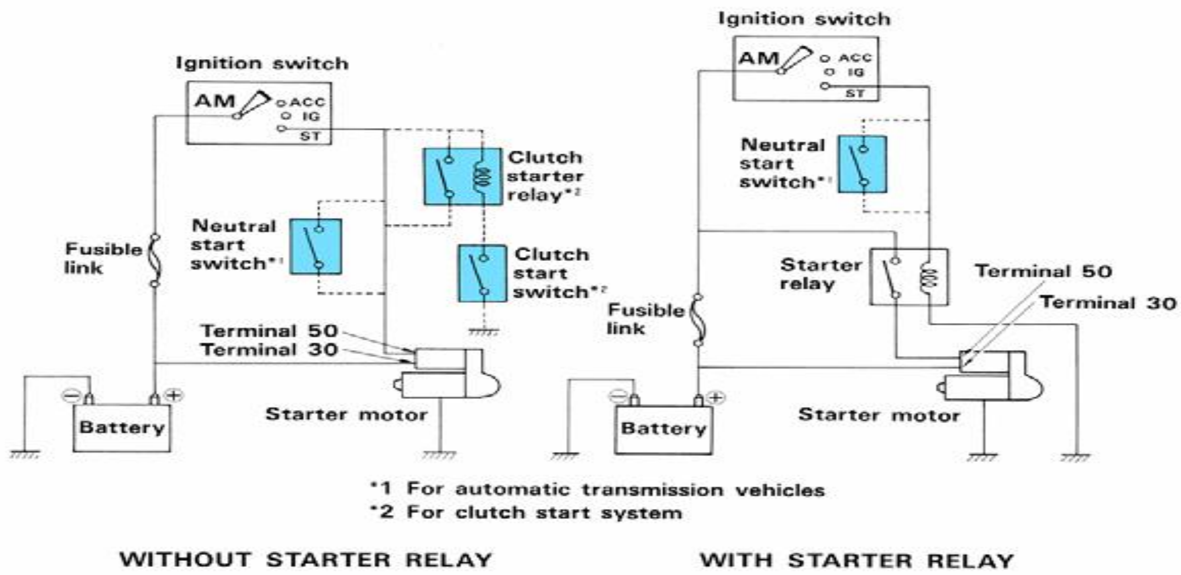


Figure.1.1.Electrical cranking system and gear reduction[1]

Batteries

The lead acid storage battery provides the power source for starting most engines. To supply the required current for starting, the individual capacity is 6-volt, batteries are connected in series because the batteries connected series the voltage output is sum up. For maximum efficiency and long life of the storage battery, periodic inspections are essential. Batteries that serve to start engines are subjected to moderately heavy use and may require frequent charging in addition to the charging provided by the engine generator or alternator[2].

Monitoring The Charging System

An ammeter is a device that is wired into the electrical circuit to show the current flow to and from the battery and is mounted on the gauge board (panel) along with the other monitoring gauges. After the engine is started, the ammeter should register a high charge rate at the rated engine speed. This is the rate of charge received by the battery to replenish the current the battery has used to start the engine. As the engine continues to operate, the ammeter should show a decline in the charge rate to the battery.

The ammeter will not show a zero charge rate since the regulator voltage is set higher than the battery voltage. The small current registered prevents rapid brush wear in the battery-charging generator. If lights or other electrical equipment are connected into the circuit, the ammeter should show discharge when these items are operating and the engine speed is reduced[2].

Starting Motors and Drives

The starting motor for a diesel or a gasoline engine operates on the same principle as a direct current electric motor. The motor is designed to turn extremely heavy loads but tends to overheat quickly because it draws a high current (300 to 665 amperes). To avoid overheating, never allow the motor to run for more than the specified amount of time. Then allow it to cool for 2 or 3 minutes before using it again[2].

The starting motor is located near the flywheel, The drive gear on the starter is arranged so that it can mesh with the teeth on the flywheel (or the ring gear) when the starting switch is closed. The drive mechanism has two functions:

- to transmit the turning force to the engine when the starting motor runs and to disconnect the starting motor from the engine immediately after the engine has started and
- to provide a gear reduction ratio between the starting motor and the engine.

The drive mechanism must disengage the pinion from the flywheel immediately after the engine starts. After the engine starts, the engine speed may increase rapidly to approximately 1500 rpm. If the drive pinion were to remain meshed with the flywheel and locked with the shaft of the starting motor, at a normal engine speed (1500 rpm), the shaft would spin at a rapid rate of speed (between 22,500 and 30,000 rpm). At such a rate of speed, the starting motor would be badly damaged[3].

1.2.2 Hydraulic Engine Starting Systems

Hydraulic starting systems are used on various small diesel engines, it has no detrimental effect upon the hydrostarter system. Engine starting torque for a given pressure will remain fairly constant regardless of the ambient temperature. The hydrostarter system consists of a reservoir, an engine-driven charging pump, a manually operated pump, a piston-type accumulator (with a fluid side and a nitrogen side), a hydraulic, vane-type starting motor, and connecting lines and fittings. Hydraulic fluid oil flows by gravity (or by a slight vacuum) from the reservoir to the inlet of either the engine-driven pump or the hand pump[3,4].

Fluid discharged by either pump is forced at high pressure into the accumulator and is stored at approximately 224.1bar under the pressure of compressed nitrogen gas. (Nitrogen is used instead of compressed air because nitrogen will not explode if seal leakage permits oil to enter the nitrogen side of the accumulator).

When the starter is engaged with the ring gear on the fly-wheel of the engine, and the control valve is opened, high pressure fluid is forced out of the accumulator by the action of the piston from the expanding nitrogen gas.

The fluid flows into the starting motor, which rapidly accelerates the engine to a high cranking speed. When the starting lever is released, the spring action disengages the starting pinion and closes the control valve. This action stops the flow of hydraulic oil from the accumulator.

The used fluid returns from the starter directly to the reservoir. During engine operation, the engine-driven charging pump runs continuously and automatically recharges the accumulator.

When the required pressure is attained in the accumulator, a valve within the pump body opens and the fluid discharged by the pump is bypassed to the reservoir. When the system is shut down, the pressure in the accumulator will be maintained. The hand pump of the hydrostarter system is a double-action piston pump. It serves to pump fluid into the accumulator for initial cranking when the accumulator has exhausted all the fluid stored in it. The starter is protected from high speeds of the engine by the action of an over-running clutch.

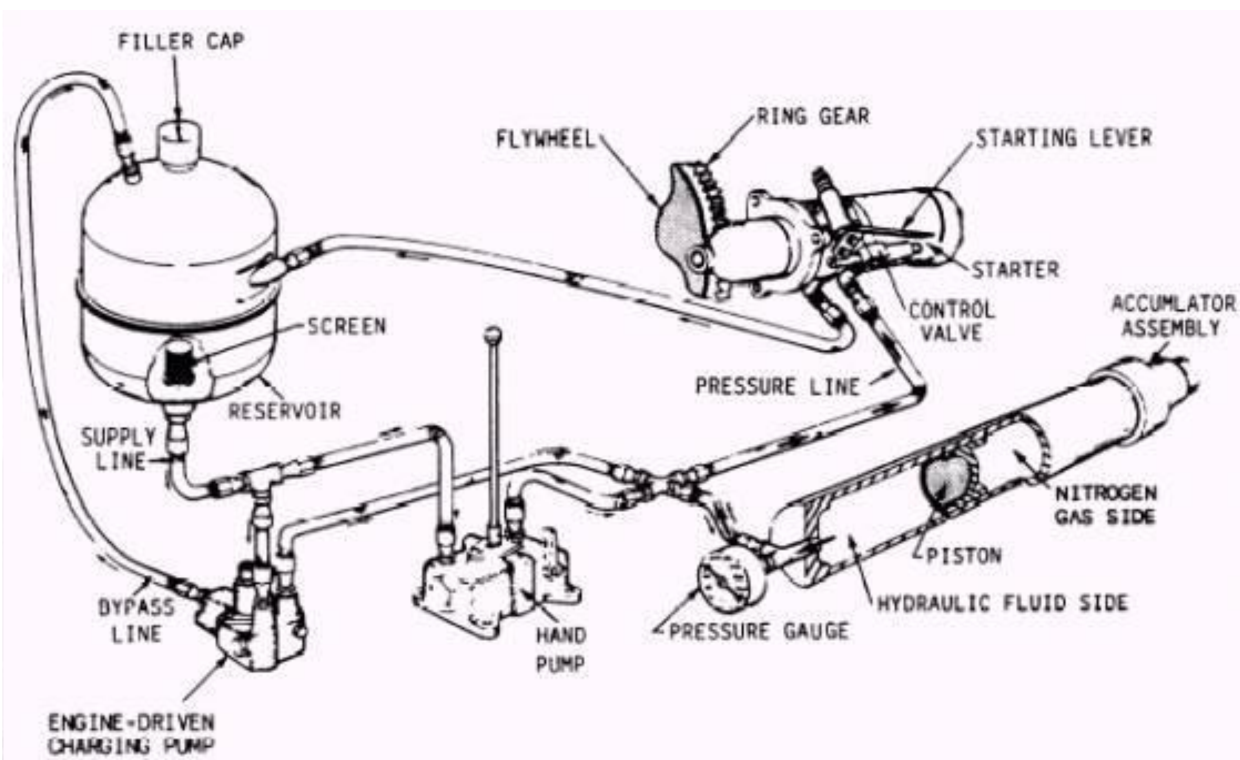


Figure 1.2. Hydraulic starting system[3]

1.2.3 Air Starting Systems

The types of starting systems that derive their power from the pressure of compressed air.

- Air Starting Motor System
- Compressed Air Admission System

1.2.3.1 Sources of Starting Air

Starting air comes directly from the medium-pressure (MP) or high-pressure (HP) air service line or from starting air flasks which are included in some systems for the purpose of storing starting air. From either source, the air, on its way to the starting system, must pass through a pressure-reducing valve, which reduces the higher pressure to the operating pressure required to start a particular engine.

A relief valve is installed in the line between the reducing valve and the starting system. The relief valve is normally set to open at 12% above the required starting air pressure. If the air pressure leaving the reducing valve is too high, the relief valve will protect the system by releasing air in excess of a preset value and permit air only at safe pressure to reach the starting system of the engine[5].

1.2.3.1.1 Air Starting Motor System

Some larger engines and several small engines are cranked over by starting motors that use compressed air. Air starting motors are usually driven by air pressures varying from 6.206bar to 13.79bar. Figure below shows an exploded view of an air starting motor with the major components identified. the principles by which the air starting motor functions.

starting air enters through piping into the top of the air starter housing "1" and flows into the top of the cylinder "2". The bore "3" of the cylinder has a larger diameter than the rotor. The rotor "4" inside the cylinder is a slotted rotating member which is offset with the bore of the cylinder. The rotor carries the vanes "5" in slots, allowing the vanes to maintain contact with the bore of the cylinder. The pressure of the starting air against the vanes forces the rotating member to turn approximately half-way around the core of the cylinder, where exhaust ports "6" allow the air to escape to the atmosphere. A shaft and a reduction gear connect the rotating member to a Bendix drive, which engages the ring gear of the flywheel to crank the engine[5,6].

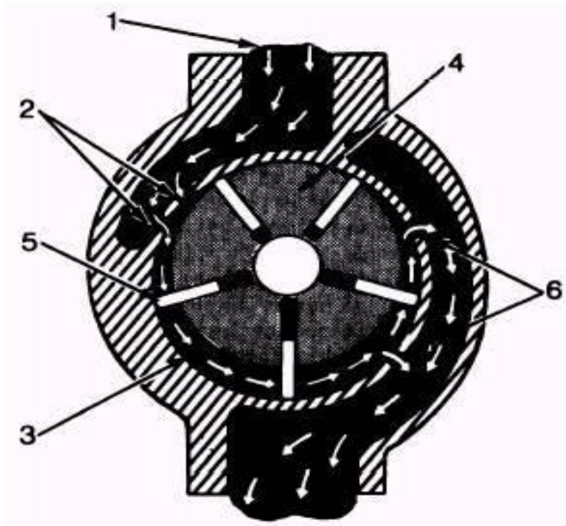


Figure 1.3.Flow of air through an air starting motor[5].

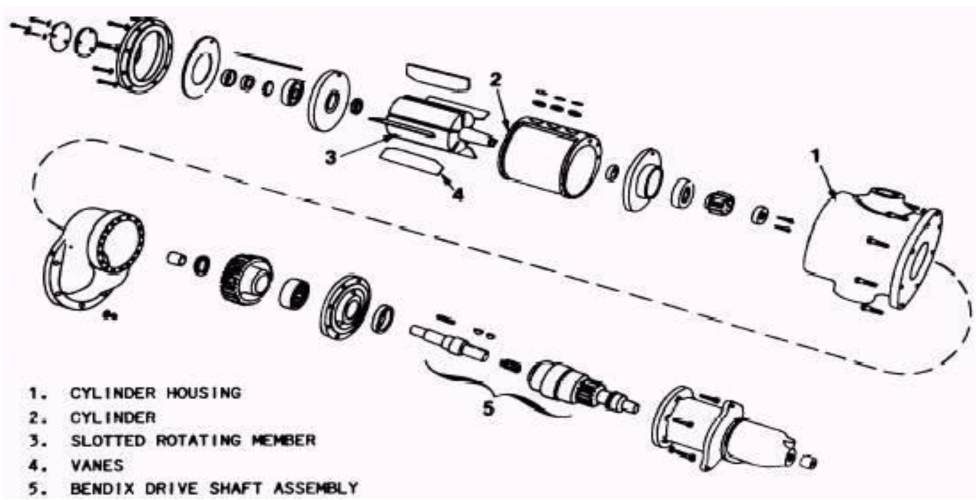


Figure 1.4.-Air starting motor (exploded view)[6].

1.2.3.1.2 Compressed Air Admission System

Most large diesel engines are started when compressed air is admitted directly into the engine cylinders. Compressed air at approximately 50bar to 100bar is directed into the cylinders to force the pistons down and thereby turn the crankshaft of the Engine. This air admission process continues until the pistons are able to build up sufficient heat from compression to cause combustion to start the engine. Figure below The Engine is started by the admitting of compressed air into the right-hand bank of Engine cylinders[6].

Control valves and devices that permit compressed air to flow to the cylinders are actuated by control air that is directed through the engine control system. The heart of the engine control system is the main air start control valve. Except when the engine is being cranked, this component acts as a simple stop valve. A small amount of air is routed through the air supply port (A) to the spring side of the starting air valve.

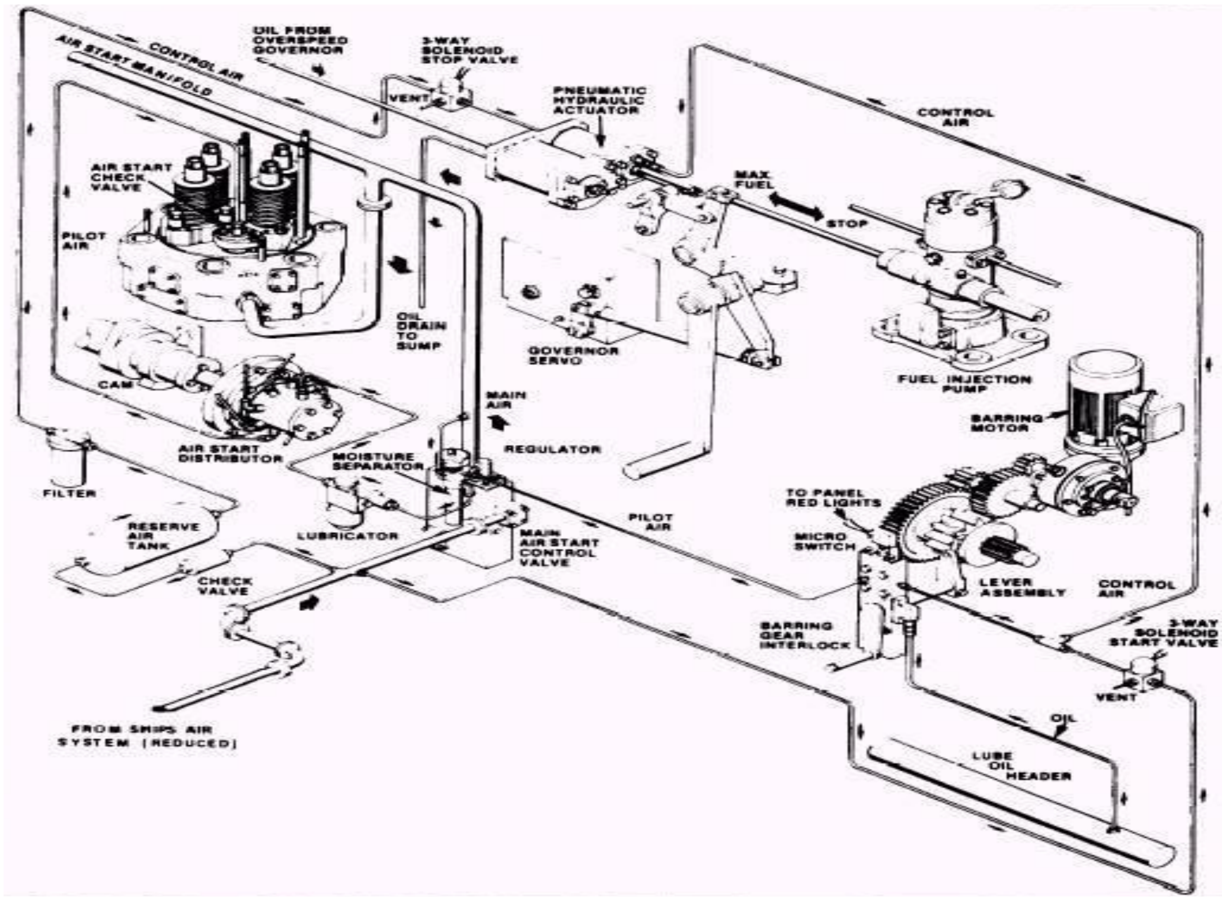


Figure 1.5.Compressed air admission starting system (Colt-Pielstick)[6].

This air pressure, along with the force of the main spring, holds the main starting air valve against its seat. (Air pressure is the same on both sides of the piston inside the valve body; however, the surface area inside the piston (at the bottom) is less than the bottom area outside the piston. So the spring is required to hold the piston down against the seat. When air pressure acting on the piston becomes unbalanced, such as when air pressure on the inside of the piston is reduced, air pressure acting on the larger surface area on the outside of the piston will force the piston up off the seat of the starting air valve.) As long as the main starting air valve remains seated, starting air cannot pass through the control valve and enter the air start manifold.

However, when compressed air is needed for the engine to start, the drain valve is forced downward, either by air that is brought in through the control air inlet by pilot air or by the pin attached to the manual start lever. When the drain valve is in the proper position, pressurized air above the main spring in the start control valve escapes through the vent ports at the top of the valve body.

The resulting release of pressure allows the main starting valve to overcome the spring force and to lift off its seat. In turn, compressed starting air is permitted to pass through the control valve and into the air start manifold. Starting air will continue to pass through the main air start control valve until the drain valve is closed. When the drain valve closes, the area above the piston re pressurizes and forces the main starting valve back on its seat. This reseating action shuts off starting air through the control valve[6].

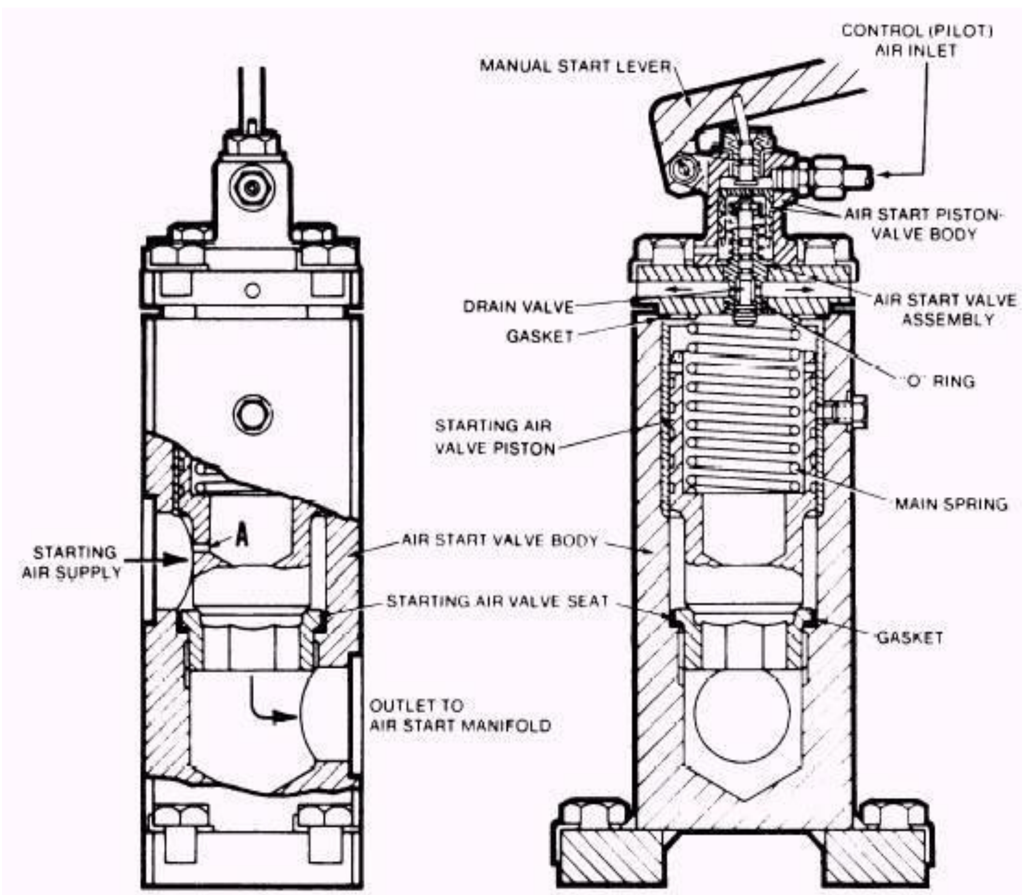


Figure 1.6. Main air start control valve[6].

The air start manifold runs parallel to the right bank of cylinders. Jumper lines are connected only to the right bank of cylinders because only one bank of cylinders needs to be attached to the air starting system. The jumper lines connect the air start manifold to the air start check valves in the cylinder head and supply the required air pressure to the cylinders.

The entry of starting air into the cylinders is controlled by pilot air. Pilot air leaves the main air start control valve, passes through an air filter and an oiler, and enters the air start distributor. The air distributor is directly driven by the right camshaft. As the camshaft rotates, pilot air is allowed to pass through a line to the appropriate air start check valve in the firing sequence. Pilot air opens each check valve to allow high-pressure starting air to enter the cylinder from the jumper line, force the piston downward, and rotate the crankshaft.

The air start check valve functions as a pilot-actuated air admission valve until the cylinders begin to fire. As the cylinders fire, pressure created by combustion exceeds the pressure of the starting air, this condition forces the air start check valves to close, preventing combustion gases from entering the air start manifold. At the same time that starting air is delivered to the cylinders, control air at 50bar is supplied to the forward end of the pneumatic auxiliary start/stop relay and the fuel rack activator piston.

The control air causes the piston to pull the fuel racks toward the full-fuel position. Once the engine is started, its speed is controlled by a hydraulic governor with a pneumatic speed mechanism attached for remote engine control. The governor is also equipped with a dial that can be used for local control of engine speed during maintenance, or in an emergency.

The control air that operates the main air start control valve passes through a primary safety device, the barring gear interlock. The barring gear interlock prevents the engine from accidentally starting or rotating and also prevents the barring gear from being engaged when the engine is running. The safety feature is provided by the barring lever assembly. When the barring lever assembly is in place, the interlock will not allow control air to reach the main air start control valve, and compressed air will not be admitted to the cylinders. Basically, all air starting systems operate similarly and contain components that are similar in function to those used in the two types of air starting systems we have discussed[6].

1.3 Thesis Objectives

1.3.1 General Objective

Based on the Literature have been conducted on the existing engine starting system that means electrical starting system only is not safe so that it needs additional system to support electrical staring system, both electrical starting system and Compressed Air Engine starting system for light Diesel-Electrical Passenger Locomotives are going to increase the efficiency dramatically.

In this MSC thesis, we will model, Analysis Third stage air compressor and pressure vessel and simulate a simple Compressed air engine starting system for light diesel electrical locomotive, system with a consideration Specification of starting system units using standard manual, the system must be compatible with cost and arrangement of locomotive, the existence starting system is not functioning that means when the battery is dead, connection loss, starter motor failed the compressed air staring system comes in to operation to start the engine.

1.3.2 Specific Objectives

This research work is one of the works which are totally engaged to Investigation of Compressed Air Engine starting system of light diesel Electrical locomotive(i.e the compressed air directly ejected in to engine cylinder according to the firing order) by numerical analysis, modeling and simulating using software. The Design pressure (the amount of air pressure to push the piston downward) should be aware, the type, deriving mechanism and capacity of third stage air compressor, the total number and the capacity of air cylinder, the opening pressure of pressure control valve, the actuating mechanism of pressure control valve and electro pneumatic valve, what types of air distributor needed for diesel electrical locomotive engine and where it get drive, different types of air filter used.

More specifically this MSC thesis has the following objectives:

1. To develop a new technology in Railway industry compressed air engine starting system and its working principle.
2. To develop mathematical analysis of the system.
3. To develop 2D/3D model of the system using CATIA and Auto cad software.
4. To develop simulation 3D model of the system using CATIA and ANSYS Workbench.

1.4 Statement Of The Problem

In the electrical starting system of diesel-electrical locomotive is a power full and more efficient but it needs additional system like compressed air engine starting system.

This Thesis is a presentation of a new contribution of railway industry for starting of the engine by compressed air without involving of electrical energy but the system installed additionally electrical starting system. It argues that considerable advantages can be obtained by the make use of compressed air engine starting. When electrical starting system or battery system may be failed the compressed air engine staring system come into operation to start locomotive engine, In addition to mention the above the system can be used cleaning of the locomotive units of dust by blowing them off with compressed air during maintenance. Compressed air engine starting system improved the efficiency of railway transport system indirectly by avoiding of traffic jamming.

This system is new in railway industry, but it is possible implemented because the system is very light weight and in expensive installation and operated cost. The system is essential to railway industry since locomotives are difficult to start by pushing or pulling like automotive.

1.5 Scope of the Research

The research work will concentrated on Investigation of compressed air engine starting system using analytical analysis, system modeling and simulate software. The scope of the research is limited to work on Physical and theoretical analysis.

1.6 limitation of the Research

Along with the research work some challenges may be encountered which may be imposed on the smooth journey of the research, shortage of latest and appropriate materials which are used for simulation and numerical analysis, shortage of recent most related papers and books.

1.7 Thesis Organization

The report consists of six chapters, first Chapter, being introduction of engine starting system. In second chapter, literature review of various literatures on the compressed air engine starting system. Third chapter, modeling of compressed air Engine starting system, Analysis, selection and operation of pressure control valve, pressure reducing valve, electro pneumatic valve and piston type third stage air compressor. Chapter four deals Analysis of pressure vessel, selection of Air dryer, selection of Air distributor, Selection of air starting valve, cost analysis and compatibly of the system. In Chapter five the results obtained by FEM or software and analytically are discussed. Chapter six concludes with the conclusion, Recommendation & scope for future work.

CHAPTER TWO: LITERATURE REVIEW

2.0 Compressed Air Engine starting system

Some gas turbine engines and Diesel engines, particularly on trucks, use a pneumatic self starter. In ground vehicles the system consists of a geared turbine, air compressor and a pressure tank. Compressed air released from the tank is used to spin the turbine, and through a set of reduction gears, engages the ring gear on the flywheel, much like an electric starter. The engine, once running, drives the compressor to recharge the pressure vessel[8].

Aircraft with large gas turbine engines are typically started using a large volume of low-pressure compressed air, supplied from a very small engine referred to as an auxiliary power unit, located elsewhere in the aircraft. Alternately, aircraft gas turbine engines can be rapidly started using a mobile ground based pneumatic starting engine.

On larger diesel generators found in large shore installations and especially on ships, a pneumatic starting gear is used. The air motor is normally powered by compressed air at pressures of 10–30 bar. The air motor is made up of a center drum about the size of a soup can with four or more slots cut into allow for the vanes to be placed radially on the drum to form chambers around the drum. The drum is offset inside a round casing so that the inlet air for starting is admitted at the area where the drum and vanes form a small chamber compared to the others. The compressed air can only expand by rotating the drum, which allows the small chamber to become larger and puts another one of the cambers in the air inlet. The air motor spins much too fast to be used directly on the flywheel of the engine; instead a large gearing reduction, such as a planetary gear, is used to lower the output speed. A Bendix gear is used to engage the flywheel[6].

Large Diesel generators and almost all Diesel engines used as the prime mover of ships use compressed air acting directly on the cylinder head. This is not ideal for smaller Diesels, as it provides too much cooling on starting. Also, the cylinder head needs to have enough space to support an extra valve for the air start system. The air start system is conceptually very similar to a distributor in a car. There is an air distributor that is geared to the camshaft of the Diesel engine; on the top of the air distributor is a single lobe similar to what is found on a camshaft. Arranged radially around this lobe are roller tip followers for every cylinder.

When the lobe of the air distributor hits one of the followers it will send an air signal that acts upon the back of the air start valve located in the cylinder head, causing it to open. Compressed air is provided from a large reservoir that feeds into a header located along the engine. As soon as the air start valve is opened, the compressed air is admitted and the engine will begin turning. It can be used on 2-cycle and 4-cycle engines and on reversing engines. On large two-stroke engines less than one revolution of the crankshaft is needed for starting[12].

Since large trucks typically use air brakes, the system does double duty, supplying compressed air to the brake system. Pneumatic starters have the advantages of delivering high torque, mechanical simplicity and reliability. They eliminate the need for oversized, heavy storage batteries in prime mover electrical systems. we had seen two research on nuclear power plant and marina technology.

2.1 Marin Engine Air starting system

This research performed by GARANG D. in 2000, Large marine diesel engines use high pressure compressed air to start them. The air flows into the cylinder when the piston is moving down the cylinder on the power stroke. To minimize the risk of an air start explosion, fuel is not injected into the cylinder whilst the air is being admitted.

Air start systems vary in their design and can be quite complex. There will be a means to start the engine locally as well as from a remote location (The Bridge or the engine control room)[14]. This system is not representative of one type of engine but is simplified to give a basic understanding.

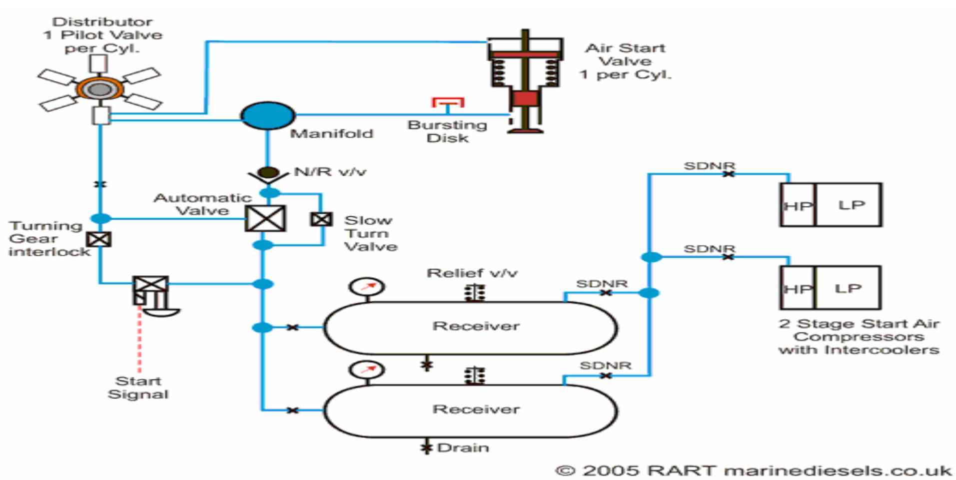


Figure:2.1 marine engine compressed air starting system[14]

Air Start Distributor

The air distributor normally consists of a series of pilot valves, one for each cylinder arranged radially around a Cam. Timed to the engine and driven from the camshaft, the distributor opens the main air start valves in the correct sequence.

Figure:2.2 engine air distributor[14]

Air Start Valve

The air start valve is located in the cylinder head. When it is opened by the air signal from the distributor, compressed air at 30 bar flows into the cylinder, forcing the piston down.

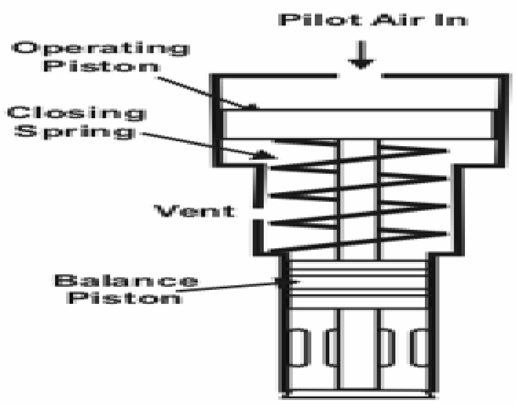


Figure:2.3 Air starting valve[14]

Bursting Disk

The connection to each air start valve is fitted with a protection device. This can be either a flame trap or a bursting disk. The flame trap will prevent any combustion in the cylinder passing to the air start line and causing an explosion, whereas a bursting disk will limit any pressure rise by bursting.

Automatic Valve

The automatic valve is only open whilst an air start is taking place. It incorporates a non return valve to prevent any explosion in the air start system getting back to the air receivers. A slow turn valve is incorporated in the smaller bore pipe work to the side of the valve. This is used to turn the engine slowly before starting, to prevent damage which could be caused if liquid had found its way into the cylinder[14].

The valve shown in the Figure and the diagram opposite is from an MAN B&W slow speed two stroke. The valve itself is a simple ball valve which is turned through 90° by pneumatic actuator. The actuator consists of a central spindle with gear teeth machined onto it. This is rotated by two racks which are driven by pistons.

Two guide rods which maintain the alignment of the pistons and racks are bored to allow air to either side of the pistons. (only one guide rod is shown on the diagram) When a start signal is given, the space behind the pistons is pressurized and they move together, rotating the spindle and opening the ball valve. At the end of the start sequence air is admitted through the second guide rod (not shown) pressurizing the space between the pistons, moving them apart and closing the valve.

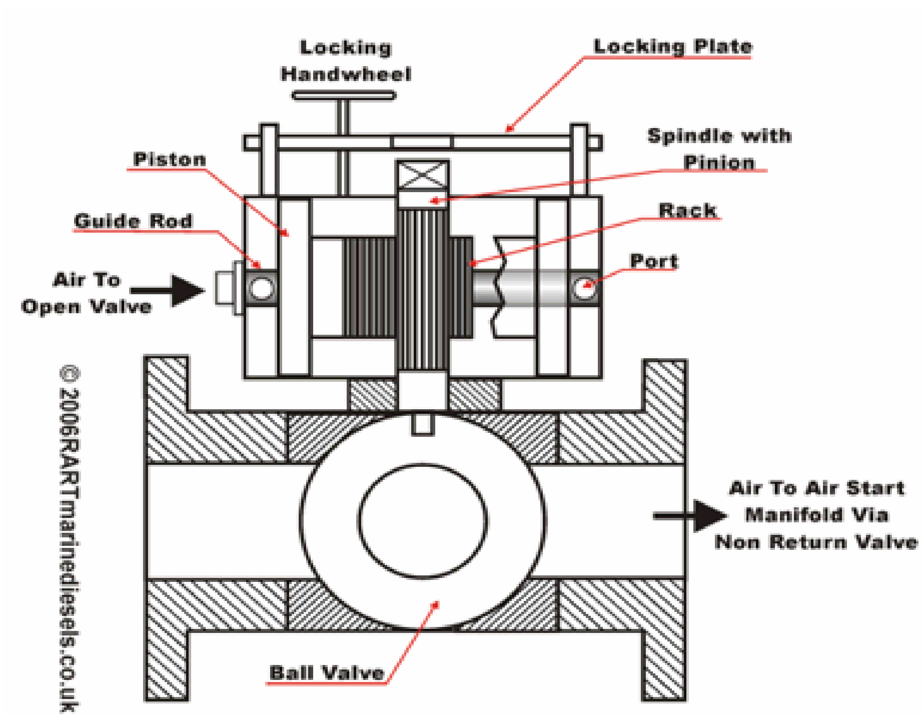


Figure:2.4 Automatic valve operating principle MAN B and W[14]

Turning Gear Interlock

The turning gear interlock is a control valve which will not allow starting air to operate the system when the turning gear is engaged.

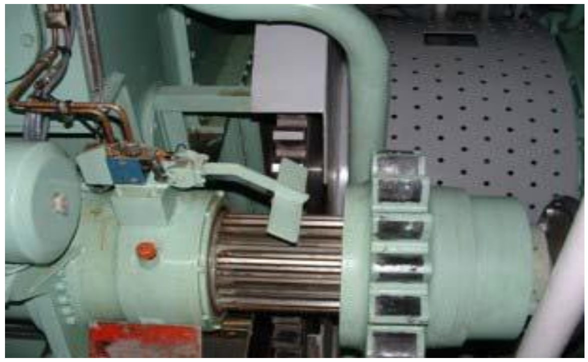


Figure:2.5 Turning Gear Interlock[14]

Air Receiver

Two air start receivers are fitted. The total capacity of the receivers must be sufficient to start the Engine 12 times alternating between ahead and astern without recharging the receivers. In the case of a unidirectional engine, then the capacity must be sufficient for 6 starts. The air receiver will be fitted with a relief valve to limit the pressure rise to 10% of design pressure. A pressure gauge and a drain must also be fitted. A manhole gives access to the receiver for inspection purposes.

Figure:2.6 Air Receiver[14]

The Air Compressor

Two air start compressors are normally supplied which must be capable of charging the air receivers from empty to full it takes one hour. They are usually two stage reciprocating with inter and after stage cooling. Relief valves will be fitted to each stage which will limit the pressure rise to 10% of design pressure, and a high temperature cut out or fusible plug to limit the HP discharge to 121°C. Intercoolers are also fitted with bursting disks or relief valves on the water side[14].

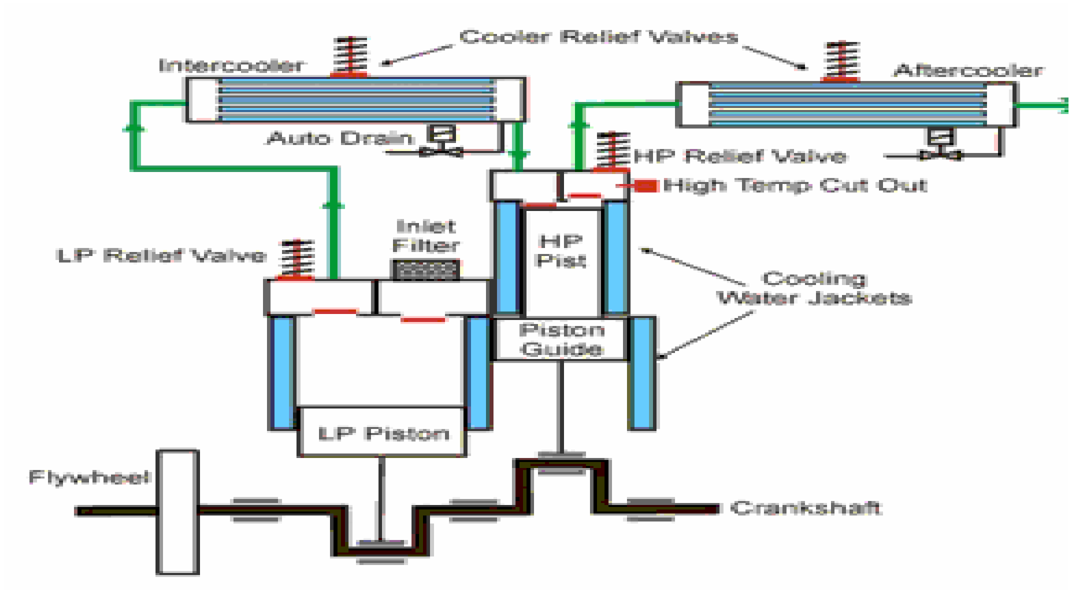


Figure:2.7 Tow stage starting air compressor[14]

2.2 Starting Air Systems Nuclear plant application

This research performed by JHONE G. in 2001, There are two types of compressed air starting systems used in nuclear plant applications: the direct air injection system and the air motor system shown in Figure below. In both systems, air is stored under pressure in air receiver tanks. Quick opening solenoid operated air admission valves prevent air from reaching the engine until actuated by an Engine start signal[12].

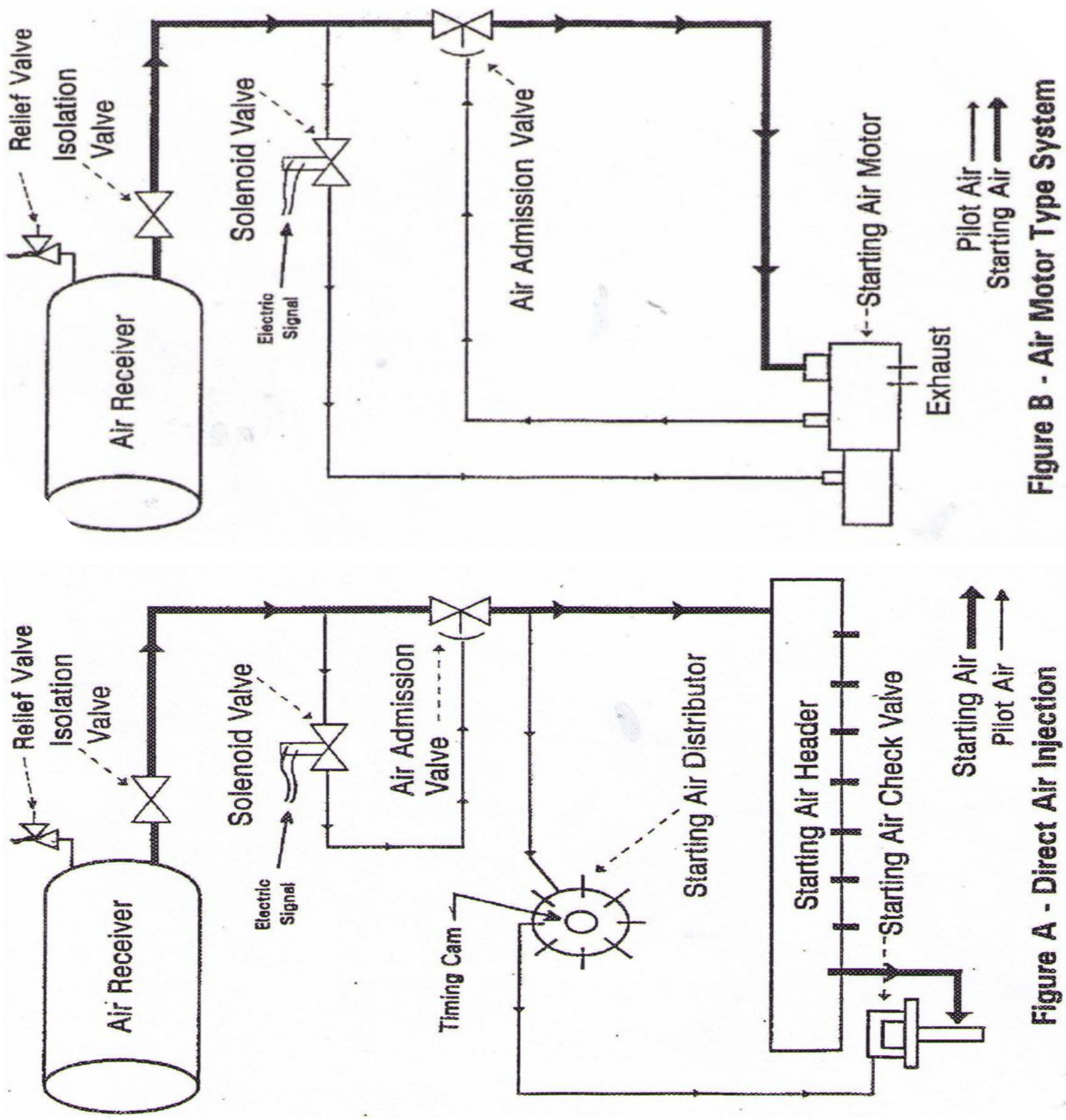


Figure.2.8 nuclear plant Starting Air System[12]

2.2.1 Direct Air Injection Systems

This system is also often referred to as the 'air over piston' or the 'cylinder air start' system. With the direct air injection system (*Figure A*), air under pressure is applied to the inlet of the Starting Air Admission Valve and to the Starting Air Solenoid Valve. With no start signal present, both valves remain closed. Upon receiving a "start" signal, the Solenoid Valve opens, directing air to the Air Admission Valve pilot section, causing the main valve to open. Downstream of the Air Admission Valve, the air flow is split[12].

The major portion of the air is passed to the Starting Air Header. A small portion of the air is also directed "t" the Starting Air Distributor. Air from the Starting Air Header enters each of the Starting Air Check Valves. Starting Air Check Valves, where used, are mounted in each of the engine cylinder heads. For the Fairbanks-Morse Opposed Piston engine, the starting air check valves mount through the wall of the cylinder liner.

The design of these valves is such that they will remain closed until acted upon by a pilot signal from the Starting Air Distributor. The Starting Air Distributor, driven by the engines camshaft or accessory drive gear train, provides a pilot air signal in proper sequence to each of the Starting Air Check Valves. At the appropriate time, usually a few degrees past Top Dead Center (TDC), the pilot air signal is applied to each Starting Air Check Valve, causing it to open.

Starting air, under pressure, is admitted directly into the engine cylinder. This air pressure rapidly pushes each piston downward in succession. As each piston is later moving upward, the air charge is compressed in preparation for injection of the fuel. As the engine achieves self-sustained operational speed, a signal from the control system shuts off the starting air supply.

This is usually controlled by a speed sensing device monitoring the engine speed (RPM). The Solenoid Valve then closes, allowing the Air Admission Valve to close. Air then vents from the Starting Air Distributor and Starting Air Header. This allows the spring-loaded Starting Air Check Valves to close and returns the engine starting air system to the standby mode.

2.2.1.1 Starting Air Admission Valves

Air Admission Valves are typical pilot-operated, poppet type Starting Admission Valve. The Solenoid Valve is generally mounded directly to the Admission Valve. In other applications, the Solenoid Valve may be mounted remotely to the Admission Valve and connected by pipe or tubing. In the standby mode, the spring-loaded valve poppet is held in the closed position by the force of the spring. Air from the air receiver enters the valve as shown below[12].

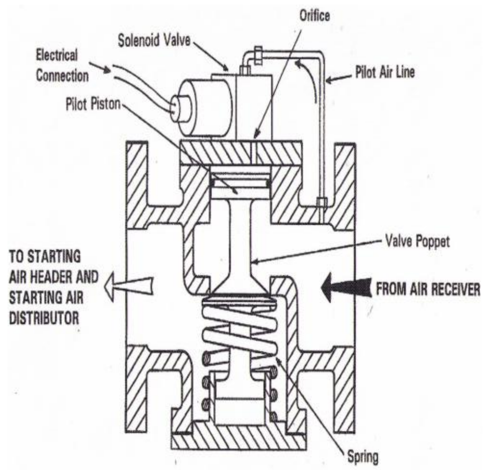


Figure 2.9 Starting Air Valve (Solenoid Activated)[15]

The design of the valve poppet is such that the air pressure acting against the poppet is balanced by the equal areas of the valve poppet. The force of the spring is sufficient to keep the valve closed. A portion of the air entering the valve (pilot air) is directed to the inlet of the Solenoid Valve. In standby, the Solenoid Valve is also closed (not energized). When a "start" signal is generated, it is applied to the Solenoid Valve. This signal energizes the solenoid coil.

The magnetic field created lifts the valve plunger, opening the valve. Pilot air is then directed through the signal port (orifice) to the top of the pilot piston (part of the valve poppet). The force of the air pressure applied to the pilot piston exceeds the force of the spring holding the poppet in the closed position. This differential force causes the valve poppet to move downward, opening the valve. Air from the air receiver passes freely through the valve and on to the Starting Air Header and Starting Air Distributor. Loss of a "start" signal closes the Solenoid Valve, which vents air from the pilot piston. Spring force causes the valve poppet to close returning to a standby mode.

2.2.1.2 Starting Air Distributor

Since the Starting Air Check Valves must open at a specific time relative to the position of the piston in the cylinder, a Starting Air Distributor is needed and provided. The Starting Air Distributor, which is timed from the engine camshaft(s) or other portion of the engine gear train, directs a pilot signal to each Start Air Check Valve, when the piston is slightly after top dead center and maintains that signal until the movement of the piston reaches approximately 120^{o} after top dead center [12].

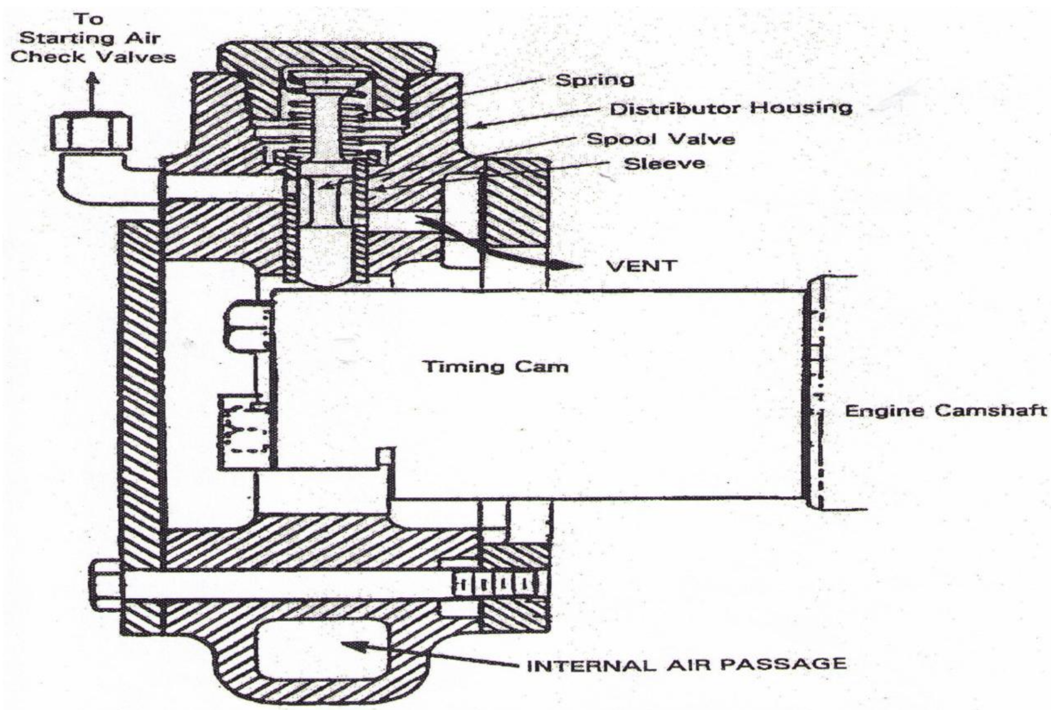


Figure 2.10 Starting Air Distributor (FM OP Engine)[12]

The Starting Air Distributor consists of a series of spool valve assemblies, one for each Starting Air Check Valve. One assembly for each cylinder is mounted in a common housing. The timing cam, timed to the engine, causes each spool valve to direct pilot air pressure to or vent the pressure from each of the Starting Air Check Valves. Operation of the spool valve assemblies is shown in Figures 2.11 through 2.13. With the starting air admission valve closed, no air is applied to the Starting Air Distributor. With no air pressure present, the distributor spool valves are held in the off cam or vent position by the force of the spring as shown in Figure 2.11. This prevents the spool valve from contacting the timing cam during normal engine operation or standby.

When a 'start' signal is generated, the air admission valve opens supplying high pressure starting air to the internal passages of the distributor housing. This air pressure forces the spool valves downward into contact with the timing cam as shown in Figure 2.12. In the high cam position, the spool valve is held in the vent position, and no pilot air signal is applies to the associated Starting Air Check Valve. Rotation of the timing cam will eventually align the low portion of the cam with the spool valve as shown in Figure 2.13[12].

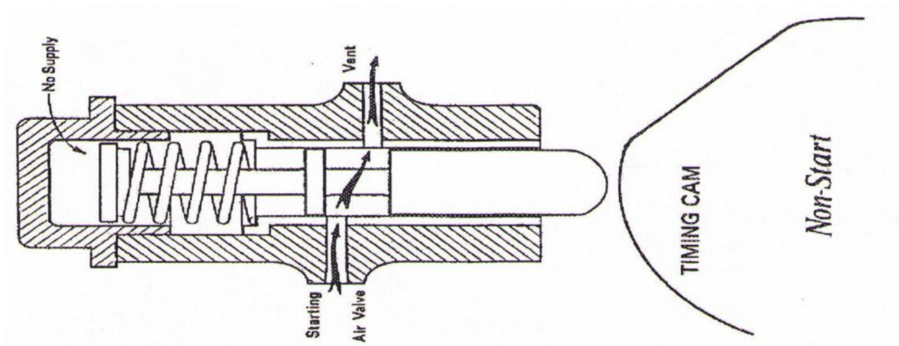


Figure 2.11 Spool Valve Off Cam[12]

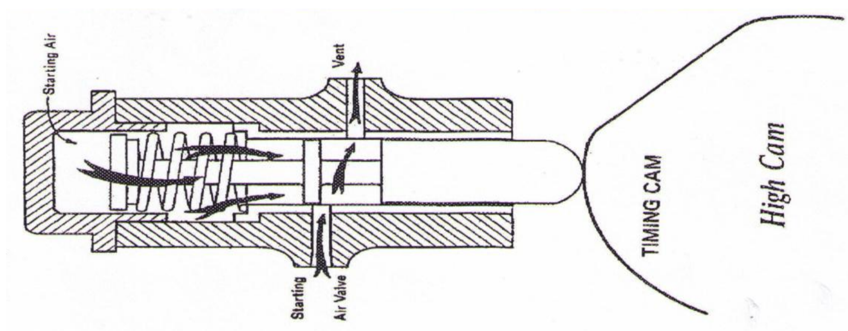


Figure 2.12 Spool Valve High Cam[12]

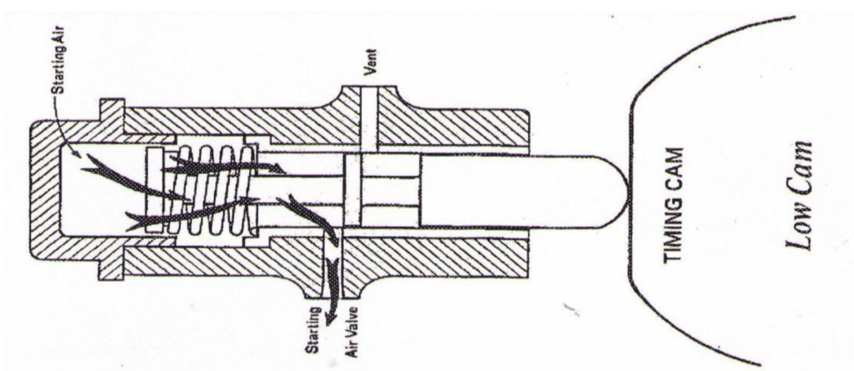


Figure 2.13 Spool Valve Low Cam[12]

The pressure of the starting air from the internal passages of the distributor housing will keep the spool valve in contact with the timing cam. The spool valve moves downward blocking the vent port and connecting the Starting Air Check Valve port. Starting air pressure is then applied to the pilot piston of the associated Start Air Check Valve, opening the valve and allowing starting air to enter the engine cylinder.

When the timing cam has moved beyond a spool valve position, the spool valve is returned to the vent position and the air signal is removed from the Starting Air Check Valve. With cessation of the 'start' signal, the Air Admission Valve closes allowing the air to vent from the Starting Air Distributor internal passages. This loss of air pressure allows the force of the spring to lift the spool valves off the timing cam. All of the spool valves will be returned to the off/vent position, and the starting process will be terminated[15].

2.2.1.3 Starting Air Check Valves

In the start mode, air from the Starting Air Header is delivered to each of the Starting Air Check Valves mounted in the engine cylinder heads. In the Fairbanks-Morse Opposed Piston engine, the Starting Air Check Valves mount through the wall of the cylinder liner adjacent to the fuel injection nozzle[12].

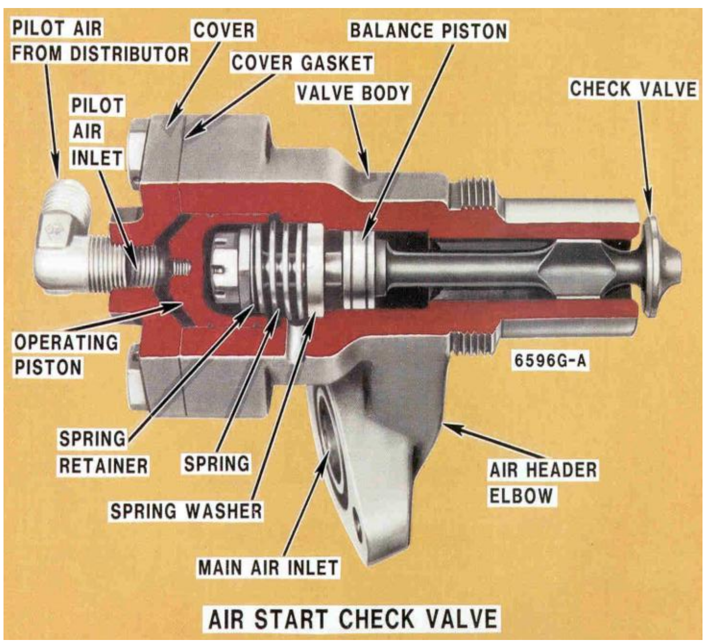
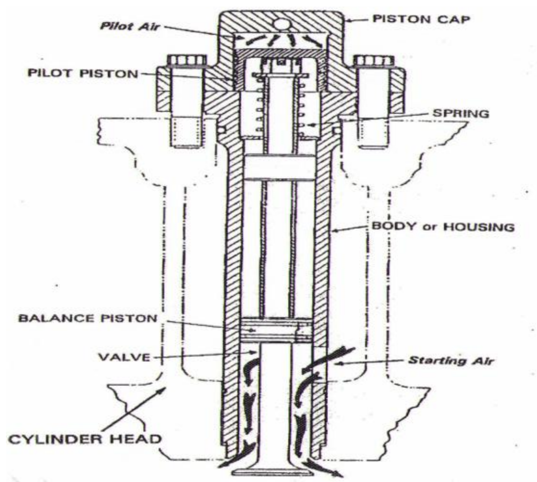


Figure 2.14. Air Start Check Valve[12]

A typical Starting Air Check Valve is shown in Figures 2.15 and 2.16. The air start check valve assembly includes a poppet type valve which is located in a precision bore cast body or housing. A spring applied to the upper end of the valve keeps the valve in the closed position.



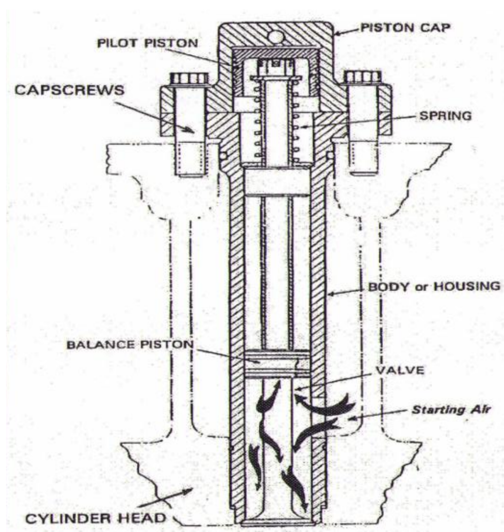


Figure 2.15 Starting Air Check Valve - Open, Figure 2.16 Starting Air Check Valve - Closed[12]

The design of the valve including the balance piston is such that the starting air entering the valve body is balanced. Starting air pressure acting against the valve is balanced by the air pressure acting against the equal area of the balance piston. The force of the spring is sufficient to hold the valve in the closed position. A pilot inside the piston cap rests against the tip of the valve. Initiation of a 'start' signal opens the Air Admission Valve allowing air to enter the housing as shown. The balanced design of the valve maintains it in the closed position until a pilot air signal is applied to the pilot piston[15].

As mentioned previously, air from the Starting Air Admission Valve is directed to the Starting Air Distributor as well as the Starting Air Header. The Starting Air Distributor is timed to direct a pilot signal to the Starting Air Check Valve, when the piston for the cylinder involved is just past top dead center. The force supplied by the pilot air signal, which is applied to the pilot piston, overcomes the spring force holding the valve closed. This opens the Starting Air Check Valve allowing starting air to enter the engine cylinder. The pressure of the starting air entering the engine cylinder is applied to the top of the engine's piston forcing the piston downward and causing rotation of the crankshaft. This rotation subsequently causes upward motion of other engine pistons.

As each engine piston moves upward on the compression stroke, the air in that cylinder is compressed to achieve the temperature required for ignition. Rotation of the crankshaft also causes rotation of the timing cam for the Starting Air Distributor. As the engine's piston nears the bottom of its stroke, the Starting Air Distributor closes the associated valve port and vents the pilot air signal from the Starting Air Check Valve.

This allows the spring to return the valve to the closed position. As this occurs, other Starting Air Check Valves are receiving the pilot signal which results in a continuous rotation of the engine until engine start is achieved or the start signal ceases. The Starting Air Check Valves have to remain closed during normal engine operation. If a Starting Air Check Valve should not close or should leak, the combustion gases from the cylinder will leak out of the Starting Air Check Valve, and it will become extremely hot and may further seize. It is important to occasionally check the Starting Air Check Valves and the associated Starting Air Supply Header to determine that none of the valves are leaking.

2.2.1.4 Air Receivers

Compressed air receivers are cylindrical pressure vessels whose design and construction is governed by the requirements of the ASME Boiler and Pressure Vessel Code, Construction of Unfired Pressure Vessels. The generic plant air receivers are designed to conform to the requirements of ASME Code. Their specific size (volume) is a function of the size of the engine they are required to start and the number and duration of start attempts specified by the FSAR. The generic FSAR requirement is to have air storage capability to provide two (2) consecutive diesel engine starts. The two(2) start capability is required to assure that the engine can help mitigate the consequences of design basis events[13].

2.2.1.5 Air Compressors

Starting air compressors are normally of the reciprocating (piston) type. They are usually multiple-stage, multi-cylinder types with cylinders configured in a “V” or “W”. They may be either air cooled. Compressor capacity is based upon the amount of time required to recharge the receiver from a specified (cut-in) pressure to the desired operational pressure.

Compressor capacity may be determined using the following formula[11]:

$$D = \frac{(V*P_2)-(V*P_1)}{15*T_1} \qquad (2.1)$$

Where: D = Compressor capacity, cubic meter of free air per second.

V = Volume of the Receiver, cubic Meter

P_1 = Cut-in Pressure, absolute (pa)

P_2 = Desired operating pressure, absolute (pa)

T = Recharging time, in second

2.2.1.6 Pressure Switches

Operation of the compressor is controlled by a pressure switch located on the air receiver. Set at a specified "cut-in" pressure, the switch starts the compressor when the pressure becomes lower than the set pressure and allows the compressor to run until the receiver is at or slightly above the desire operating pressure. As the air pressure reaches the operating pressure, the pressure switch opens the circuit, stopping (unloading) the compressor[15].

2.2.1.7 Pressure Relief Valves

To protect the system from over pressurization, pressure relief valves are installed near the discharge of the compressor and on the starting air receiver. The safety valve near the compressor discharge is normally set at 115% of the rated operating pressure. The safety valve installed on the air receiver is normally set at 110% of rated pressure[15].

2.2.1.8 After coolers

As air is compressed, its temperature increases substantially. This reduces the density of the air being discharged from the compressor. To compensate for this condition, after coolers are installed downstream of the compressors. Either air cooled or water cooled, these after coolers cool the air to 310.85K prior to its entering the air dryer. Air-cooled units often use force draft fans to increase their cooling capacity[15].

2.2.1.9 Air Dryers

Air taken from the atmosphere contains a certain amount of moisture (humidity). Air operated equipment including starting air motors and various valves may become damaged or degraded if exposed to excess moisture in the compressed air. Moisture may be carried by the air into the air receivers. This moisture may condense and collect in the receiver causing localized corrosion. This corrosion could lead to weakening of the receiver and failure of the components. Moisture carried by the air past the receiver into the system can damage the starting air system components and/or restrict air flow through small orifices and passages[13].

Air Dryers usually use a combination of desiccant and temperature reduction (refrigerants) to remove unwanted moisture from the compressed air. As the air passes through the desiccant tower, the pellets of desiccant absorb and hold the moisture. The combination of these two methods removes most of the moisture from the compressed air prior to its reaching the receiver. This protects both the receivers and the air-operated components of the system.

CHAPTER THREE: MODELING AND ANALYSIS

3.1 Modeling of Compressed Air Engine Starting System for Diesel Electrical locomotive

In Fluid and Thermo dynamic theory are applied to develop a simplified Model of compressed air Engine starting system for Diesel-Electrical locomotive described in Section 3.1. The system consists of Third stage Air Compressor, pressure Relief Valve, Mechanical Air Filter, Oil and moisture separator, Air Pressure Vessel and piping, pressure Reducing Valve, sediment Bowl, Electro pneumatic Valve, Engine Air Distributor, Air Starting Valve and Diesel Engine.

When Button Switch or lever of Electro pneumatic Valve is pressed, Electro pneumatic Valve is opened, compressed air flows from cylinder 120N/mm^2 to air distributor of the Engine Cylinder via pipeline. From the air Distributor compressed air flows into pipe line, and further though the starting Valve into Engine Cylinder whose piston is on working (power) stroke position. Under the action of compressed air the Engine crankshaft and consequently the distributing disk of air Distributor will be turned, and compressed air will be delivered into the next Engine Cylinder according to the firing order.

As the air is consumed from Cylinder 120N/mm^2 (working pressure Vessel) the air pressure there in, and, consequently in the pipe line that supplies air to the Engine is reduced. In this case compressed air from cylinder 150N/mm^2 (reserved pressure Vessel) charge to a pressure of 150N/mm^2 flows through a Filter into Air Pressure Reducer where the pressure of compressed air reduced to a value of 70N/mm^2.

From the Pressure Reducer compressed air is delivered into the Engine air mains through a 2mm Throttling orifice which regulates normal consumption of compressed air; in this case a pressure of 70bar is automatically maintained all the time. Upon starting the Engine, supply of air into Air Distributor by means of Electro pneumatic Valve is stopped. Simultaneously with the beginning of Engine operation, Compressor starts operation and replenishes the stock of compressed air in the Pressure vessel..

Compressed air from a Compressor passes in succession though Oil and Moisture separator, Filter, Pressure Relief Valve, Sediment bowl, and then flows to the Cylinder 150N/mm^2 (reserved pressure vessel). Part of compressed air passes through Filter and Pressure Reduced valve and files Pressure Vessel 120N/mm^2 (working pressure vessel) bullied up a pressure of 70N/mm^2 there in.

When compressed air pressure in Pressure Vessel 150N/mm^2 Reserved pressure vessel) reaches a value of 150+-15N/mm^2, air pressure Relief Valve switches over a Compressor for idling. When a pressure in Pressure Vessel 150N/mm^2 (Reserved pressure Vessel) becomes reduced approximately to a value of 120N/mm^2, the air pressure Relief Valve switches over the compressor for charging the Pressure Vessel with compressed air.

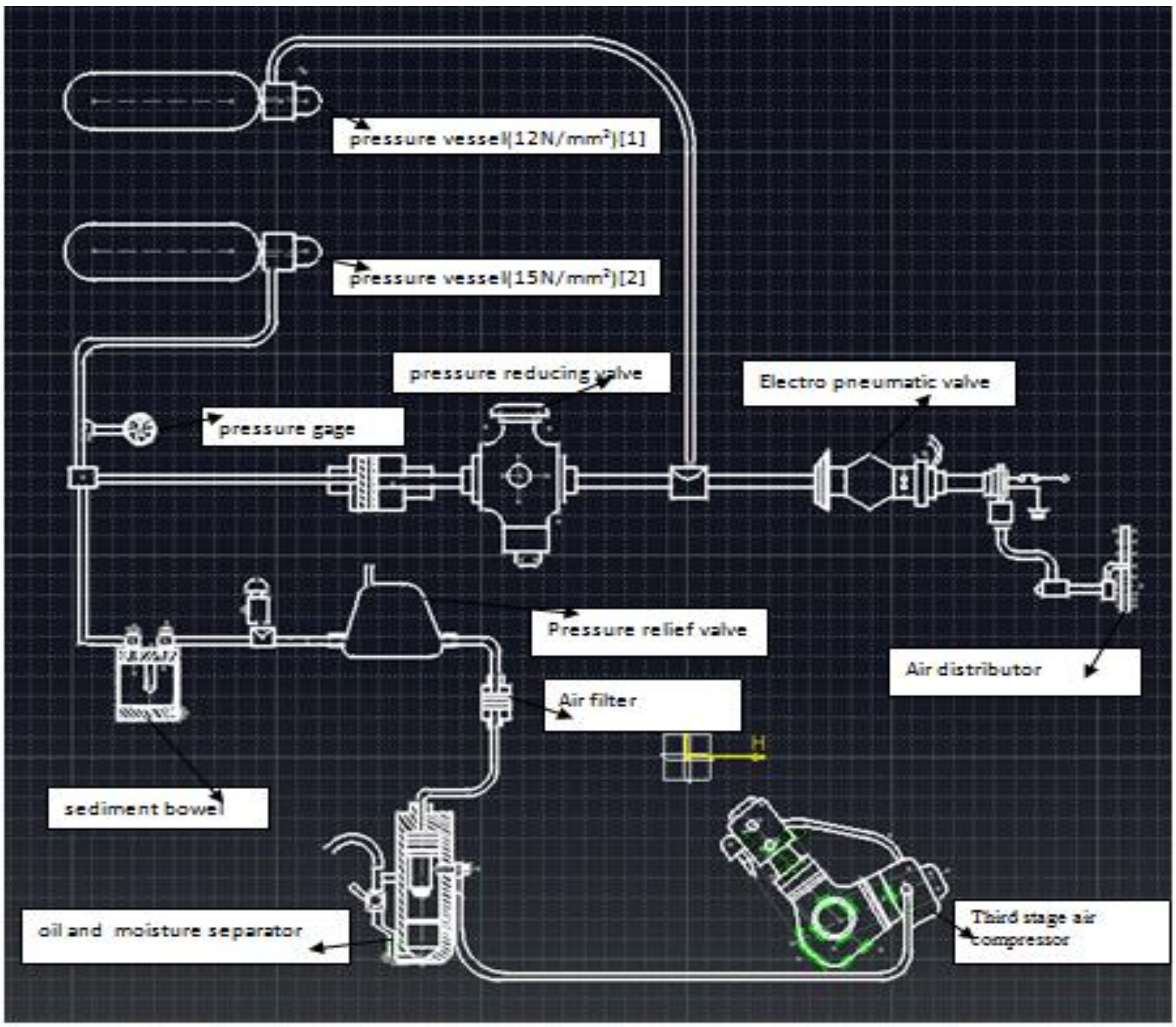


Figure 3.1:-modeling of compressed air engine starting system for Diesel-electrical Locomotive

3.2 Diesel Electrical Locomotive Engine Main Parameters

- ✓ Model 12V240ZJ
- ✓ Rated power 2400kW
- ✓ Max. service power 2200kW
- ✓ Rated speed/idle speed 1000/430r/min
- ✓ Number of cylinder/arrangement type: 12 cylinders/V type with inclusion angle of 50^0
- ✓ Combustion chamber type: Direct injection, open type
- ✓ Cylinder diameter X piston stroke 240mm×275mm
- ✓ Fuel consumption at rated power 207 + 3％g/kW·h
- ✓ Outline dimension: 4298mm×1760mm×2642mm
- ✓ Weight: 16.3t
- ✓ Diesel engine staring mode Electric motor(maximum Torque is 190Nm)
- ✓ volumetric displacement is $V_D=1.705*10^{-4}m^3/rev$
- ✓ Total displacement 149.29 L
- ✓ Compression ratio 12.4
- ✓ Max. combustion pressure ≤14 MPa
- ✓ Crankshaft rotation direction (face output end) Clockwise

3.3 Required pressure calculation

The required pressure can be calculated by:-

$T = \frac{PV_D}{6.28}$..(3.1)

Where:- P:compressed air pressure in Pascal

:-V_D: is volumetric displacement measured in m^3/rev

:-T maximum torque(N m)

Given: maximum torque is T=190Nm

: volumetric displacement is $V_D=1.705*10^{-4}m^3/rev$

$P = \frac{6.28*T}{V_D} = \frac{6.28*190Nm}{17.05} = 7Mpa$=7N/mm^2

3.4 Design Of Third Stage Air Compressor

3.4.1 Introduction

The standard practice in the compressed air Engine starting system application of compressors is to apply positive displacement Third stage air cooled and Reciprocating type of compressor for low flows usually within limits of about 900 m^3/min. Over and above this, the dynamic design Centrifugal type Compressors is the choice by most application area. Reciprocating Compressor Equipment Manufacturers, design various sizes to meet with the variable process conditions. In the dual-acting design, compression occurs either when the piston moves on the inward stroke, toward the crankshaft, or outward stroke, away from the crankshaft. Thus, compression occurs on both sides of the piston. For better operational stability and to minimize dynamic loads in operation, a variant of the third-acting with opposed cylinder-piston connection of equal sizes, in pairs, arranged horizontally and connected to a common shaft called a Tandem cylinder arrangement, allows for load balancing. Such units are usually applied to handle larger gas compression requirements and are made up of third stages of compression, with each stage being a cylinder-piston connection.

Equipment Sizing or Selection for a Reciprocating Compressor is on a stage-by-stage evaluation basis. Important Considerations in such analysis as stated by [10] will include amongst others:

- Temperature rise across a stage
- Frame or Piston rod loading
- Total pressure rise across a stage and Valve design and losses

3.4.2 Mathematical relationships for reciprocating compressors

3.4.2.1 Overall Compression ratio

$$r_c = \frac{P_2}{P_1} \qquad (3.2)$$

Where P_2= Discharge or Final Delivery Pressure (bar) **,** P_1 = Suction Pressure (bar)

Knowledge of overall or total compression ratio, r_c , i.e. the ratio of the final delivery pressure to the initial suction pressure, is a guide to first estimation of the number of stages, with corrections being made based on temperature rise limitations. Neerken [1] advises to limit compression ratio for a stage to $\leq = 3.5$.

Gas Properties And Specific Heat Ratio

Gas for compression can be a single gas or a mixture made up of several gases. A gas analysis is usually conducted to determine mixture properties. A good program development guide is to have a stored database of critical properties of different gases. With knowledge of the percentage by volume of the gases, a mole percentage analysis can be conducted to obtain the final mixture properties.

The ratio of specific heats, *k*, is obtained from the table of specific heats at constant pressure.

$$K = \frac{C_p}{C_v} = \frac{C_{pmix}}{C_{pmix}-8.314} \quad (3.3)$$

C_p = Specific heat at constant pressure

C_v = Specific heat at constant volume

C_{pmix}= Gas mixture specific heat

Proper estimation of the value of the ratio of specific heats, *k*, should be based on average of the values at suction and discharge, K_{ave}. Linear interpolation and extrapolation is applied to obtain the mid-point and out of range value for temperature falling between and outside the ratios of specific heats, *k*, in the database.

$$K_{ave} = K_o + \frac{T_{ave}-0}{100}(K_{100} - K_0) \quad (3.4)$$

Or

$$K_{ave} = \frac{C_{p-ave}}{C_{p-ave}-8.314} = \frac{C_{p-0}}{C_{p-0}-8.314} + \frac{T_{ave}-0}{100}\left(\frac{C_{p-100}}{C_{p-100}-8.314} - \frac{C_{p-0}}{C_{p-0}-8.314}\right) \quad (3.5)$$

Where, T_{ave}= Average temperature (K)

$$T_{ave} = \frac{T_d+T_s}{2} \quad (3.6)$$

T_d = Discharge temperature (K)

T_s= Suction temperature (K)

Relative Density and Density of gas

$$\gamma_{sp.sg} = \frac{M_{w-gas}}{M_{w-air}} = \frac{\rho_{gas}}{\rho_{air}} \quad (3.7)$$

3.4.2.2 Loss Factor

Frictional losses in the pistons and rings, piston rod packing, pulsations due to gas surge effects, and oscillations of valves are often accounted for using one of several methods. Scheel [2] suggest that estimates of Pulsation or Pulse Damper Loss be made in line with:

Pulsation or Pulse Damper Loss = 1% of absolute pressure at suction and discharge has provided a curve fit relationship of the Ludwig loss factor, *Lo*, curve as a function of the pressure ratio :

$L_o = -0.002188{r_c}^7 + 0.05778{r_c}^6 - 0.6356{r_c}^5 + 3.7711{r_c}^4 - 13.036{r_c}^3 + 26.331{r_c}^2 - 29.019r_c \quad + 14.929$ ………….(3.8)

3.4.2.3 Approximate Compression Ratio per Stage

$r_{c-stg} = r^{1/N_{stg}}$……………………………………………………………………………………………….(3.9)

Where:

N_{stg} = Number of stages

r_{c-stg}= compression ratio per stage

It was stated earlier that, Single cylinder applications are for low pressure ratios. For high pressure applications, multistage units are the choice to cope with the temperature variations required for efficient performance delivery. The overall compression and hence compression ratio is shared into a number of stages in line with the above equation to allow for equal work in each stage. Such multistage units often installed nearer to the Engine Fan.

Suction Pressure per Stage

Accounting for the pulsation loss, the corrected suction pressure per stage is:

$P_s = P_1 - \mathrm{PulsationLoss}$ ……………………………………………………………………………………….(3.10)

3.4.2.4 Approximate Discharge Pressure Per Stage

$P_{d-stg} = \left(r_{c-stg}\right)(P_s)$……………………………………………………………………………………….(3.11)

3.4.2.5 Inter-Stage Pressure Drop

$P_{i-drop} = 0.1(P_{d-stg})^{0.7}$………………………………………………………………………………….(3.12)

Gas cooling between stages requires, making allowance for the pressure drop in the intercoolers. In selecting compression ratios for a system unit, this is to be considered and adjustments made to computation of the actual discharge pressure in line with above equation.

3.4.2.6 Actual Discharge Pressure

Actual discharge pressure will require that the inter-stage pressure drop be accounted for by adjusting the approximate discharge pressure which is a function of the adjusted suction pressure due to the pulsation losses, and the approximate stage compression ratio.

$P_{d-actural} = P_{d-stg} + p_{i-drop}$ ....(3.13)

3.4.2.7 Actual Compression Ratio per Stage

$r_{c-actual} = \frac{P_{d-actural}}{P_s}$..(3.14)

Optimum gas power is achieved when compression ratios in each of the stages are equal for multistage units. However, limits on the gas power capacity of individual cylinders, makes it impossible to obtain equal and hence, balanced compression ratios.

Suction Valve loss Experienced in filling Cylinder

$\theta_s = (aU)^2[\frac{M_w}{10^4 T_1}]$..(3.15)

Where:

θ_s= Suction valve loss

a = Piston/valve area ratio

U = Average piston speed in meter per second (m/s)

M_w= Gas Molecular weight

T_1 = Suction temperature in Kelvin (K)

Scheel [2] gives Piston/Valve area ratio values between 8 and 12 for modern compressors. The Piston/valve area ratio is dependent on the pressures to be handled and valve opening lift, with higher pressures requiring lower valve lifts. Equally, lower Piston/valve area ratio requires a high lift valve. For good design practice suggests an average value of 10[10].

Required mean pressure to Exhaust the Cylinder displacement

$\theta_d = \frac{\theta_s}{r^{(k+1}/_{k)}}$..(3.16)

Average Piston Speed

$U = 2NL$..(3.17)

U = Piston speed in meter per minute (m/min)

L = Piston Stroke, in meter (m)

N = Design rotating Speed, in RPM

Intrinsic correction factor

$B = \frac{(1+\theta_d)}{(1-\theta_S)}$..(3.18)

The intrinsic correction factor, *B*, extends the normal compression ratio $'r_c'$ to represent the actual effective, compression ratio, $r_{c-actual}$, within the cylinder .

3.4.2.8 Compression Efficiency

This is the ratio of adiabatic efficiency to mechanical efficiency.

$\eta_c = \frac{[r^{(k-1/k)}-1]}{[Br^{(k-1/k)}-1]}$..(3.19)

3.4.2.9 Suction Inlet capacity to the stage

$Q_s = \frac{Q_{std}}{60*24} * \frac{P_{std}}{P_s} * \frac{T_s}{T_{std}} * \frac{Z_s}{1.0}$..(3.20)

Standard pressure and temperature have been set by the international committee on weights and measures. For metric standards, the following values apply:

- P_{std}= atmospheric pressure ≈ 1.01325 bar
- T_{std}=standard temperature=293.15 K (20 deg C)These values are applied in the program.

This is also the standard adopted by the American Society of Mechanical Engineers (ASME) Power Test Codes Committee [7].

Q_{std} = Inlet Capacity volume at standard condition (standard cubic meter per day or m^3/day)

Q_s = Inlet Capacity volume at inlet condition (m^3/min)

P_s = inlet pressure (bar)

T_s = inlet temperature (K) and Z_s = Compressibility at inlet

Mass flow

The mass flow rate or weight flow of the gas.

$M = \frac{Q}{v} = \rho_{gas} * Q$...(3.21)

Where: M = Mass flow rate of gas (kg/min)

Q = Volume flow rate (m^3/min)

ν = Specific volume of gas (m^3/kg) = $\frac{1}{\rho_{gas}}$

Specific volume

$v = \frac{ZRT}{P}$..........(3.22)

Where: Z = Gas Compressibility

R = Gas Constant = 8314/M_w and M_w = Gas Molecular Weight

Adiabatic head of compression

$H_{ad} = Z_{av}RT\left(\frac{k}{k-1}\right)[r^{(k-1}/_{k)} - 1]$..........(3.23)

Where: H_{ad} = adiabatic head of compression (kN.m/kg)

Z_{av}= average gas compressibility at suction and discharge

Note that the adiabatic head equation is also applied on a stage-by-stage basis.

3.4.2.10 Compression Power

$GP = \frac{MH_{AD}}{\eta_c}$ ....(3.24)

GP = Compression Power or Gas Power (kW)

3.4.2.11 Volumetric Efficiency

$\varepsilon_v = 0.97 - C_c[\frac{r_c{}^{1}/_{k-1}}{(\frac{Z_2}{Z_1})}]$..........(3.25)

Volumetric efficiency, ε_v, defined as the amount of gas volume displaced with each stroke of the piston, decreases with an increase in compression ratio. The retained volume at the end of a piston stroke is the Cylinder clearance, C_c, expressed as a percentage of the swept volume and is usually provided by compressor equipment manufacturers. Scheel [2] gives a minimum clearance value of about 10 %. Suggested value is within the range of 10 % to 15 % for equipment selection purposes [10].

3.4.2.12 Cylinder displacement required

$C_{dis} = \frac{Q_s}{\varepsilon_v}$..........(3.26)

Cylinder Diameter

Cylinder diameter, *D*, is based on the area of the Head-end.

$D = \sqrt{(\frac{4A_{he}}{\Pi})}$..........(3.27)

Area of head-end

$A_{hd} = \pi \frac{D^2}{4}$..(3.28)

Where: A_{hd}= Area of Head-end (m^2)

D = cylinder diameter (mm or m)

Area of crank-end

$A_{ce} = A_{hd} - \frac{\Pi d^2}{4}$...(3.29)

Where:

d = piston rod size diameter (mm or m)

A_{ce} = Area of Crank-end (m^2)

3.4.2.13 Frame load in compression

Frame loads are the maximum permissible forces that can be sustained due to the pressures acting on the piston.

$F_{LC} = P_d A_{hd} - P_s A_{ce}$...(3.30)

F_{LC}= frame load in compression (N)

3.4.2.14 Frame load in tension

$F_{LT} = P_s A_{hd} - P_d A_{ce}$...(3.31)

F_{LC}= frame load in tension (N)

Compressibility calculation by Redlich-Kwong equation of state

The calculation of compressibility factor based on the Redlich-Kwong Equation of State (EOS) Corresponding State Method can be defined by the functional relationship, Z=$f(T_r, P_r)$. Results were within reasonable accuracy. The Redlich- Kwong Corresponding State EOS.

$Z^3 - Z^2 + \left[\frac{\Omega_a P_r}{{T_r}^{2.5}} - \frac{\Omega_b P_r}{T_r}\left(1 + \frac{\Omega_b P_r}{T_r}\right)\right] Z - \frac{\Omega_b \Omega_b {P_r}^2}{{T_r}^{3.5}} = 0$...(3.32)

Where: Ω_a= 0.42748 and Ω_a= 0.08664

P_r= Reduced pressure =$\frac{P}{P_c}$

T_r= Reduced temperature = $\frac{T}{T_c}$

P_r= Critical Pressure (bar)

T_r= Critical Temperature (K)

Handling gas mixture

Gas Mixture analysis with the Redlich-Kwong Equation of State is conducted in line with the relations:

$$\frac{T_{cm}}{P_{cm}} = \sum_{i}^{N} x_i \left(\frac{T_{ci}}{P_{ci}}\right)$$

$$\frac{T_{cm}^{2.5}}{P_{cm}} = \left[\sum_{i}^{N} x_i \left(\frac{T_{ci}^{1.25}}{P_{ci}^{0.5}}\right)\right]^2$$

$$T_{cm} = \left\{\frac{\left[\sum_i^N x_i \left(\frac{T_{ci}^{1.25}}{P_{ci}^{0.5}}\right)\right]}{\sum_i^N x_i \left(\frac{T_{ci}}{P_{ci}}\right)}\right\}^{\frac{2}{3}}$$

$$T_{CM} = \frac{T_{cm}}{\sum_i^N x_i \left(\frac{T_{ci}}{P_{ci}}\right)} \qquad (3.33)$$

T_{Cm}= Mixture Critical Temperature (K)

P_{Cm}= Mixture Critical Pressure (bar)

T_{CI}= Individual gas Critical Temperature (K)

P_{Ci}= Individual gas Critical Pressure (bar)

x_i= mol percent or percent by volume of each gas

3.4.3 Practical application given Data with program input/output

Estimate the sizing and performance requirements for a new compressor application in which 81.6m^3/day of a gas mixture is to be compressed in a 3-stage Reciprocating Compressor. The given data are from compressor specification manual:

- Initial Suction Pressure = 14.3 bar absolute
- Final Discharge Pressure = 150 bar absolute

Table 3.1: stage specification of compressor

	1st stage	2nd stage	3rd stage
Cylinder clearance (%)	15	15	15
Piston/Valve area ratio	10	10	10
Suction temperature (deg. C)	38	38	38

Design Requirements:

- Initial suction pressure drop = 1%
- Final pressure drop = 1%
- Number of cylinders per stage = 1
- Stroke length = 125mm
- Piston rod size = 12.5 mm
- Desired operating speed = 2000rpm

3.4.4 Selection And Construction Of Air Starting System Compressor

We select the Piston type, two Cylinders, three stage Air cooled Compressor AK-150CB[Figure 3.2 Third stage Air Compressor], serves to fill the Pressure Vessel with compressed air up to required pressure, the working pressure developed by the compressor reaches the value of 150bar; at the speed of the compressor crankshaft of 2000 rpm the compressor output constitutes 0.0567m^3/min. The compressor is installed on the Diesel Engine lower half from the right hand side the direction of locomotive movement; it is driven from the Engine drive shaft. Lubrication forced to the Compressor under pressure from Engine lubrication system after the oil filter.

The main parts and units of the compressor are case 7, first and second stage Cylinder 20, Third stage cylinder 13, first and second stage Cylinders differential Piston 21 with compression and oil control Rings, Piston 12 of third stage Cylinder with compression Rings and oil control Rings, eccentric shaft 9, connecting rod 6 and 10, Inlet (4, 16 and 23), Delivery (2, 15 and 16) valves.

Case 7 is the cast of aluminum alloy and consists of two halves Interconnected by bolts. The joint surface of the case are sealed by means of a silk cord impregnate with sealing compound.

The case is provided two holes arranged at 90^0 to each other. The holes terminate in flanges attached to which with studs are the cylinders.

The Cylinders are a special castings made of an aluminum alloy provided with machined holes press fitted into which steel liners. On the outer surface of the cylinder concentrically arranged are cooling ribs coast integral with the cylinders. The side surface of the cylinders are provided with two bosses having threaded holes for Inlet and Delivery Valves.

Provided on the upper flange of the first and second stage cylinders are eight holes interred for screws attaching the head, and a groove for sealing the cylinder-to-head joint surface. The cylinder head mounts Inlet 23 and Delivery 2 valves, and pipe union 1 for supply of air to the compressor. The head, similarly to the cylinders, has ribs for cooling.

The pistons are of the stage type, made of an aluminum alloy. Piston 21 of first and second stage cylinder at its upper part has four grooves for the compressor rings that seal the first stage cylinder, and at the lower part four grooves for the compression rings that seal the second stage cylinder; the lower most groove is indented for the oil control Ring. At its the upper parts piston 12 of the third stage cylinder has five compression Ring and one control Ring, and at its lower part seven compression Rings and one oil control Rings are made of the cast iron.

The lower parts of both pistons are provided with a hole having press-fitted bronze bushings that serve as bearings for piston pins. Both Piston made hollow with view to reduced their mass. The crank gear of the compressor consists of eccentric shaft 9, webs with counterweight, Connecting rod 6, 10 and a race. The Eccentric Shaft rotates in two ball bearings installed in the crankcase. The front end of shaft has splines for connection with the drive shaft. The Connecting Rod big end together with steel race press-fitted into big end rotates on needle Bearing 8. The big end of articulated connecting rod 10 slides over the outer surface of the needle Bearing race. The Connecting Rod small ends are connected to the Piston by Piston Pins, inserted into the piston pin ends face are end caps made of an aluminum alloy; they protected the cylinder faces against scoring by the Piston Pins.

3.4.5 Operation of Third Stage Air Compressor

Air in the compressor is compressed in each of its three stages interconnected in series and construction ally made in two cylinder: first and second stage in one Cylinder and third stage other Cylinder. During rotation of Eccentric Shaft "13" (Figure 3.3 Third stage Air Compressor Operation Diagram[4]) of the Compressor, Pistons "15"and"10" perform reciprocating movements. When the Piston "15" goes down (from TDC), the Cylinder space in the first stage Cylinder is increased, and rarefaction is created above the Piston. Under the action of rarefaction Inlet Valve "2" gets opened and the first stage of the Compressor is filled with a charge of cleaned air which is supplied from air Cleaner head through Pipe line.

When the Piston moves upwards (to TDC), the Inlet Valve gets closed and the air in the first stage Cylinder is Compressed. When the air pressure in the first stage Cylinder reaches a value which exceeds the force of the Delivery Valve spring, the Delivery Valve gets opened and the air is supplied into the second stage Cylinder space via Pipe line "17" though the second stage Inlet valve. With the next movement the Piston "15" down the process of compression of air in the second stage Cylinder of the Compressor takes place. Under the pressure of Compressed air Delivery Valve "5" gets opened and air is supplied into the third stage Cylinder space via Pipe line "6" though Inlet Valve "7".

When the third stage Cylinder Piston moves upward, Inlet Valve "7" gets closed and a process of compression of air in the Compressor third stage Cylinder takes place. The compression process contuse until the air pressure overcomes the pressure force of the spring of Delivery valve "9". Upon opening of Valve "9" air via the Pipe line gets into Oil and Moisture separators where it is cleaned of oil and moisture, and further runs into Pressure Vessel though air pressure Relief Valve.

3.5 Compressed air system pipe design

Marley MDPE pipe systems are ideal for the transmission of compressed air both in the high and low pressure range. The use of compressible fluids in PE pipes requires a number of specific design considerations as distinct from the techniques adopted for fluids such as compressed air. In particular: Safety factor for different gases should be considered in any design[9].

- ✓ Natural gas 2.0
- ✓ Compressed air 2.0.

- ❖ Compressed air may be at a higher temperature than the ambient air and PE pipes require temperature re-rating accordingly. For air cooled compressors the air temperature averages 15ºC above the surrounding air temperature.
- ❖ For underground applications the surrounding temperature may reach 30ºC and the pipe properties require adjustment accordingly.
- ❖ High pressure lines must be protected from damage, especially in exposed installations.
- ❖ Valve closing speed must be reduced to prevent a buildup of pressure waves in the compressible gas flow.
- ❖ The compressed air must be dry and free from liquid contamination(by oil and moisture separator, sediment bowl) which may cause stress cracking in the PE pipe walls.
- ❖ MDPE pipes should not be connected directly to compressor outlets or air receivers. A connector of metal pipe should be inserted between the air receiver and the start of the PE pipe to allow for cooling of the compressed air.
- ❖ Dry gases and gas/solids mixtures may generate static electrical charges and these must be dissipated to prevent the possibility of explosion.
- ❖ Compressed air must be dry and filters installed in the line to prevent stress cracking in the PE pipe.
- ❖ The fitting systems used must be pressure compatible with the pipe and pressure compatible with the pipe and where meta; couplings and shouldered ends are used, the maximum pressure is limited to 200bar.

3.5.1 Compressed Air Pipe Design Sequence

- Locate and Identify each process, starting Engine, or to start the Engine using compressed air. This is known as the total connected load. These elements should be located on a plan, and a complete list should be made to simplify record keeping.
- Determine volume of air used to start the engine.
- Determine pressure range required starting the engine.
- Determine conditioning requirements for each item, such as allowable moisture content, particulate size, and oil content.
- Establish how much time the individual process will be in actual use for a specific period of time. This is referred to as the duty cycle. This information will help determine the simultaneous use factor by eliminating some locations during periods of use at other locations.
- Establish the maximum number of locations that may be used simultaneously on each branch, main, and for the project as a whole. This is known as the use factor.
- Establish the extent of allowable leakage.
- Establish any allowance for future expansion.
- Make a preliminary piping layout, and assign preliminary pressure drop.
- Select the air compressor type, conditioning equipment, equipment and air inlet locations making sure that consistent standard unit is used for both the system and compressor capacity rating.
- Produce a final piping layout, and size the piping network.

3.5.2 Compressed Air Formula

It is usual to find the inside Diameter of the pipe by using formulas such as those shown below. The formulas used are generally for approximation purposes only, guessing that the temperature of the compressed air corresponds roughly to the induction temperature. You will obtain an acceptable approximation through the following equation.

$$Di = \sqrt[5]{\frac{450 * l * \frac{dV^{1.85}}{dt}}{\Delta p . p}} \quad \text{.......}(3.34)$$

Where: Δp = pressure decrease (bar)

p = working pressure (bar)

V = volumetric flow rate (l/s)

$\frac{dV}{dt}$ = atmosphere (l/s)

l = pipe length (m) and D_i = inside diameter (mm).

3.5.3 Flow characteristics

The pressure losses for fluid flow through pipes are strongly dependent on the flow characteristics. The main defining parameter is Reynolds number, which for flow in circular pipes is defined as:

$$R_e = \frac{\rho w D}{\mu} \qquad (3.35)$$

where: w: is the flow velocity

D: the pipe diameter, and μ is the dynamic viscosity of the fluid.

The three regimes of fluid flow are laminar, transition, and turbulent flow. Low Reynolds numbers gives laminar flow. For flow in circular pipes the accepted value for transition is Re = 2300 (White [1998]). Fully turbulent flow is obtained for values of Reynolds number in the order of 10^3 -10^4 and higher.

The piping of the compressed air engine starting system usually has diameters < 100 [mm]. For the flow velocities encountered in the piping, this gives Reynolds numbers above 10^5. That is, in the turbulent range. Another important characteristic of the flow is its compressibility. When a fluid flows at velocities comparable to its speed of sound, the effects of density changes become significant, and the flow is termed compressible. The speed of sound for a fluid is given by:-

$$V_s = \sqrt{kRT} \qquad (3.36)$$

The condition for treating a gas as incompressible is that the Mach number, defined as the ratio between the fluid velocity, and the speed of sound of the gas is much lower than one (White [1998]). The speed of sound for gases is low, giving strong compressible effects. The flow velocities in the pipe of the starting system will exceed the speed of sound of air. Hence, the theory of compressible flow must be utilized.

3.5.4 Pressure drop equations

For calculation of pressure drop in straight pipes, usually no heat loss (adiabatic) or constant entropy (isentropic) conditions is assumed. For long pipes the flow conditions more closely approximates an isentropic flow (Liu [2003]).

An equation for the pressure drop for isentropic compressible flow in pipes of constant area, can be developed from

- ❖ Conservation of mass.
- ❖ Conservation of momentum (Navier-Stokes equation).
- ❖ Conservation of energy for the flow (first law of thermodynamics) and The ideal gas law.

In addition it is assumed that the wall shear is correlated by the dimensionless Darcy friction. factor "f" such that the wall shear can found from:

$$T_w = \frac{f\rho w^2}{8} \quad (3.37)$$

Using these five equations, we can derivative the isentropic flow equation. This is done in many books, e.g. Liu [2003] and White [1998], to arrive at

$$P_2 = \sqrt{P_1^2 - P_1\rho_1 w_1^2[2\ln\left(\frac{w_1}{w_2}\right) + \frac{fL_e}{D}]} \quad (3.38)$$

where Le is the length of the pipe, and the indices "1" and "2" relates to the start and the end of the pipe, respectively. Eq. 3.38 requires knowledge of the flow velocity at both ends of the pipe. Typically the change in velocity over the pipe is small (Beater [2007]). If no change of velocity over the pipe is assumed, then the logarithmic term disappear and the pressure at the end of the pipe can be found from:-

$$P_2 = \sqrt{P_1^2 - \frac{P_1\rho w^2 fL_e}{D}} \quad (3.39)$$

The flow velocity w, is found by dividing the mass flow by the cross-sectional area of the pipe and the density. The density ρ is given by the air cylinder mass divided by the air cylinder volume. We get

$$P_2 = \sqrt{P_1^2 - \frac{16P_1V\ \ ^2 fL_e}{\Pi^2 mD^5}} \quad (3.40)$$

3.5.5 Friction factor

The Darcy-friction factor "f" is strongly dependent on the Reynolds number. Many different equations for this factor, based on empirical data or theory, exists for the different flow regimes. A schematic representation of the friction factor as a function of Reynolds number and relative pipe roughness, ε/D is given in the so-called Moody diagram, presented in a simplified version. As the diagram shows, the dependence on pipe roughness disappears in the laminar flow regime. In this regime the diagram is based on the Poiseulle equation:

$$f = \frac{64}{R_E} \qquad (3.41)$$

The Darcy friction factor as a function of Reynolds number and the relative pipe roughness. In the transition and turbulent flow regiments the diagram is based on an iteration equation developed by Colebrook.

$$\frac{1}{\sqrt{f}} = -2\log_{10}^{[\frac{\frac{\varepsilon}{D}}{3.7}+\frac{2.51}{R_e\sqrt{f}}]} \qquad (3.42)$$

$$\frac{1}{\sqrt{f}} = -1.8\log_{10}^{[(\frac{\frac{\varepsilon}{D}}{3.7})^{1.1}+\frac{2.51}{R_e}]} \qquad (3.43)$$

According to White [1998], the Haaland equation varies with less than two percent from the Colebrook equation. Many other alternative formulas for friction factors for different flow conditions have been proposed. The choice of method for finding the friction factor depends on the required accuracy and the flow characteristics. We assume that the Haaland equation gives a sufficiently accurate estimate of the friction factor. Alternatively we could use interpolation in the Moody diagram.

3.6 Design of Pressure Relief Valve

3.6.1 Introduction

The function of a Pressure Relief Valve is to protect pressure vessels, piping systems, and other equipment from pressures exceeding their design pressure by more than a fixed predetermined amount. The permissible amount of overpressure is covered by various codes and is a function of the type of equipment and the conditions causing the overpressure.

It is not the purpose of a Pressure Relief Valve to control or regulate the pressure in the vessel or system that the valve protects, and it does not take the place of a control or regulating valve. The aim of safety systems in compressed air Engine starting system is to prevent damage of equipment, avoid injury to personnel and to eliminate any risks of compromising the welfare of the community at large and the environment. Proper sizing, selection, manufacture, assembly, test, installation, and maintenance of a Pressure Relief Valve are critical to obtaining maximum protection.

A Pressure Relief Valve must be capable of operating at all times, especially during a period of system failure or the system no need of compressed air; therefore the sole source of power for the pressure relief valve is the process compressed air.

The Pressure Relief Valve must open at a predetermined set pressure, flow rated capacity at a specified overpressure, and close when the system pressure has returned to a safe level. Pressure relief valves must be designed with materials compatible with many process fluids from simple air to the most corrosive media.

The standard design safety Relief Valve is spring loaded with an adjusting ring for obtaining the proper blow down and is available with many optional accessories and design features. Pilot-operated valves are available with the set pressure and blow down control located in a separate control pilot. This type of valve uses the line pressure through the control pilot to the membrane in the main Relief Valve and thereby maintains a high degree of tightness, especially as the set pressure is being approached. Another feature of the pilot-operated valve is that it will permit a blow down as low as 2%. The disadvantage of this type of valve is its weakness to contamination from foreign matter in the Compressed air stream.

3.6.2 Codes And Standards Of Pressure Relief Valve

Since pressure Relief Valves are safety devices, there are many Codes and Standards in place to control their design and application. The purpose of Codes and Standards which govern the design and use of pressure relief valves.

3.6.3 Spring Loaded Design

The basic spring loaded pressure relief valve has been developed to meet the need for a simple, reliable, system actuated device to provide overpressure protection. The valve consists of a Valve inlet or nozzle mounted on the pressurized system, a disc held against the nozzle to prevent flow under normal system operating conditions, a spring to hold the membrane closed, and a body to contain the operating elements. The spring load is adjustable to vary the pressure at which the Valve will open.

3.6.4 Back Pressure Considerations

Pressure Relief Valves on clean non-toxic, non-corrosive systems may be vented directly to atmosphere because there is Oil and Moisture separator before Pressure Relief Valve. Pressure Relief Valves on corrosive, toxic or valuable recoverable compressed air are vented into closed systems. Valves that vent to the atmosphere, either directly or through short vent stacks, are not subjected to elevated back pressure conditions. For Valves installed in a closed system, or when a long vent pipe is used, there is a possibility of developing high back pressure. The back pressure on a Pressure Relief valve must always be evaluated and its effect on valve performance and relieving capacity must be considered.

3.6.5 Pressure Relief Valve Design Consideration

1. **Cause of overpressure**

Overpressures that occur in compressed air Engine starting system and refineries have to be reviewed and studied, it is important in preliminary steps of Pressure Relief system design. It helps the designer to understand the cause of overpressure and to minimize the effect. Overpressure is the result of an unbalance or disruption of the normal flows of material and energy that causes the material or energy, or both, to build up in some part of the system.

- Blocked Discharge
- Fire Exposure
- Check Valve Failure and Thermal Expansion and Utility Failure

2. Application of Codes, Standard, and Guidelines

Designed pressure relieving devices should be certified and approved under Code,

- ✓ ASME- Boiler and Pressure Vessel Code Section I, Power Boilers, and Section
- ✓ VIII, Pressure Vessels.
- ✓ ASME- Performance Test Code PTC-25, Safety and Relief Valves.

3.6.6 Design Procedure pressure Relief Valve

General procedure in the design of protection against overpressure as below:

- ❖ **Consideration of contingencies**: All condition which will result in process equipment overpressure is considered; the resulting overpressure is evaluated and the appropriately increased design pressure; and each possibility should be analyzed and the relief flow determined for the worse case.
- ❖ **Selection of pressure relief device**: The appropriate type for pressure relief device for each item of equipment should be proper selection based on the service required.
- ❖ **Pressure relief device specification:** Standard calculation procedures for pressure relief device should be applied to determine the size of the specific pressure relief device.
- ❖ **Pressure relief device installation**: Installation of the pressure relief valve should be at the correct location, used the correct size of inlet and outlet piping.

3.6.7 Numerical Analysis Of Pressure Relief Valve

Gas Sizing 10% Overpressure (kg/hr)

The following formula is used for sizing valves for compressed air when required flow is expressed as a mass flow rate, kilograms per hour. Correction factors are included to account for the effects of back pressure, compressibility and subcritical flow conditions.

$$A = \frac{13160W\sqrt{TZ}}{CKP_1K_b\sqrt{M}} \quad \text{..........(3.44)}$$

$$A = \frac{13160*2475.2\sqrt{368.15*1}}{315*0.975*16601*1\sqrt{29.97}} = 20mm^2$$

Where:

A = Minimum required effective discharge area, square millimeters(mm^2).

C = Coefficient determined from an expression of the ratio of specific heats of the gas at standard conditions (see Table 3.2). Use C = 315 if value is unknown.

K = Effective coefficient of discharge, Use K = 0.975

K_b = Capacity correction factor due to back pressure. For standard valves with superimposed (constant) back pressure exceeding critical see Table 3.3. For bellows or Series BP valves with superimposed or variable back pressure see Figure 3.10.

M = Molecular weight of the gas obtained from standard tables =28.97kg.

P_1 = Relieving pressure, kilo Pascals absolute. This is the set pressure (kPa) + overpressure (kPa) + atmospheric pressure (kPa)=15000kpa+1500kpa+101kpa=16601kpa.

T=Absolute temperature of the fluid at the valve inlet, K degree(°C + 273)=95+273.15=368.15k.

W = Required relieving capacity, kilograms per hour=1980m^3/hr*1.25kg/m^3=2475.2kg/hr.

Z = Compressibility factor (see Figure 3.11), Use Z = 1.0 if value is unknown.

Gas Sizing 10% Overpressure (m^3/min)

The following formula is used for sizing valves for gases when required flow is expressed as a volumetric flow rate in m^3/min. Correction factors are included to account for the effects of back pressure, compressibility and subcritical flow.

$$A = \frac{179400*Q\sqrt{TGZ}}{C*K*P_1*K_b} \qquad (3.45)$$

$$A = \frac{179400*33\sqrt{368*1*1}}{356*0.95*16601*1} = 20mm^2$$

Where:

A = Minimum required effective discharge area, square millimeters(mm^2).

C = Coefficient determined from an expression of the ratio of specific heats of the gas at standard conditions (see Table 3.2), Use C = 315 if value is unknown.

K = Effective coefficient of discharge, Use K = 0.975

G = Specific gravity of the gas.

K_b = Capacity correction factor due to back pressure. Standard valves with superimposed (constant) back pressure exceeding critical see Table 3.3. For bellows or Series BP valves with superimposed or variable back pressure, see Figure 3.10 .

P_1 = Relieving pressure, kiloPascals absolute. This is the set pressure (kPa) + overpressure (kPa) + atmospheric pressure (kPa)=16601kpa.

T=Absolute temperature of the fluid at the inlet, degrees Kelvin (°C + 273)=95+273.15=368.15k.

Q = Required relieving capacity, standard cubic meters per minute (m^3/min)=33m^3/min.

Z = Compressibility factor (see Figure 3.11). Use Z = 1.0 if value is unknown.

Compressibility Factor, Z

The gas formulae of this handbook are based on perfect gas laws. Many real gases, however deviate from a perfect gas. The compressibility factor Z is used to compensate for the deviations of real gases from the ideal gas. The compressibility factor may be determined from Figure 3.5 by first calculating the reduced pressure and the reduced temperature of the gas. The reduced temperature is equal to the ratio of the actual absolute inlet gas temperature to the absolute critical temperature of the gas.

$T_R = \frac{T}{T_c}$..(3.46)

$$T_R = \frac{T}{T_c} = \frac{368k}{132.6} = 2.775$$

Where:

T_R = Reduced temperature

T = Inlet fluid temperature, (°C + 273.15)=95+273=368.15k

T_C = Critical temperature, (°C + 273)=132.6k

The reduced pressure is equal to the ratio of the actual absolute inlet pressure to the critical pressure of the gas.

$P_R = \frac{P}{P_C}$..(3.47)

$$P_R = \frac{P}{P_C} = \frac{16601}{3770} = 4.403$$

Where:

P_R = Reduced pressure

P = Relieving pressure (set pressure + overpressure + atmospheric pressure), (kPa)=16601kpa

P_C = Critical pressure (kPa) =3770kpa

Enter the chart at the value of reduced pressure, move vertically to the appropriate line of constant reduced temperature. From this point, move horizontally to the left to read the value of Z. In the event the compressibility factor for a gas or vapor cannot be determined, a conservative value of Z = 1 is commonly used.

Ratio of specific heats, k, and Coefficient, C

The following formula equates the ratio of specific heats to the coefficient, C, used in sizing methods for gases. Figure 3.8 and Table 3.4 provide the calculated solution to this formula.

$$C = 520\sqrt{(K)[\frac{2}{K+1}]^{\frac{(K+1)}{(K-1)}}} \qquad (3.48)$$

Where: k = Ratio of specific heats

Noise Level Calculations

The following formulae are used for calculating noise level of gases as a result of the discharge of a pressure relief valve. The expressed formulae are derived from API Recommended Practice 521. Table 3.4 lists relative noise levels.

$$L_{100} = L + 10LOG_{10}(0.29354WK\frac{T}{M}) \qquad (3.49)$$

Where:

L_{100} = Sound level at 100 feet from the point of discharge in decibels.

L = Noise intensity measured as the sound pressure level at 30.48m from the discharge. Reference Figure 3.10.

W = Maximum relieving capacity, 0.1260gram/sec.

k = Ratio of specific heats of the fluid. Reference Table 3.2 (For Gas, k = 1.3 if unknown.)

T = Absolute temperature of the fluid at the valve inlet, (^{0}C + 273).

M= Molecular weight of the gas obtained from standard tables or Table 3.2.

When the noise level is required at a distance of other than 30.48m, the following equation shall be used:

$$L_P = L_{100} - 20LOG_{10}(\frac{r}{100}) \qquad (3.50)$$

Where:

L_P = Sound level at a distance, "r" from the point of discharge in decibels.

r = Distance from the point of discharge.

Reaction Forces spring loaded Pressure Relief Valve

The following formulae are used for the calculation of reaction forces for a pressure relief valve discharging gas, gas directly to atmosphere without discharge piping. The expressed formulae produce results consistent with the API Recommended Practice 520 for gas:

$$F = \frac{CKP_1\sqrt{\frac{K}{K+1}}}{332.7} + F_g \qquad (3.51)$$

$$F_g = \left(\frac{KAP_1K_r}{1.383A_O} - P_a\right)A_O \qquad (3.52)$$

If F_g is less than or equal to 0.0 use Fg = 0.0.

Where:

F =Total reaction force at the point of discharge to atmosphere.

F_g = Component of reaction force due to static pressure at the valve outlet for gas applications.

F_s= Component of reaction force due to static pressure at the valve outlet for gas applications.

C = Coefficient determined from an expression of the ratio of specific heats of the gas at standard conditions. (Reference Table 3.2).

K = Effective coefficient of discharge. K = 0.975.

A = Effective discharge area, square millimeter.

A_o= Outlet cross-sectional area, square millimeter.

P_1 = Relieving pressure(kpa). This is set pressure (kpa) + overpressure (kpa) + atmospheric pressure (kpa).

P_a = Atmospheric pressure (kpa).

k = Ratio of specific heats of the fluid. Reference Table 3.2.

K_n= High pressure air correction factor.

K_r = Correction for ratios of specific heats of other than 1.4 (Reference Table 3.5).

3.6.8 Function and Construction of Pressure Relief Valve

We are select spring loaded and pilot pressure operated Valve according to the above specification and the function of Pressure Relief Valve

- Transfer the Compressor into idling when air pressure in the system reaches the value of 135-165bar.
- Transfer the Compressor into Pressure Vessel filling mode, when the air pressure in the lower Pressure Vessel (working Pressure Vessel) gets reduced approximately to 120bar.

When the Compressor is idling, the pressure in the delivery line reaches a value of in excess of 150bar. The main parts of air pressure Relief Valves are[Figure 3.4 Air pressure Relief valve[11]] housing 6, disengagement valve 5, plate-type arrangement valve 13, membrane 17, non return valve 2, reducing valve 3, and protective case 10 with rubber valve 8 for release of air into atmosphere from the space inside the case.

3.6.9 Operation of pressure Relief Valve

From[Figure 3.3 Air pressure Relief Valve[11]] when the Compressor operates for filling the Cylinders, Non-return Valve "2" is open, and air freely flows into the main line via the main duct of the air pressure Relief Valve, in this case the Valve "5" with its needle closes the duct under the Valve, and the above-Valve space is connected with the atmosphere through an orifice in valve "13". As the pressure rises in the Pressure Vessel, Membrane "17" bucking upwards presses upon Dowel "14" of Engagement Valve "13" and closes it (when the air pressure reaches a value of approximately 140bar). In this case the duct under Valves is disconnected from the atmosphere.

When the air pressure in the lower Pressure Vessel(Working Pressure Vessel) rises up to the value of 150bar, disengagement needle valve "5" gets opened and air from the main line running from the Compressor will be expelled into the atmosphere though Pressure Relief Valve "3" and Rubber Valve "8".

The Compressor started idling at a back-pressure of the compressed air of 10-15bar which is obtained by adjusting the spring of Pressure Relief Valve "3". In this case Valve "5" is retained in the open position by the air pressure of less than 15bar, and the Valve working surface considerably exceeds that the needle. Upon opening Valve "3" and "5" the pressure in the main line before Valve "2" starts dropping, and when it becomes equal to that after the Valve, the latter will get closed under the action of the spring. In this case the pressure in the main line will be dropping to a Value of 15bar at the Compressor starting idling.

If the air pressure in the lower Pressure Vessel(working Pressure Vessel) becomes less than 12bar, the springs will buckle Membrane "17" downwards and open Engagement Valve "13"; the delivery line will be connected with the atmosphere through an orifice in adjusting nut "12". As a result, the air pressure on Disengagement Valve "5" sharply drops and the Valve gets closed. In this case the compressed air deliver from the Compressor overcoming the resistance of Non-Return Valve "2", will be supplied for charging the Pressure Vessel. The air pressure Relief valve is installed in the sealed case which protects the unit against dust and dirt. The case has a rubber Valve '8' for the release of air when the Compressor is idling, and plug "11" for scavenging the valves of the air pressure Relief Valve when their normal operation is disturbed.

3.7 Air Pressure Reducer

We are select pressure Reducing Valve according to specification. Air pressure Reducing Valve with a filter serves to reduced the air pressure from 150bar to 70bar supplied to the consumers from the Pressure Vessel cut into the system before the reducer. It is located in front of the air Pressure Vessel. The Pressure Reducer consists [Figure 3.5 Air pressure Reducing valve[11]] of body 5, high pressure valve 4, push rod 2, membrane 6, piston 1, spring 7 located in the piston body, and safety valve 8. The pressure of the air leaving the Pressure Reducer is automatically regulated due to closing the nozzle by high-pressure valve "4". Air from upper Pressure-Vessel(Reserved Pressure Vessel) flows through Filter "3" into space under Valve "4", and having passed through the nozzle and grooves of Push Rod "2", exerts pressure on Membrane 6. Under the pressure the Membrane starts buckling and compressed spring "7" through Piston "1".Valve "4" together with Push Rod "2" will be shifted downwards, thus reducing the nozzle cross-sectional area. Due to this fact the air pressure at the outlet Pressure Reducer will be maintained the limits of 70 bar.

when the air pressure Reduced, under the action of Piston "1" and spring "7" Membrane "6" buckles in the reverse direction, Push Rod "2" and Valve "4" are shifted upward, thus increasing the nozzle cross-sectional area, so the pressure at the Pressure Reducer outlet is restored. Safety Valve "8" serves for the release of air into the atmosphere when the Pressure Reducer is faulty. Felt filter "3" installed before the Pressure Reducer protects the Pressure Reducer valves against clogging. The Pipe Union with a 2-mm throttling orifice installing in the air outlet pipe reduces the pressure drop, thus improving the operating conditions of the Pressure Reducing valves.

3.8 Electro pneumatic Valve

We are select AK-48 Electro pneumatic valve from standard, Electro Pneumatic Valve serves to deliver air from the air system to the actuating mechanisms. It consists of the following main parts[Figure 3.6 Electro pneumatic valve AK-48[4]]: Pipe Unions 1 and 4, Inlet Valve 3, Outlet Vale 5, Piston 6, Servo Valve 13, Solenoid Switch 10, and Body 16. Pipe Union 1 is connected with a union nut to the pipe of the air system, and pipe union 4, to the pipe running to the actuating mechanism. The Electro pneumatic operate in a pressure at the system not less than 25bar as follows. When the Lever "9" depressed or when the solenoid switch is engaged, Servo Valve "13" goes down, opens the orifice of Inlet Valve "3", and with its ball ring closes the orifice which connects the under the valve space to the atmosphere. Air from the space of Pipe Union "1" passes through the inner holes and holes in Piston "6", and further gets into space under the valve. In this case under the action of the air pressure the piston goes up, simultaneously forcing Inlet Valve "3" to move until it is fully opened, and outlet Valve "5" until it is closed. Thus, the air feed into the space of Pipe Union "1" gets there from into the space of Pipe Union "4", and further to actuating mechanism.

When the Lever "9" is not longer depressed or the magnet of the Electro pneumatic Valve cut off, Servo Valve "13" will be returned to initial positions under the action of Spring "12". The orifice of Inlet Valve "3" gets closed, and air from the space under the Valve is expelled in to the atmosphere. In this case the pressure exerted on Piston "6" drops, and the Piston under the action of Valve Spring "2" goes down, carrying away Inlet Valve "3" until it is closed, and Outlet Valve "5" until it is opened.

CHAPTER FOUR:SELECTION AND DESIGN

4.1 Air Dryer(Filter) Selection

Just as the degree of contamination in compressed air varies, so do the requirements for purity of the system and at various points of use. These requirements must be established prior to the selection of filters in the compressed air Engine starting system. Filters should be selected by their ability to meet established design criteria. The manufacturer, who is the most knowledgeable about specific conditions, should be consulted as part of the selection process. The following items must be considered[17]:

- ✓ Maximum flow rate expected.
- ✓ Desired pressure drop across Filter.
- ✓ Temperature of compressed air.
- ✓ Contaminants to be removed (requirement for filter type and housing material).
- ✓ Pressure rating.
- ✓ Is drain trap required? (automatic type preferable).
- ✓ Filter efficiency.

4.1.1 Filter Selection Procedure

- ❖ Calculate the maximum flow rate of air expected. This figure should be in the same units as the manufacturer’s rating units.
- ❖ Determine the lowest pressure that the filter can be expected to operate with during operation of locomotive compressed air Engine starting system. This pressure must be used in conjunction with the pressure lost in other system equipment.
- ❖ Based on the filter’s position in the system, determine what the contaminant or contaminants to be removed will be.
- ❖ Determine the maximum pressure drop across the filter. Manufacturers often use such values as ‘‘wetted pressure drop 'and ‘‘dry pressure drop’’ that do not take into consideration dirty elements. The average pressure drop ranges between 4 to 11 N/mm^2.
- ❖ Determine whether monitoring of the filter element for replacement (such as watching for a color change) be required.
- ❖ Select the appropriate filter from a manufacturer’s catalog.

The two most important factors to consider when selecting a filter are effectiveness and pressure drop. Do not specify a filter that will produce cleaner air than actually necessary. If one station or process requires a purity much higher than all other points, use a point-of-use filter for only that area and a less restricted filter for the main supply. It is possible for a filter to have the largest pressure drop of any equipment in the system. In general, a filter will produce a 21 to 69 kPa pressure drop when dirty. If the actual figure proves to be too high, it would be a good idea to oversize the filter to cut down the pressure drop, or to use two filters in parallel.

4.1.2 Oil And Moisture Separator

From the above procured and specification we are select Oil and Moisture separator. Oil and moisture separator is intended for trapping the vapors of the oil and moisture from the compressed air delivered from Compressor. It is installed between Compressor and Filter. The main parts of Oil and Moisture separator are[Figure 4.1 Oil and Moisture separator[4]] Housing 1, Filtering Element Filtering Gauze 6, and Gauze Filter 7.

Air is delivered into Housing "1" though a Pipe Union hole tangentially to the wall and at a certain angle of the steam downwards. Due to this fact a swirling movement is created which facilitates settling of oil and moisture vapors, condensed into drops, on the wall of housing "1", a Filtering Gauze, and Gauze Filter "7". Gauze "6" and Filter serve to damp oscillations of the sediment accumulated on the Housing bottom, and to prevent getting of sediment accumulated on the Housing bottom, and to prevent getting of sediment on to Filtering Element "2". Filtering Element "2" serves for conditional cleaning of air and consists of four felt rings separated with metal screens for strengthening.

When the air leaves the Oil and Moisture separator, it additionally passes though the Filter "3" installed before the air pressure Relief Valve. This Filter serves as an additional safety device for the Valves of the air pressure Relief Valve protecting them against clogging. To reduced carrying away of sediment at sharp drop of pressure, installed at the Oil and Moisture separator outlet is a Pipe Union with a throttling orifice, 1mm in diameter.

4.1.3 Sediment Bowl

We are select Sediment Bowl from catalog. Sediment Bowl[Figure 4.2 sediment bowl air bleed valve[4]] serves to trap moisture condensate and oil drops resaved together with the air from the compressor. It is installed on the left hand side before the Air Pressure Vessel. The Sediment Bowl is cylinder body 1 to whose end are welded bottom 7 and cover 5.

4.2 Design of Compressed Air Engine Starting System Pressure Vessels

4.2.1 Introduction

The term Pressure Vessel referred to those reservoirs or containers compressed air, which are subjected to internal or external pressures. The pressure vessels are used to store compressed air under pressure. The compressed air being stored may undergo a change of state inside the pressure vessels as in case of air starting system. Pressure vessels we use two vessel different pressure in compressed air engine starting system those are 120bar and 150bar. The material of a pressure vessel may be brittle such as cast iron, or ductile such as mild steel.

4.2.2 Construction Of Compressed Air Engine Starting System Pressure Vessels

- A solid wall vessel produced by forging or boring a solid rod of metal.
- A cylinder formed by bending a sheet of metal with longitudinal weld.
- Shrink fit construction in which, the vessel is built up of two or more concentric shells, each shell progressively shrunk on from inside outward.
- A vessel built up by wire winding around a central cylinder. The wire is wound under tension around a tension around a cylinder of about 6 to 10 mm thick.
- A Vessel built up by wrapping a series of sheets of relatively thin metal tightly round one another over a core tube, and holding each sheet with a longitudinal weld. Rings are inserted in the ends to hold inner shell round while subsequent layers are added. The liner cylinder generally up to 12mm thick, while the subsequent layers are up to 7mm thick.

5.2.3 Types Of Compressed Air Engine Starting System Pressure Vessels:

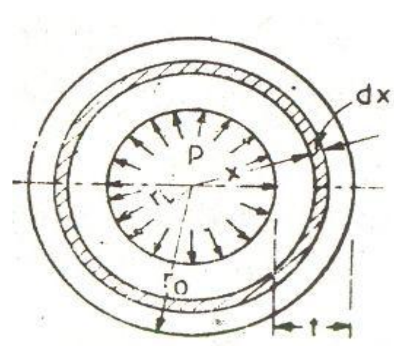


Figure 4.3 Solid Wall Vessel[12]

4.2.4 Factors Considered In Designing Pressure Vessels

- ✓ Dimensions-Diameter, length and their limitations.
- ✓ Operating conditions Pressure and temperature.
- ✓ Available materials and their physical properties and cost.
- ✓ Corrosive nature of reactants and products.
- ✓ Theories of failure.
- ✓ Method of Fabrication.
- ✓ Fatigue, Brittle failure and Creep and Economic consideration.

Design of Solid Walled Pressure Vessel

A solid wall vessel consists of a single cylindrical shell, with closed ends. Due to high internal pressure and large thickness the shell is considered as a 'thick' cylinder. In general, the physical criteria are governed by the ratio of diameter to wall thickness and the shell is designed as thick cylinder, if its wall thickness exceeds one-tenth of the inside diameter.

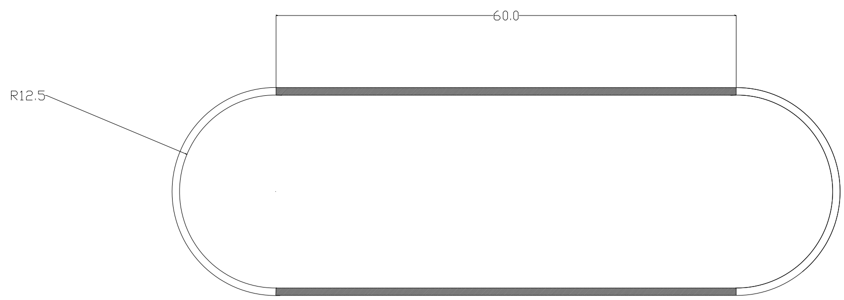


Figure 4.4 Drawing of solid wall pressure vessel

4.2.5 Input Data for design of pressure vessel(Reserved pressure vessel)

- Design pressure p = 150bar=15N/mm^2
- Design Temperature T = 20^{o}C
- Design Code - ASME Sec.VIII Division-1
- Inside radius of wheel R_i = 125mm
- Inside Diameter of vessel D_i =250mm
- Joint Efficiency J = 1
- Permissible stress for shell S = 1752bar=175.2N/mm^2
- Safety Factor F.S = 4 and Corrosion Allowance, C.A = 3.0mm

4.2.6 Design Parameters

The design of the upper(150bar=15N/mm^2) or reserved Pressure Vessel includes:

- Design of vessel thickness.
- Design of Dished ends thickness.
- Calculation of Hydrostatic Test Pressure.
- Calculation of Bursting Pressure.

a. Design of vessel thickness is calculated from the equation:

$t = R_i\left[\sqrt{\frac{(SJ+p)}{(SJ-p)}} - 1\right] + C.A$ ....(4.1)

$$t = 125mm\left[\sqrt{\frac{123 * 1 + 15}{123 * 1 - 15}} - 1\right] + 3mm = 21mm$$

Provided thickness of Liner (Core Tube) = 21mm

Thickness of each layer =7 mm

Number of Layers = 3

Where: t: Pressure Vessel thickness

p:design pressure

R_i :inside radius of shell

C.A: Corrosion Allowance

(b) Thickness of the dished end is given by:

$t_d = \frac{pR_i}{2SJ-0.2P} + C.A$...(4.2)

$$t_d = \frac{15 * 125}{2 * 123 * 1 - 0.2 * 15} + 3mm = 11mm$$

(c) Calculation of Hydrostatic Test Pressure

Hydrostatic Pressure is taken as 1.3 times design pressure.

$P_H = 1.3 * \text{Design pressure}$..(4.3)

$$P_H = 1.3 * 15\, {}^{N}/_{mm^2} = 19.5\, {}^{N}/_{mm^2}$$

- **Stress Developed during Hydrostatic Test**
  - ✓ **In Vessel**

$t = R_i \left[\sqrt{\frac{(SJ+p)}{(SJ-p)}} - 1\right]$ ……………………………………………………………(4.4)

$$20mm = 125[\sqrt{\frac{S*1+19.5}{S*1-19.5}} - 1]$$

$$S = 132.35\,\frac{N}{mm^2}$$

The stress developed (157.33N/mm2) is less than the allowable stress value (240.8N/mm2) which is 90% of the yield stress.

- ✓ **In Dished End**

The Stress developed inside the Dish is given by the equation,

$s_{Hd} = \frac{p_H*R_i+0.2p_H t}{2*t}$ ……………………………………………………………(4.5)

$$s_{Hd} = \frac{p_H*R_i+0.2p_H t}{2*t} = \frac{19.5*125+0.2*19.5*20}{2*20} = 62.89\,\frac{N}{mm^2}$$

The stress developed ($62.89N/mm^2$) is less than allowable stress value (240.8N/mm2 which is 90% of the yield stress).

(d) Calculation of Bursting Pressure

Ultimate tensile strength of material $=492N/mm^2$.

$k = \frac{\text{outer Diametre}}{\text{Inner Diametre}}$ ……………………………………………………………(4.6)

$$k = \frac{\text{outer Diametre}}{\text{Inner Diametre}} = \frac{250mm+2*20mm}{250mm} = 1.16$$

Bursting Pressure is calculated as per Lame's method

$P_B = U.T.S*\frac{K^2-1}{K^2+1}$ ……………………………………………………………(4.7)

$$P_B = U.T.S*\frac{K^2-1}{K^2+1} = 492*\frac{1.16^2-1}{1.16^2+1} = 72.49\,\frac{N}{mm^2}$$

Stress Developed During Bursting Test:

Stress developed inside the Dished ends is given by equation

$S_{Bd} = \frac{P_B*R_i+0.2P_B t}{2*t}$ ……………………………………………………………(4.8)

$$S_{Bd} = \frac{P_B*R_i+0.2P_B t}{2*t} = \frac{72.49*125+0.2*72.49*20}{2*20} = 233.78\,\frac{N}{mm^2}$$

Stress developed ($233.78N/mm^2$) is less than allowable stress value ($267.6N/mm^2$ which is 100% yield stress). Hence the Design is safe.

4.2.7 Design of Multilayer High Pressure Vessel(working pressure vessel)

The design of lower($12N/mm^2$=120bar) pressure vessel includes:

- Design of shell thickness.
- Design of Dished end thickness.
- Calculation of hydrostatic Test Pressure.
- Calculation of Bursting pressure.

Input data:

- ✓ Design pressure P =$12N/mm^2$=120bar
- ✓ Inside radius of shell, R_i =125mm
- ✓ Inside diameter of the shell, Di = 250mm
- ✓ Corrosion allowance, C.A =3.0mm
- ✓ Joint efficiency J =1.0
- ✓ Permissible stress for shell S =$116N/mm^2$=1160bar
- ✓ Thickness of Shell, t = ?

a. The thickness of the shell is calculated from the ASME modified membrane theory equation as

$$t = \frac{pR_i}{SJ-0.6P} + C.A \qquad (4.9)$$

$$t = \frac{pR_i}{SJ-0.6P} + C.A = \frac{12*125}{1*164*-0.6*12} + 3mm = 14mm$$

Provided thickness of Liner (Core Tube) = 12mm

Thickness of each layer = 7mm

Number of Layers = 2

b. Check for Minimum Shell Thickness:

The minimum shell thickness is required is checked by the equation as per APL-ASME code for welded pressure vessel is as

$$t = \frac{PD_i}{2*S*\eta-P} + C.A \qquad (4.10)$$

Where S = Design stress value for total thickness and is given by

$$S = \frac{S_c t_c + nS_1 t_1}{t_c + nt_1} \qquad (4.11)$$

S_c = Allowable stress at design temperature of liner = 164MPa.

S_1 = Allowable stress at design temperature of layers = 164MPa.

n = Number of layers.

t_c = Thickness of liner or core tube.

t_1 = Thickness of layer.

$S = \frac{S_c t_c + n S_1 t_1}{t_c + n t_1}$..(4.12)

$$S = \frac{S_c t_c + n S_1 t_1}{t_c + n t_1} = \frac{164 * 12 + 2 * 164 * 7}{12 + 2 * 7} = 164 \frac{N}{mm^2}$$

η= Weld efficiency is given by equation

$n = \frac{\eta_c t_c + (n - 1 + n_1) t_1}{t_c + n t_1}$...(4.13)

$$n = \frac{\eta_c t_c + (n - 1 + n_1) t_1}{t_c + n t_1} = \frac{1 * 12 + (2 - 1 + 0.8) * 7}{12 + 2 * 7} = 0.946$$

The minimum required shell thickness

$t = \frac{P D_i}{2 * S * \eta - P} + C.A$..(4.14)

$$t = \frac{P D_i}{2 * S * \eta - P} + C.A = \frac{12 * 250}{2 * 164 * 0.946 - 12} + 3mm = 14mm$$

Provided thickness (14mm) is more than the required , Hence the Design is safe.

4.3 compressed Air Distributor

Function of air Distributor to distribute compressed air in to each Engine Cylinder on the time according to the firing order of the Engine Cylinders.

4.3.1 Construction of Air Distributor

The air Distributor is fixed on the drive gear box of the fuel Injection pump and air Distributor. It consists of the air distributor Body and Hood, Drive Shaft, Adjusting Tooth Sleeve, Distributor Disk and Cover, Spring, spacer and Pin (Figure:4.5 Air Distributor[12]).

Body: There are 12 Air Ports and associated air passages and air outlets. Each of them connects to air starting Valve of associated Engine Cylinder though the compressed air pipe according to the firing order of Engine Cylinders.

Air Distributor hood: It is screwed on the Body and forms the enclosed inner chamber with the body. On the center of the Hood, there is an air Inlet connected with the Electro-pneumatic Valve of the air starting system though the compressed air Pipe.

Derive shaft: It is installed in the center of the Body. One of its ends is inserted in the notch on the center part of the Drive Gear of the fuel Injection Pump and air Distributor. The other end is splined and the adjusting toothed sleeve is mounted on it.

Adjusting toothed sleeve: On the sleeve, there are inner and outer spline which are used to adjust timing of the in taking of compressed air.

Air distributing Disk: It is sheathed on the outer splines of the adjusting Toothed Sleeve. One end of the spring presses against the air Distributing Disk, the other end, against the spacer which is held by the pin passing through the Drive Shaft. Under the tension of the spring, the air Distributing Disk is in closed contact with air Distributor Body. On the air Distributing Disk, there is an air distributing port which at whatever position can only align with one or two air ports on the body while the other air ports are covered by Distributing Disk. When the air Distributing Disk rotates counter clock wisely, the air distributing port will line up with the air ports one after another according to the firing order of the Engine Cylinders.

4.3.2 Operation of air Distributor

During opening the cock on the lower compressed air Pressure Vessel(working Pressure Vessel) and pressing the air Starting button, the compressed air will flow from the pipe on the air distributor hood into the inner chamber of the air distributor, and pass though air port(one or two) and air passage aligned with air distributing port on the air Distributing Disk, and then flow along the compressed air pipe to the air Starting Valve and enter in to associated Pressure Vessel, thus pushing the Piston moving down ward to rotate the Crank Shaft. The rotation of the Crank Shaft brings the Drive Shaft of the air Distributor[Figure:4.6 Scheme of the operation of the air distributor[12]] Adjusting Tooth Sleeve and air Distributing Disk to rotate counter clockwise by the drive mechanism. The air Distributing Port of the air Distributing Disk aligns with air ports and the passages one after another by every 60^0 of the Crank Shaft rotation. The compressed air is distributed to each Engine Cylinder according to the firing order, thus the Crank Shaft rotates continuously until the Engine is started. The time of the compressed air entering in to the Engine Cylinder (i.e the begging of the air distributing port aligning with the air ports) is 7^0+-3^0 TDC of the compression stroke. Because the center angle subtended by the air Distributing Port is 43^0+-0.5^0 , and that by the air port is 15^0, so, the rotated angle of the air Distributing Disk from opening to closing the air port is 58^0, corresponding to 116^0 rotating angle of the Crank Shaft. This time interval is the air delivery period.

If it wants a reliable and quick start of the Engine by the compressed air starting system, the key point is that the compressed air, besides its sufficient pressure, must inter into Engine Cylinders in proper time. The begging of air flowing into the associated Engine Cylinder should be just at the begging of the expansion stroke. In this case, it is stipulated that in assembly, the air Distributing port of the air Distributing Disk must be cut the air port of the associated Engine Cylinder as Piston is moved to the position at 7^0+-3^0 before TDC of the compression stroke(figure:4.6).

With the increase of the service time of the Engine, because of the wear of the gears and splines from the Crank Shaft to air Distributing Disk, the time of begging to deliver the compressed air into the Engine Cylinder will be delayed. But the Engine can still be started by the compressed air starting system. Usually it is not necessary to be adjusted in operation. If the hollow bolt is loosed, tighten it on time, or the pressure air will leak in starting.

4.4 Air Starting Valve

The function of Air Starting Valve to open and closed the compressed air passage to the Engine Cylinder when start the Engine with compressed air.

4.4.1 Construction of Air Starting Valve

There are 12 air Starting Valves. They air screwed in their installing holes in the Engine Cylinders respectively. It consists of Valve Body, Valve, Valve Spring, Adjusting Nut (Figure:4.7 Air Starting Valve[12]**)**.

4.4.2 Operation of air Starting Valve

When the pressure of the compressed air entering into the Valve Body exceeds the sum of the air pressure within the Engine Cylinder concerned and the Spring Tension, the compressed air will forced the Valve open, press the spring and enter into the Engine Cylinder. When the air Distributing Disk closes the passage to the Engine Cylinder or the gas in the Engine Cylinder is ignited, the pressure of the gas within the Engine Cylinder plus the Spring Tension will exceed the pressure of the compressed air within Valve Body, the air Spring Valve will be closed and thus prevents the gas in the Engine Cylinders from flowing to the air starting system.

4.5 Cost Analysis of compressed air Engine starting system .

No.	Unit	Capacity	model	Price(US$)
1.	Third stage air Compressor	900m^3/min	AK-150CB	10,000
2.	Pressure Vessel	15N/mm^2	-	600
3.	Pressure Vessel	12N/mm^2	-	535
4.	Special air Pipe	17N/mm^2	MDPE	100US$/m*25=2500
5.	Oil and Moisture separator	-	HK-D	820
6.	Sediment Bowl	-	DF-41	156.97
7.	Pressure Relief Valve	16.5N/mm^2	-	565.95
8.	Pressure Reducing Valve	12N/mm^2	-	590
9.	Electro pneumatic Valve	7N/mm^2	AK-48	470
10.	Mechanical air Filter	-	-	70
11.	Air Distributor with accessory	-	-	10,000
12.	Pressure Gauge	20N/mm^2	-	150
13.	Air Starting Valve	8N/mm^2	AK-72	120*12=2400
			Total	**28,291.95**

4.6 The system compatibility with Diesel-Electrical Locomotive Structure

Compressed air Engine Starting system is compatible with the structure of Diesel-Electrical Locomotive because of it is light weight, it needs less space, it operate less cost and it consume less energy from the Engine.

CHAPTER FIVE: RESULTS OF STATIC ANALYSIS

This Chapter contains plots of the stress contours obtained in the static Test Finite Element Analysis. The Engine components are modeled and assemble by CATIA software then import in to ANSYS workbench. The results shown are the Von-Misses stresses and the total displacements calculated in the analysis. Keep in mind that the unit of all the workbench ANSYS results now on wards are in SI units.

The mesh shown below is the Diesel Engine modeled as Beam and plate elements, the meshing size are automatic, program control, Corse and fast, specifically consisting of 15702 Elements and 30829 nodal points. The structure optimum Pressure has been calculated equal to 70bar.

ANSYS

Project

First Saved	Sunday, August 24, 2014
Last Saved	Sunday, August 24, 2014
Product Version	12.0.1 Release

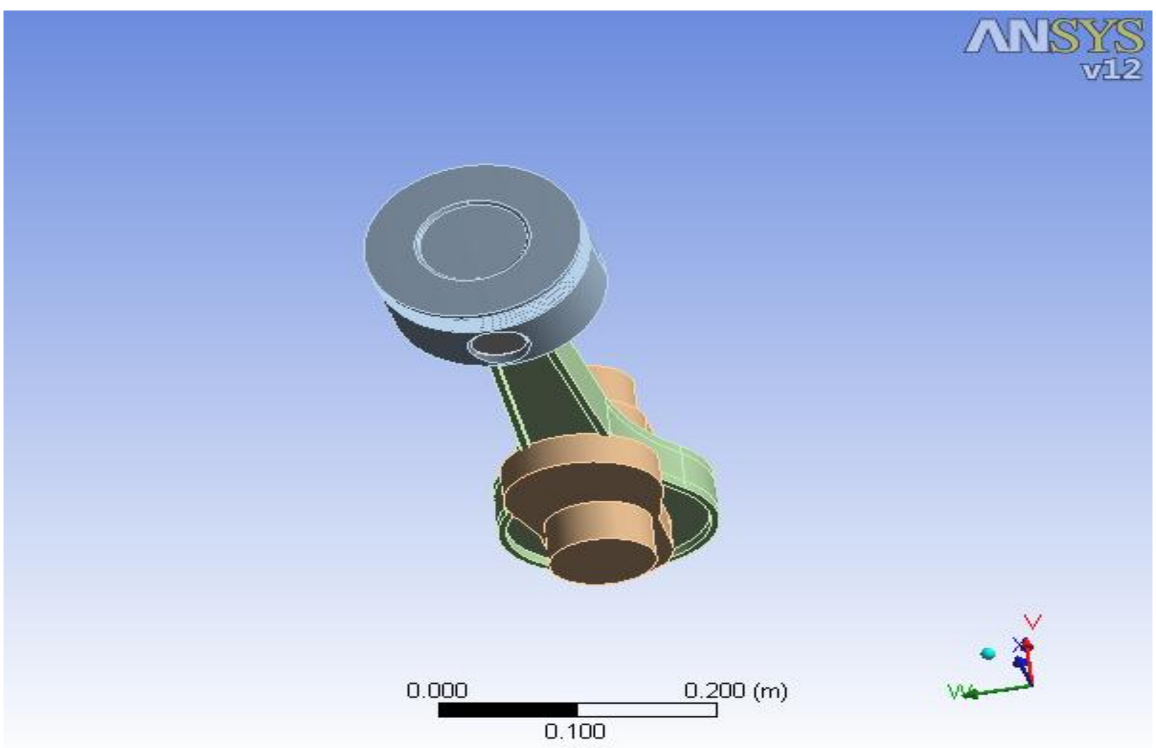


figure:5.1 FE Model of the Diesel Engine for Diesel-Electrical locomotive

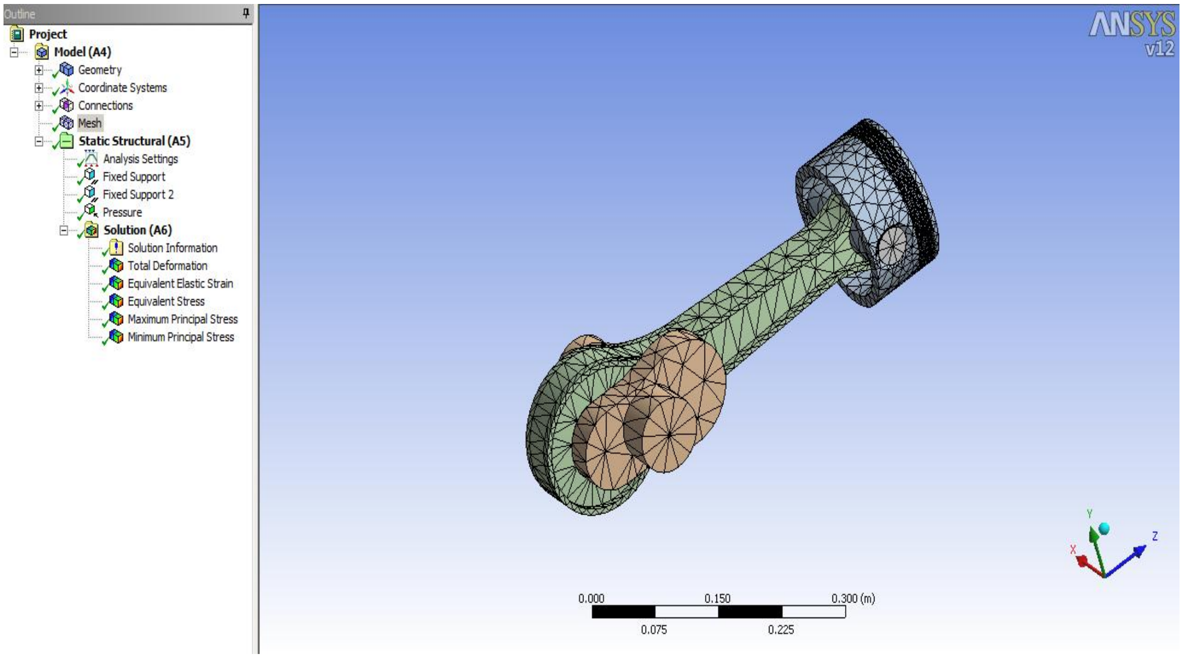


figure:5.2 meshing of the Diesel Engine for Diesel-Electrical locomotive

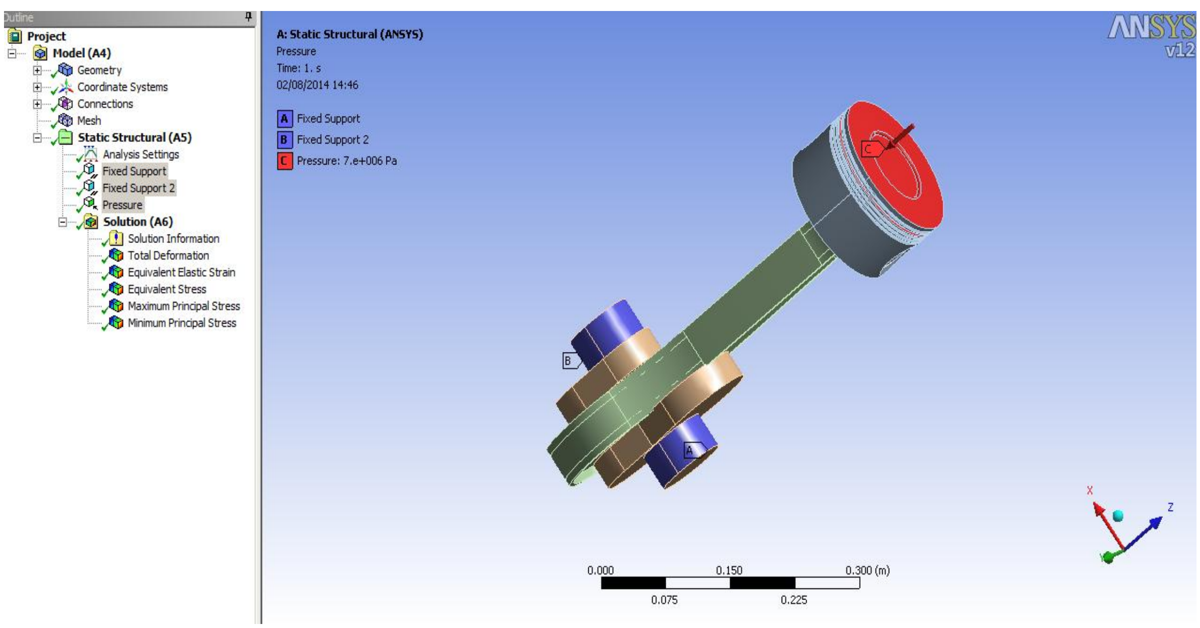


figure:5.3 apply fixed support and load of the Diesel Engine for Diesel-Electrical locomotive

5.1 Static Analysis result using ANSYS workbench on Single Engine cylinder

In the static analysis, 70bar pressure load has been applied on the top of Piston Engine Model in the axial direction, and the end corners of the upper and bottom Crank Shaft of Crank Pin have used as fixed points only for translation for boundary condition. After several runs, the optimum value for the discussed geometric parameters has been finalized with a safety factor equal 2.88 for the maximum Von-Misses stress of 983.33 MPa and Yield strength of the material is 2827 MPa. Therefore the pressure applied on the top of the Piston is found safe where the margin of safety for diesel Engine is 1.05. The highest stress value is observed on the Crank Pin which connect the Piston and smaller end of Connecting Rod show in the figure:5.5. The total deformation in the Diesel Engine of locomotive maximum 0.0022169m appear on the top of the piston, so that it is recommended or within standard limit show in figure:5.4.

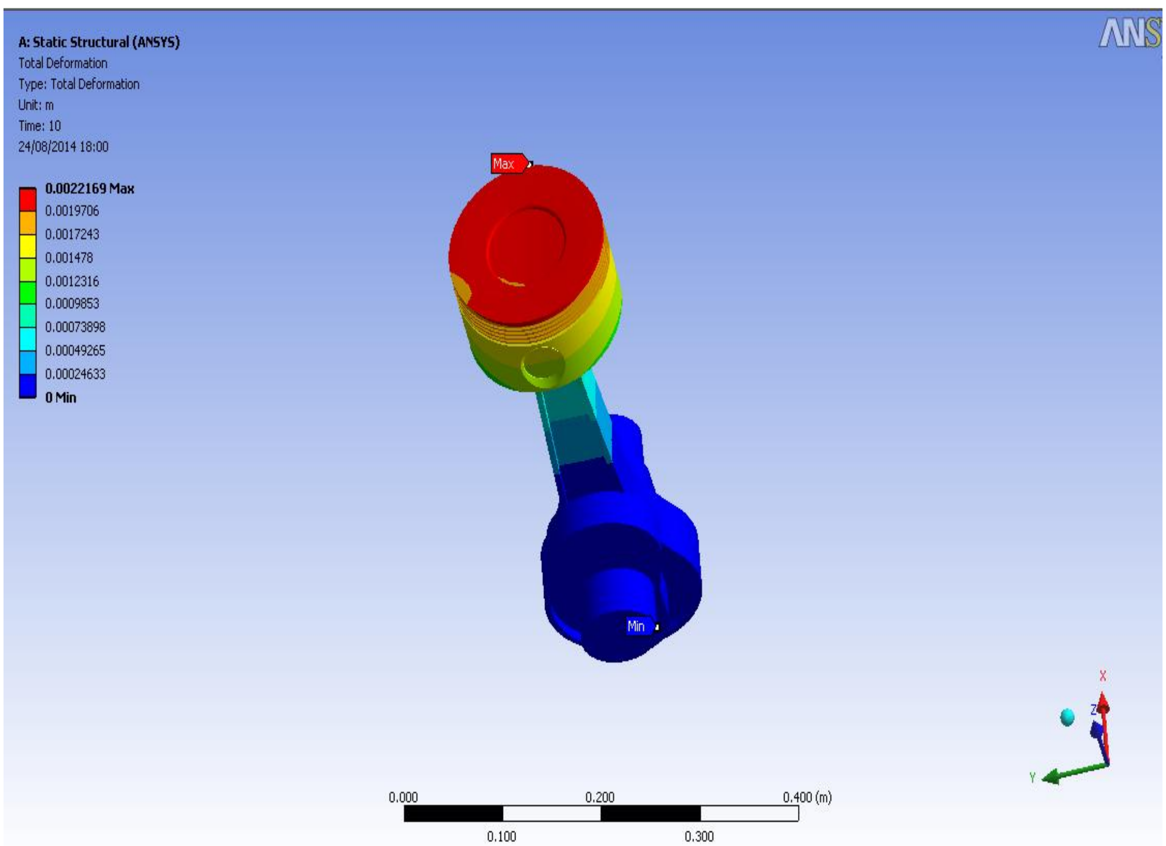


Figure 5.4 Total Deformation in the Diesel Engine for Diesel-Electrical locomotive

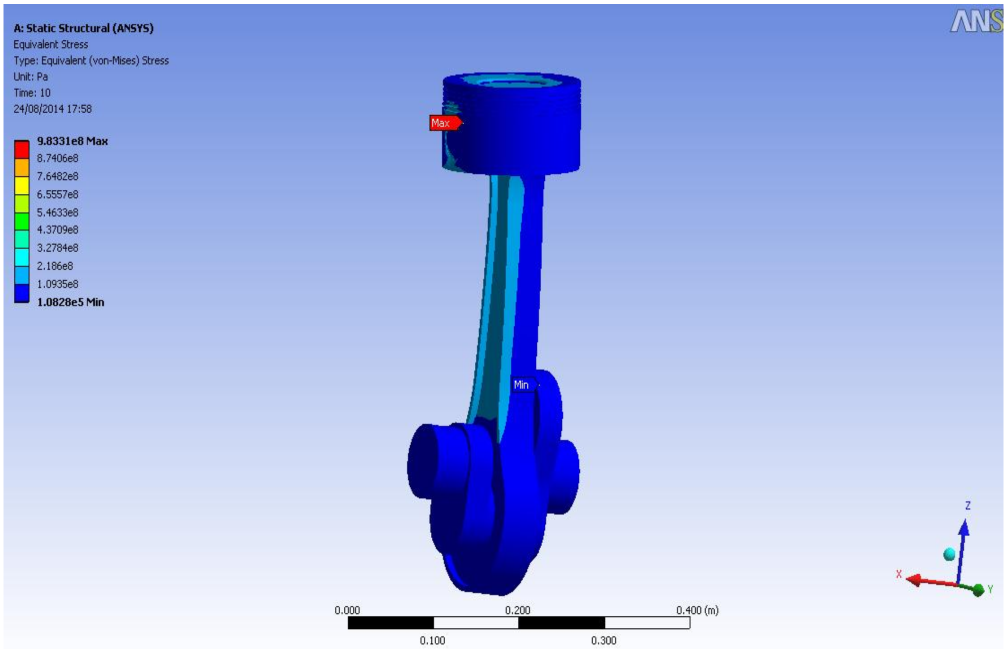


Figure 5.5 Von-Mises Stress on the diesel Engine for Diesel-Electrical locomotive

5.2 Result summery for Static Analysis using ANSYS workbench

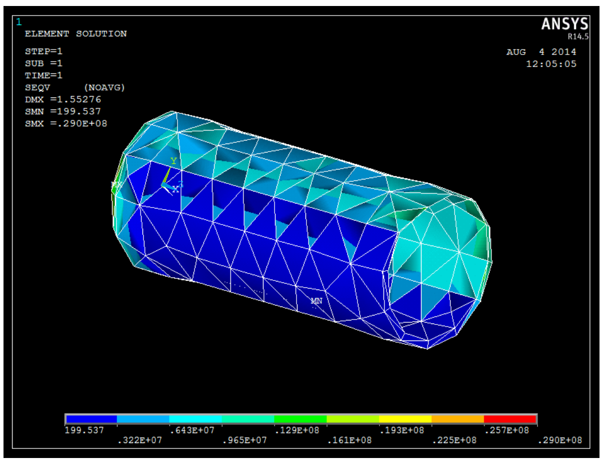
TABLE 5.1
Model (A4) > Static Structural (A5) > Solution (A6) > Results

Object Name	*Equivalent Stress*	*Total Deformation*
State	Solved	
Scope		
Scoping Method	Geometry Selection	
Geometry	All Bodies	
Definition		
Type	Equivalent (von-Mises) Stress	Total Deformation
By	Time	
Display Time	Last	
Calculate Time History	Yes	
Use Average	Yes	
Results		
Minimum	1.0828e+005 Pa	0. m
Maximum	9.8331e+008 Pa	2.2169e-003 m
Minimum Occurs On	Crank shaft	
Maximum Occurs On	Crank pin	
Minimum Value Over Time		
Minimum	1.0828e+005 Pa	0. m
Maximum	1.0828e+005 Pa	0. m
Maximum Value Over Time		
Minimum	9.8051e+008 Pa	2.2169e-003 m
Maximum	9.8331e+008 Pa	2.2172e-003 m

5.3 Static Analysis result using ANSYS on Pressure Vessel

The analysis is carried out in Finite Element Method using ANSYS. A Cylindrical segment is loaded by internal pressure on the internal surface. Those pressure applied on the internal surface 120bar and 150bar, In the case of 120bar pressure applied on the pressure vessel we get maximum Von-Misses stress and total mechanical deformation 290bar and 0.176552mm respectively from this result factor of safety is 4.

In the case of 150bar internal pressure applied on Pressure Vessel we get the maximum Von-Missis stress and total mechanical deformation 438bar and 0.249741mm respectively from the above result factor of safety is 4.



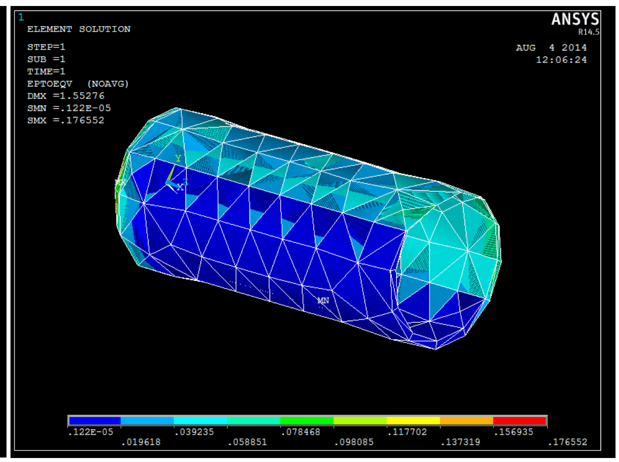
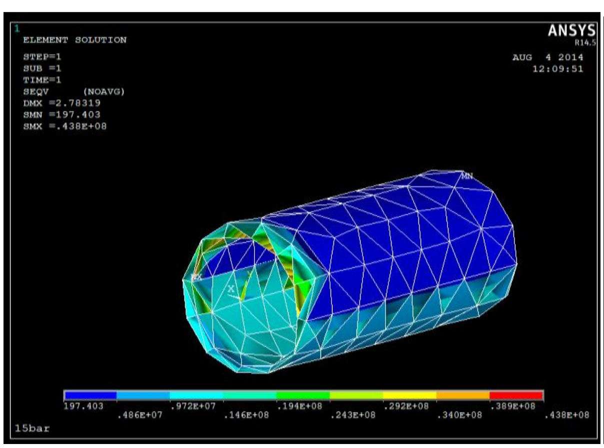


Figure 5.6: *Von-Mises Stress on the 12N/mm^2 Pressure vessel*

Figure 5.7: *Total deformation on the 12N/mm^2 Pressure vessel*



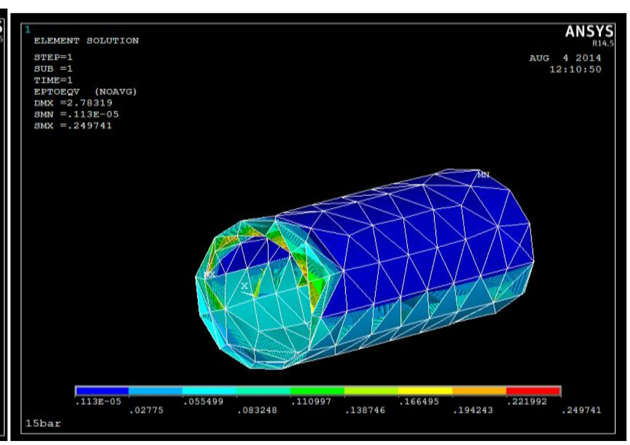


Figure 5.8: *Von-Mises Stress on the 15N/mm^2 Pressure vessel*

Figure 5.9: *Total deformation on the 15N/mm^2 Pressure vessel*

CHAPTER SIX: DISCUSSION AND CONCLUSIONS

In this Chapter a general discussion on the results is presented, the important conclusions are listed and some suggestions for further investigations are given.

6.1 Observations and Discussion

The work presented mainly concerns with compressed air Engine starting system directly Injected into the Engine Cylinder. Engine starting system by compressed air similar to Engine starting by Electrical Engine starting system but compressed air starting system is an auxiliary starting system.

6.2 Conclusions

In spite of the many successful applications of compressed air Engine Starting System, a number of Interesting Research in Nuclear Power Plant and Marin Engineering but it is new technology in Railway Industry. Some problems of current starting system or Electrical Starting System are: Loss connection, dead of Battery or failed of Starter Motor, compressed air Engine starting system come into operation and start the locomotive Engine. Obtaining the parameter of Engine data for validation of the Technique, calculating the amount of pressure.

Therefore the following conclusions can be drawn from the present study in an effort to evolve a new and better method starting system.

- ANSYS and ANSYS work bench is effective software and has Analytical techniques to predict the compressed air Engine Starting System based on the static structure with design pressure.
- The results of this Thesis show that the Von-Misses stresses which are produced during the operations are less than the design stress. Also the distribution of the total deformation is in prescribed limit. The average Piston stress and deformation 983.3Mpa and 0.0022169m respectively. So the Engine is safe to Resists Design pressure from literature point of view.
- The maximum von-Missis stress on pressure Vessel within predetermine value, So it is safe.
- The Maximum Von-Misses stresses occur on the Crank Pin and small ends of Connecting Rod so that it needs special attention but maximum total deformation occur on the top of the Piston.

- Although this work was aimed at the evaluation of the method for its application to Railway Engineering, it is clear that the same conclusions apply to a whole range of hardware.

6.3 Recommendation

From the above result Compressed Air Engine starting system is applicable in Railway Industry so that Ethiopia railway corporation can be installed this system in light Diesel-Electrical locomotive because it is light weight, in expensive price, compatible with locomotive structure, Indirectly it can be avoid traffic gaming and improve country economy.

6.4 Suggestions for future work

Some of the suggestions for further investigations are:

- The numerical and software analysis of compressed Air Engine Starting System applicable in hardware.
- To verify the method through an experiment in a controlled laboratory set up to see its ease of application and effectiveness compressed Air Engine Starting System.
- Modeling and Simulation of a Three-stage Air Compressor based on Dry Piston Technology.
- To analysis of Pressure Reducing Valve, Pressure Relief Valve and Electro pneumatic Valve with CFD (computational fluid dynamic).
- Loose calculation specially on the Third Stage Air Compressor

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Appendix A: Result In Put for CATIA and ANSYS Workbench

Model (A4)

Geometry

TABLE 5.2
Model (A4) > Geometry

Object Name	*Engine Assembly*
State	Fully Defined
Definition	
Type	Iges
Length Unit	Meters
Element Control	Program Controlled
Display Style	Part Color
Bounding Box	
Length X	0.187 m
Length Y	0.16 m
Length Z	0.54047 m
Properties	
Volume	3.5393e-003 m³
Mass	27.784 kg
Scale Factor Value	1.
Statistics	
Bodies	4
Active Bodies	4
Nodes	18849
Elements	9374
Mesh Metric	None
Analysis Type	3-D
Mixed Import Resolution	None
Enclosure and Symmetry Processing	Yes

TABLE 5.3
Model (A4) > Geometry > Parts

Object Name	*crankshaft*	*Connecting rod*	*piston*	*Piston pin*
State	Meshed			
Graphics Properties				
Visible	Yes			
Transparency	1			
Material				
Assignment	Structural Steel			
Nonlinear Effects	Yes			
Thermal Strain Effects	Yes			
Bounding Box				
Length X	0.15 m	0.16 m	4.6e-002 m	0.187 m
Length Y	4.e-002 m	0.16 m		0.115 m

Length Z	4.e-002 m	9.e-002 m	0.52 m	0.204 m
Properties				
Volume	1.8849e-004 m³	6.7366e-004 m³	1.2118e-003 m³	1.4653e-003 m³
Mass	1.4796 kg	5.2882 kg	9.513 kg	11.503 kg
Centroid X	0.22205 m	0.222 m	0.21374 m	0.21007 m
Centroid Y	-1.047e-004 m	4.4341e-004 m	4.3508e-004 m	-3.155e-007 m
Centroid Z	0.35086 m	0.37233 m	9.2691e-002 m	-8.8252e-003 m
Moment of Inertia Ip1	2.924e-004 kg·m²	1.6042e-002 kg·m²	0.21863 kg·m²	2.5999e-002 kg·m²
Moment of Inertia Ip2	2.9071e-003 kg·m²	1.5801e-002 kg·m²	0.20894 kg·m²	4.9829e-002 kg·m²
Moment of Inertia Ip3	2.9072e-003 kg·m²	2.3251e-002 kg·m²	1.163e-002 kg·m²	3.651e-002 kg·m²
Statistics				
Nodes	758	20926	4609	4536
Elements	245	10978	2302	2177
Mesh Metric	None			

Mesh

TABLE 5.4
Model (A4) > Mesh

Object Name	*Mesh*
State	Solved
Defaults	
Physics Preference	Mechanical
Relevance	0
Sizing	
Use Advanced Size Function	Off
Relevance Center	Coarse
Element Size	Default
Initial Size Seed	Active Assembly
Smoothing	Medium
Transition	Fast
Span Angle Center	Coarse
Minimum Edge Length	2.e-003 m
Inflation	
Inflation Option	Smooth Transition
Transition Ratio	0.272
Maximum Layers	5
Growth Rate	1.2
Statistics	
Nodes	18849
Elements	9374
Mesh Metric	None

Static Structural (A5)

TABLE 5.5
Model (A4) > Analysis

Object Name	*Static Structural (A5)*
State	Solved
Definition	
Physics Type	Structural
Analysis Type	Static Structural
Solver Target	ANSYS Mechanical
Options	
Environment Temperature	22. °C
Generate Input Only	No

TABLE 5.6
Model (A4) > Static Structural (A5) > Loads

Object Name	*Fixed Support*	*Pressure*
State	Fully Defined	
Scope		
Scoping Method	Geometry Selection	
Geometry	4 Faces	
Definition		
Type	Fixed Support	Pressure
Suppressed	No	
Define By		Normal To
Magnitude		7.e+006 Pa (ramped)

Material Data

Structural Steel

TABLE 5.7
Structural Steel > Constants

Structural Steel > Constants

Density	7850 kg m^-3
Coefficient of Thermal Expansion	1.2e-005 C^-1
Specific Heat	434 J kg^-1 C^-1
Thermal Conductivity	60.5 W m^-1 C^-1
Resistivity	1.7e-007 ohm m

TABLE 5.8
Structural Steel > Compressive Ultimate Strength

Compressive Ultimate Strength Pa
0

TABLE 5.9
Structural Steel > Compressive Yield Strength

Compressive Yield Strength Pa
2.5e+008

TABLE 5.10
Structural Steel > Tensile Yield Strength

Tensile Yield Strength Pa
2.5e+008

TABLE 5.11
Structural Steel > Tensile Ultimate Strength

Tensile Ultimate Strength Pa
4.6e+008

TABLE 5.12
Structural Steel > Alternating Stress

Alternating Stress Pa	Cycles	Mean Stress Pa
3.999e+009	10	0
2.827e+009	20	0
1.896e+009	50	0
1.413e+009	100	0
1.069e+009	200	0
4.41e+008	2000	0
2.62e+008	10000	0
2.14e+008	20000	0
1.38e+008	1.e+005	0
1.14e+008	2.e+005	0
8.62e+007	1.e+006	0

TABLE 5.13

Structural Steel > Strain-Life Parameters

Strength Coefficient Pa	Strength Exponent	Ductility Coefficient	Ductility Exponent	Cyclic Strength Coefficient Pa	Cyclic Strain Hardening Exponent
9.2e+008	-0.106	0.213	-0.47	1.e+009	0.2

Appendix B: Figure for compressor, valve and filter

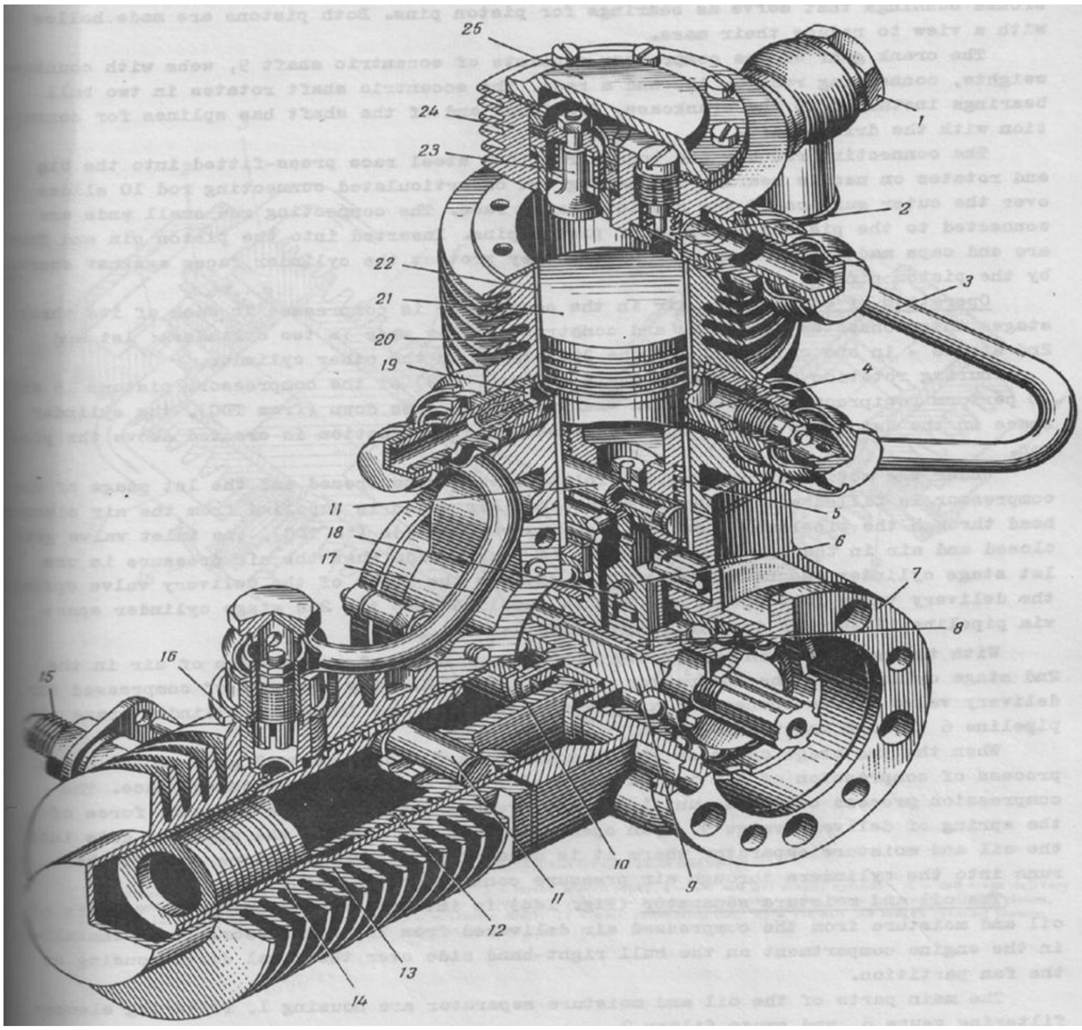


Figure.3.2 Third stage Air Compressor[4]

1-pipe union for supply of air from air cleaner: 2-1st stage delivery valve: 3-1st-to-2nd stage connecting pipe: 4-2nd stage inlet valve: 5-comprssion ring: 6-main connecting rod: 7-case: 8-needle bearing: 9-eccentric shaft: 10-articulating connecting rod: 11-pistons pins: 12-3rd stage cylinder piston: 13-3rd stage cylinder: 14 and 22- cylinders liners: 15-3rd stage delivery valve: 16-3rd stage inlet valve: 17-pipe: 18-screw: 19-2nd stage delivery valve: 20-1st and 2nd stages cylinder: 21-1st and 2nd stages cylinder piston: 23-1st stage inlet valve: 24-1st and 2nd stages cylinder head:

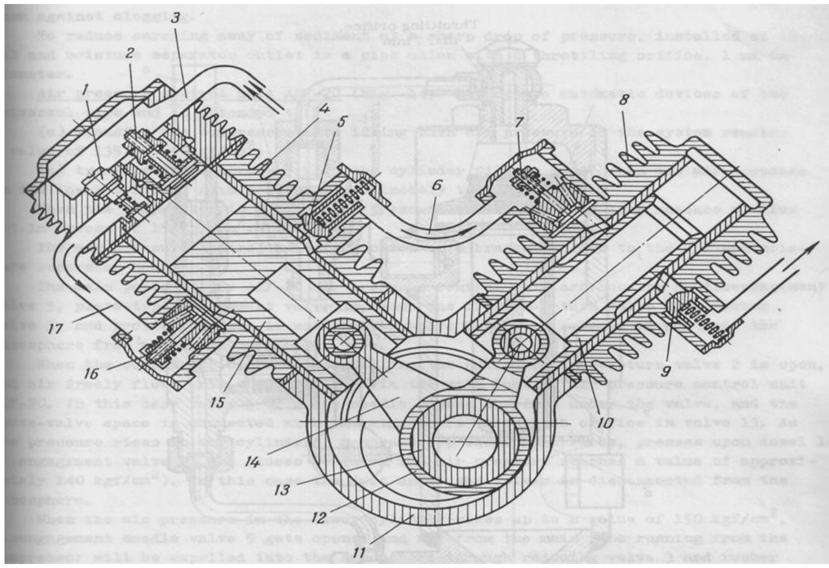


Figure 3.3 Third stage Air Compressor Operation Diagram[4]

1-1st stage delivery valve: 2-1st stage inlet valve; 3-intake branch pipe; 4-1st and 2nd stages cylinder; 5-2nd stage delivery valve; 6 and 17-pipelines; 7-3rd stage inlet valve; 8-3rd stage cylinder; 9- 3rd stage delivery valve; 10-3rd stage cylinder piston; 11- case; 12-articulating connecting rod; 13-eccentric shaft; 14-main connecting rod; 15- 1st and 2nd stages cylinder piston; 16-2nd stage inlet valve.

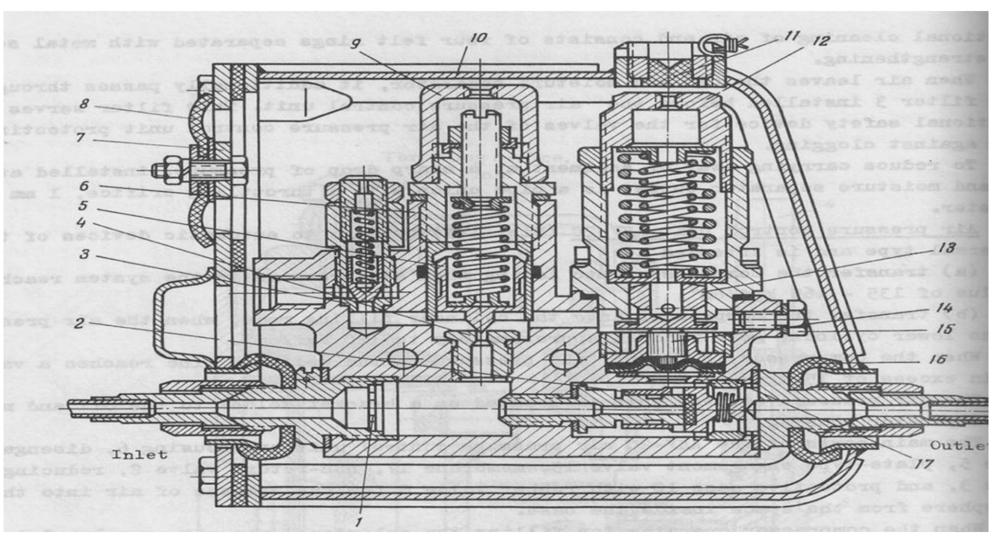


Figure 3.4 Air pressure reducing valve[11]

1-filter; 2-non return valve; 3-engagement reducing valve; 4-valve seat; 5-disengagement needle valve; 6-housing; 7-case cover; 8-valve; 9-cap; 10-case; 11-plug for scavenging valves; 12-nut; 13-plate-type engagement valve; 14-dowel; 15-plug; 16-sealing; 17-membran.

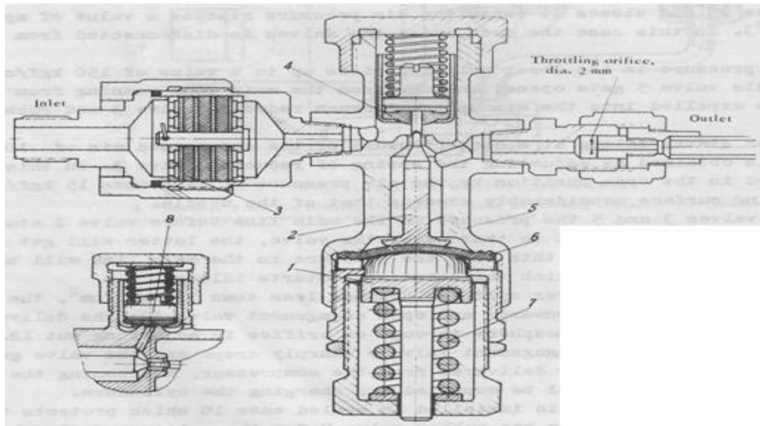


Figure 3.5 Air pressure Reducing valve[11]

1-piston; 2-push rod; 3-filtering element; 4-high pressure valve; 5-reducer body; 6-membran; 7-spring; 8-safety valve

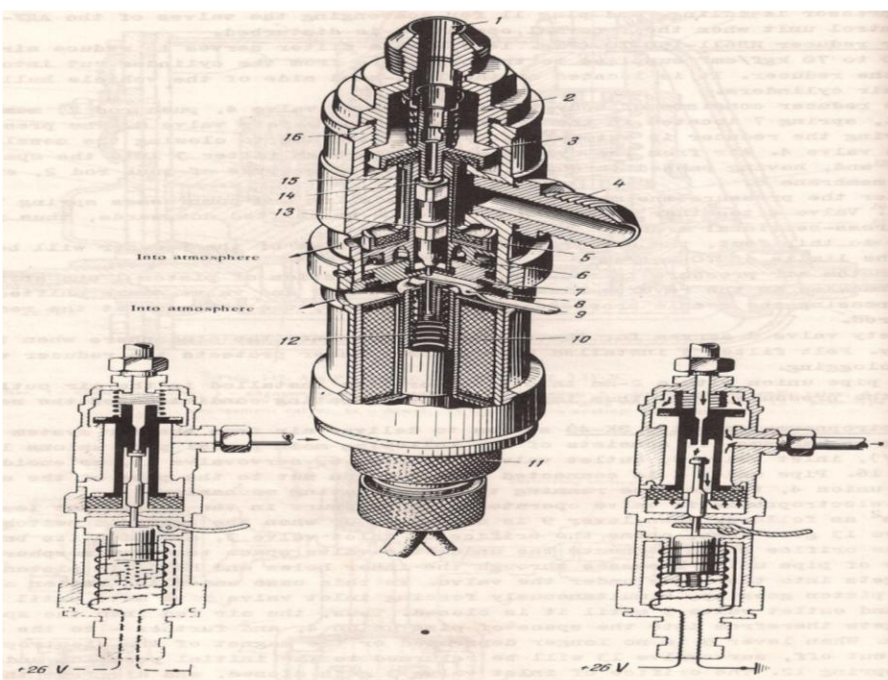


Figure 3.6 Electro pneumatic valve AK-48[4]

1-pipe union; 2-valve spring; 3-inlet valve; 4-pipe union; 5-outlet valve; 6-piston; 7-holder; 8-shaft; 9-lever; solenoid switch; 11-plug connector; 12-spring; 13-servovalve; 14-adjusting gasket; 15-distance busing; 16-body.

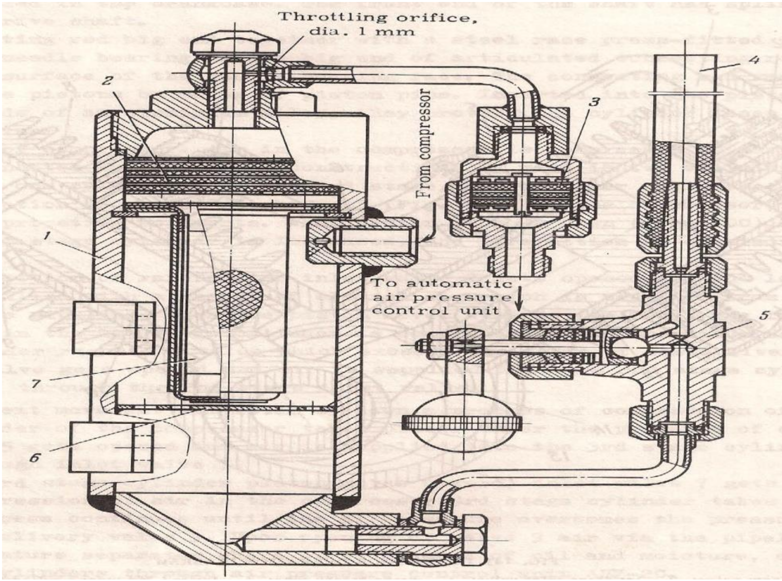


Figure 4.1 Oil and moisture separator[4]

1-housing; 2 and 3-filtering element; 4-hose for draining sediment; 5-cock for draining sediment; 6-filtering gauze; 7-gauze filter

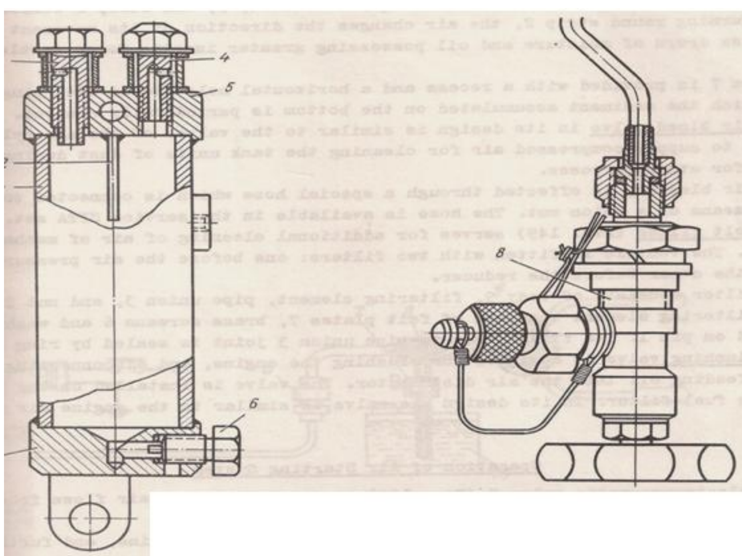


Figure 4.2 sediment bowl air bleed valve[4].

1-bowl body; 2-strip; 3,4- pipe union; 5-covre; 6-plug; 7-bowl bottom; 8-air bleed valve

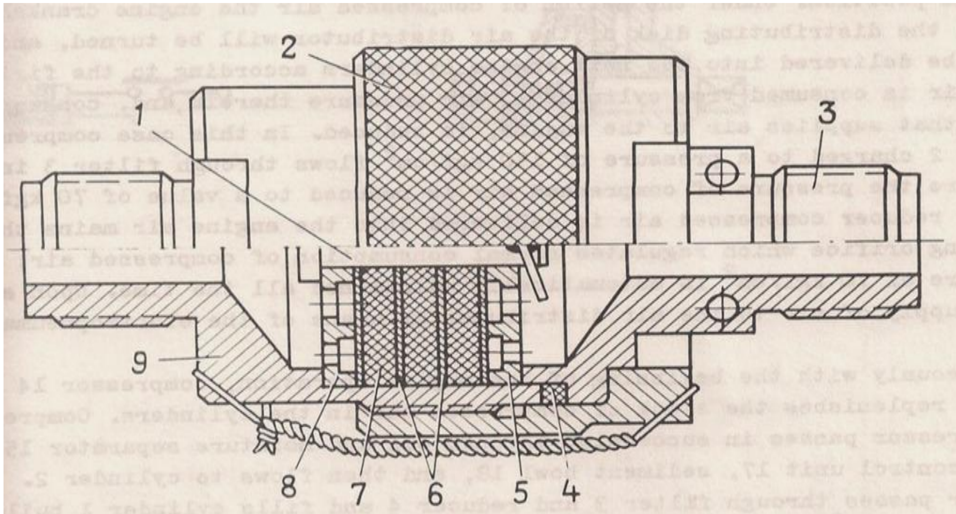


Figure 4.3 mechanical air Filter[4]

1-filtering element pin; 2-nut; 3-pipe union; 4-packing ring; 5 and 8-washers; 6-screens; 7-felt plate; 9-filter body

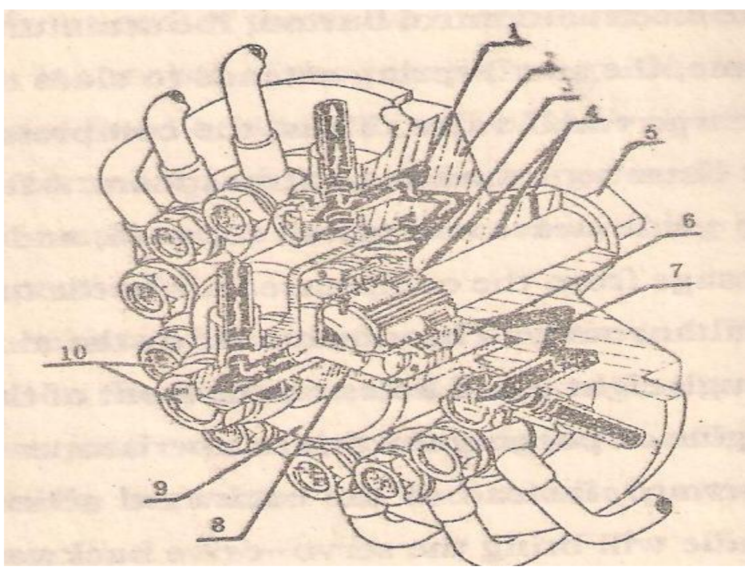


Figure:4.5 Air Distributor[12]

1. Air port, 2. body 3. Adjusting toothed sleeve 4.Oil holes 5.Dirive shaft 6. Distributing disk, 7.air passage, 8.Distributing disk cover, 9.Air distributing hood, 10.Hollow bolt.

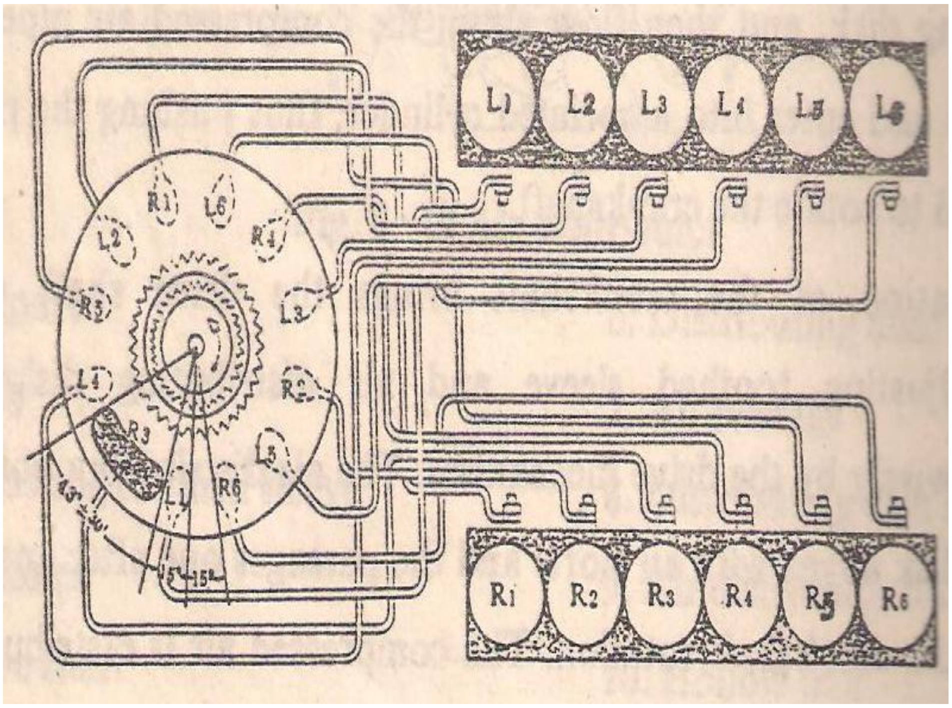


Figure:4.6 scheme of the operation of the air distributor[12]

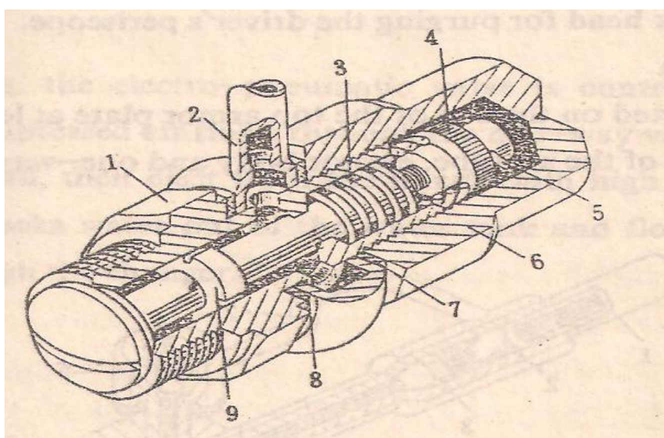


Figure:4.7 Air Starting Valve[12]

1. body, 2.connection of air pipe, 3.valve spring, 4.Adjusting nut, 5 cotter pin. 6.cap nut, 7.8.seal ring, 9.valve

Appendix C: pressure reducing valve Table and chart

Table 3.2:Typical properties of Gases

Gas or Vapor	Molecular Weight M	Ratio of Specific Heats k (14.7 psia)	Coefficient C*	Specific Gravity	Critical Pressure psia	Critical Temp.(°R) (°F+460)
Acetylene	26.04	1.25	342	0.899	890	555
Air	28.97	1.40	356	1.000	547	240
Ammonia	17.03	1.30	347	0.588	1638	730
Argon	39.94	1.66	377	1.379	706	272
Benzene	78.11	1.12	329	2.696	700	1011
N-Butane	58.12	1.18	335	2.006	551	766
Iso-Butane	58.12	1.19	336	2.006	529	735
Carbon Dioxide	44.01	1.29	346	1.519	1072	548
Carbon Disulphide	76.13	1.21	338	2.628	1147	994
Carbon Monoxide	28.01	1.40	356	0.967	507	240
Chlorine	70.90	1.35	352	2.447	1118	751
Cyclohexane	84.16	1.08	325	2.905	591	997
Ethane	30.07	1.19	336	1.038	708	550
Ethyl Alcohol	46.07	1.13	330	1.590	926	925
Ethyl Chloride	64.52	1.19	336	2.227	766	829
Ethylene	28.03	1.24	341	0.968	731	509
Freon 11	137.37	1.14	331	4.742	654	848
Freon 12	120.92	1.14	331	4.174	612	694
Freon 22	86.48	1.18	335	2.985	737	665
Freon 114	170.93	1.09	326	5.900	495	754
Helium	4.02	1.66	377	0.139	33	10
N-Heptane	100.20	1.05	321	3.459	397	973
Hexane	86.17	1.06	322	2.974	437	914
Hydrochloric Acid	36.47	1.41	357	1.259	1198	584
Hydrogen	2.02	1.41	357	0.070	188	60
Hydrogen Chloride	36.47	1.41	357	1.259	1205	585
Hydrogen Sulphide	34.08	1.32	349	1.176	1306	672
Methane	16.04	1.31	348	0.554	673	344
Methyl Alcohol	32.04	1.20	337	1.106	1154	924
Methyl Butane	72.15	1.08	325	2.491	490	829
Methyl Chloride	50.49	1.20	337	1.743	968	749
Natural Gas (Typical)	19.00	1.27	344	0.656	671	375
Nitric Oxide	30.00	1.40	356	1.036	956	323
Nitrogen	28.02	1.40	356	0.967	493	227
Nitrous Oxide	44.02	1.31	348	1.520	1054	557
N-Octane	114.22	1.05	321	3.943	362	1025
Oxygen	32.00	1.40	356	1.105	737	279
N-Pentane	72.15	1.08	325	2.491	490	846
Iso-Pentane	72.15	1.08	325	2.491	490	829
Propane	44.09	1.13	330	1.522	617	666
Sulfur Dioxide	64.04	1.27	344	2.211	1141	775
Toluene	92.13	1.09	326	3.180	611	1069

*If "C" is not known then use C = 315

If the ratio of specific heats " k" is known, reference page 7-9 to calculate "C".

Table 3.3: Correction Factor for Vapors and Gases, K_o for conventional Valves With Constant Backpressure and Styles JPV/JPVM Pilot valves With Back Pressures Exceeding Critical pressure.

P_b/P_1=Back Pressure Percentage

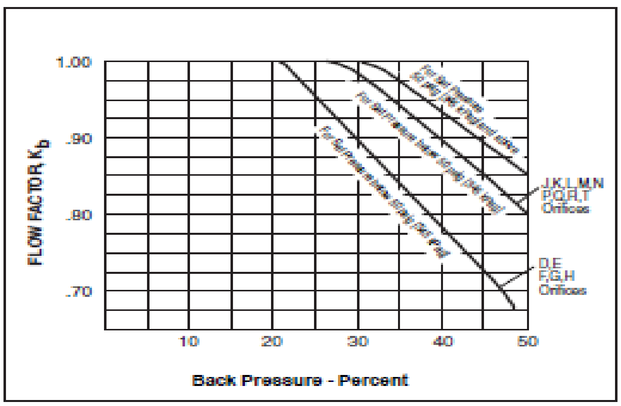
=Back Pressure (absolute)/ Reliving Pressure (absolute)*100

P_b / P_1	K_b	P_b / P_1	K_b	P_b / P_1	K_b
55	1.00	72	0.93	86	0.75
60	0.995	74	0.91	88	0.70
62	0.99	76	0.89	90	0.65
64	0.98	78	0.87	92	0.58
66	0.97	80	0.85	94	0.49
68	0.96	82	0.81	96	0.39
70	0.95	84	0.78		

* Critical pressure is generally taken as 55% of accumulated inlet pressure, absolute

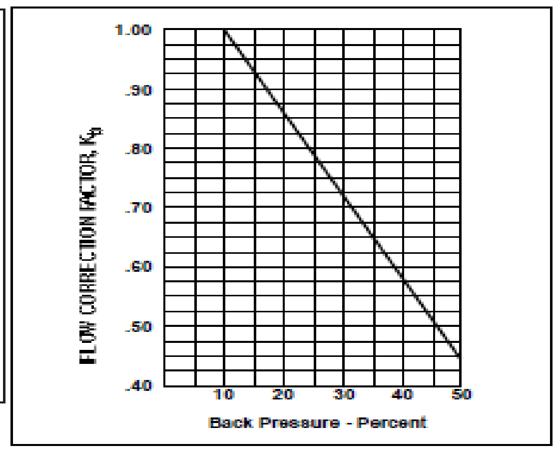
Correction Factor for Vapors and Gases, K_b

Figure F7-2A
Style JBS Valves - 10% Overpressure



$\frac{\text{Back Pressure (gage)}}{\text{Set Pressure (gage)}}$ x 100

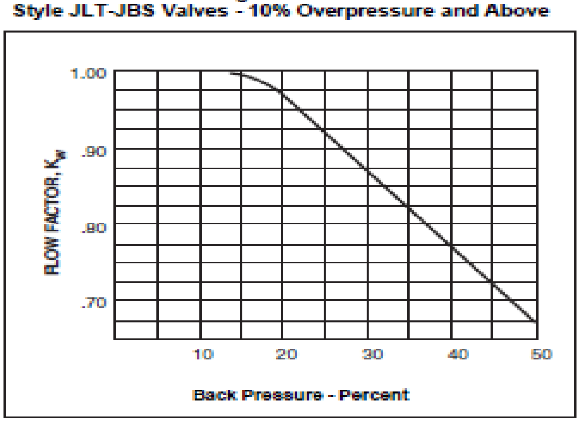
Figure F7-2B
Series BP Valves - 10% Overpressure



$\frac{\text{Back Pressure (gage)}}{\text{Set Pressure (gage)}}$ x 100

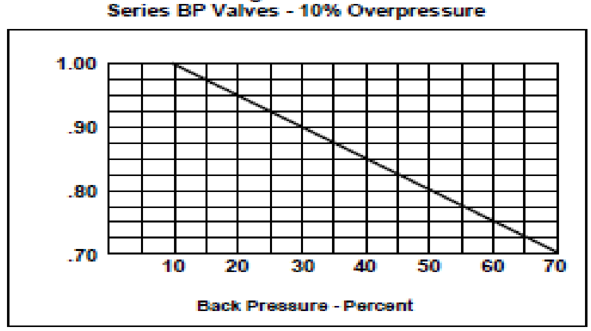
Correction Factor for Liquids, K_w

Figure F7-3A
Style JLT-JBS Valves - 10% Overpressure and Above



$\frac{\text{Back Pressure (gage)}}{\text{Set Pressure (gage)}}$ x 100

Figure F7-3B
Series BP Valves - 10% Overpressure



$\frac{\text{Back Pressure (gage)}}{\text{Set Pressure (gage)}}$ x 100

Figure 3.7:-Correction Factor for High Pressure Steam. K_o

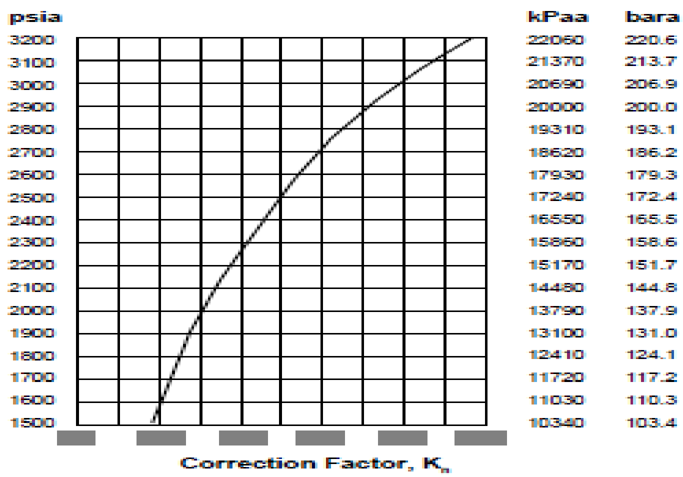


Figure 3.8:

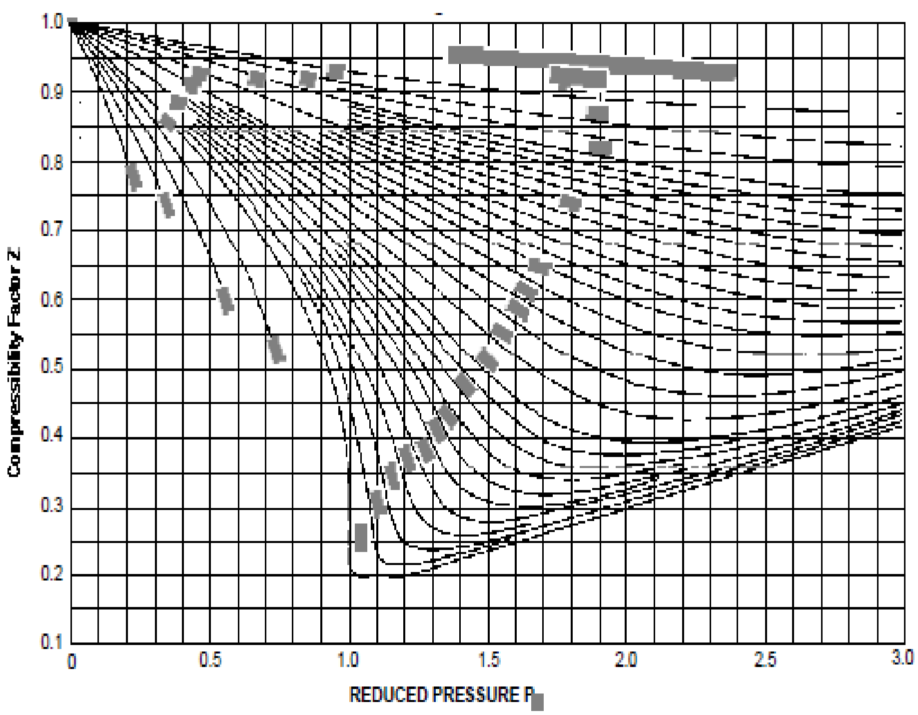


Figure 3.9:ratio of specific heat, k vs Coefficient, c

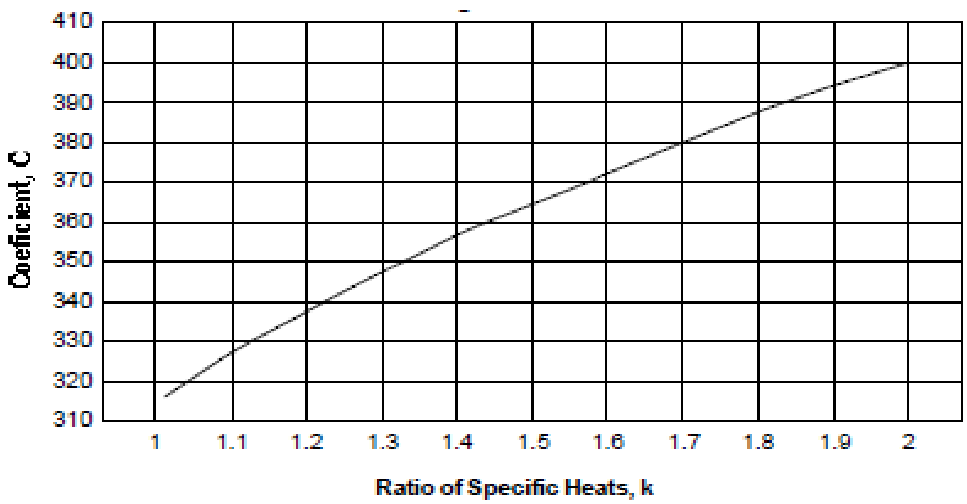


Table 3.4

k	C	k	C	k	C	k	C	k	C
1.01	317	1.21	338	1.41	357	1.61	373	1.81	388
1.02	318	1.22	339	1.42	358	1.62	374	1.82	389
1.03	319	1.23	340	1.43	359	1.63	375	1.83	389
1.04	320	1.24	341	1.44	360	1.64	376	1.84	390
1.05	321	1.25	342	1.45	360	1.65	376	1.85	391
1.06	322	1.26	343	1.46	361	1.66	377	1.86	391
1.07	323	1.27	344	1.47	362	1.67	378	1.87	392
1.08	325	1.28	345	1.48	363	1.68	379	1.88	393
1.09	326	1.29	346	1.49	364	1.69	379	1.89	393
1.10	327	1.30	347	1.50	365	1.70	380	1.90	394
1.11	328	1.31	348	1.51	365	1.71	381	1.91	395
1.12	329	1.32	349	1.52	366	1.72	382	1.92	395
1.13	330	1.33	350	1.53	367	1.73	382	1.93	396
1.14	331	1.34	351	1.54	368	1.74	383	1.94	397
1.15	332	1.35	352	1.55	369	1.75	384	1.95	397
1.16	333	1.36	353	1.56	369	1.76	384	1.96	398
1.17	334	1.37	353	1.57	370	1.77	385	1.97	398
1.18	335	1.38	354	1.58	371	1.78	386	1.98	399
1.19	336	1.39	355	1.59	372	1.79	386	1.99	400
1.20	337	1.40	356	1.60	373	1.80	387	2.00	400

Figure 3.10:

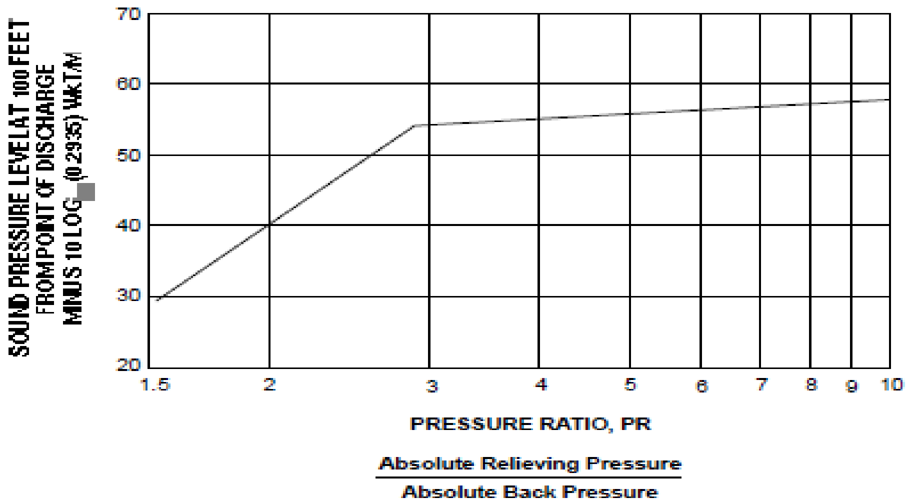


Table 3.4:

Relative Noise Levels			
130	Decibels	-	Jet Aircraft on Takeoff
120	Decibels	-	Threshold of Feeling
110	Decibels	-	Elevated Train
100	Decibels	-	Loud Highway
90	Decibels	-	Loud Truck
80	Decibels	-	Plant Site
70	Decibels	-	Vacuum cleaner
60	Decibels	-	Conversation
50	Decibels	-	Offices

Table 3.5:

k	K_r
1.01	1.15
1.05	1.13
1.10	1.11
1.15	1.09
1.20	1.07
1.25	1.05
1.30	1.03
1.35	1.02
1.40	1.00
1.45	0.98
1.50	0.97
1.55	0.95
1.60	0.94
1.65	0.93
1.70	0.91
1.75	0.90
1.80	0.89
1.85	0.87
1.90	0.86
1.95	0.85
2.00	0.84

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Symbol	Definition
D	Compressor capacity
V	Volume of the Receiver
P_1	Cut-in Pressure
P_2	Desired operating pressure
T	Recharging time
r_c	Overall or total compression ratio
C_p	Specific heat at constant pressure
C_v	Specific heat at constant volume
C_{pmix}	Gas mixture specific heat
T_{ave}	Average temperature (K)
T_d	Discharge temperature (K)
T_s	Suction temperature (K)
N_{stg}	Number of stages
r_{c-stg}	Compression ratio per stage
θ_s	Suction valve loss
a	Piston/valve area ratio
U	Average piston speed in meter per second
M_w	Gas Molecular weight
T_1	Suction temperature
L	Piston Stroke
N	Design rotating Speed, in RPM
η_c	Compression Efficiency
Q_{std}	Inlet Capacity volume at *standard condition*
Q_s	Inlet Capacity volume at *inlet condition*
P_s	Inlet pressure
T_s	Inlet temperature
Z_s	Compressibility at inlet

M	Mass flow rate of gas (kg/min)
Q	Volume flow rate (m^3/min)
ν	Specific volume of gas (m^3/kg) = $\frac{1}{\rho_{gas}}$
Z	Gas Compressibility
R	Gas Constant = 8314/M_w
M_w	Gas Molecular Weight
H_{ad}	Adiabatic head of compression (kN.m/kg)
Z_{av}	Average gas compressibility at suction and discharge
GP	Compression Power or Gas Power (kW)
ε_v	Volumetric Efficiency
A_{hd}	Area of Head-end
D	Cylinder diameter
d	piston rod size diameter
A_{ce}	Area of Crank-end
F_{LC}	Frame load in tension
P_r	Critical Pressure
T_r	Critical Temperature
T_{Cm}	Mixture Critical Temperature
P_{Cm}	Mixture Critical Pressure
T_{CI}	Individual gas Critical Temperature
P_{Ci}	Individual gas Critical Pressure
x_i	Mol percent or percent by volume of each gas
Δp	Pressure decrease
p	Working pressure
V	Volumetric flow rate
$\frac{dV}{dt}$	Atmosphere
l	Pipe length
D_i	Outside diameter
μ	Dynamic viscosity of the fluid

R_e	Reynolds number
f	Frictional factor
A	Minimum required effective discharge area
C	The ratio of specific heats of the gas at standard conditions.
K	Effective coefficient of discharge
K_b	Capacity correction factor due to back pressure
BP	Variable back pressure
M	Molecular weight of the gas
W	Required relieving capacity
G	Specific gravity of the gas
Q	Required relieving capacity
L_{100}	Sound level at 100 feet from the point of discharge
L_P	Sound level at a distance
r	Distance from the point of discharge
F_g	Component of reaction force due to static pressure
F_s	Component of reaction force due to static pressure
A_o	Outlet cross-sectional area
K_n	High pressure steam correction factor
K_r	Correction for ratios of specific heats
C.A	Corrosion Allowance
S_c	Allowable stress at design temperature of liner
S_1	Allowable stress at design temperature of layers
n	Number of layers.
t_c	Thickness of liner or core tube.
t_1	Thickness of layer