İSTANBUL TECHNICAL UNIVERSITY ★ INSTITUTE OF SCIENCE AND TECHNOLOGY

THE CORRELATION OF EXPERIMENTAL TESTS FOR PNEUMATIC VALVES OF VACUUM SYSTEM IN DIESEL ENGINES BY USING 1 D SIMULATION METHOD

M.Sc. Thesis by
Ömer CEYHUN

Department : Mechanical Engineering

Programme : Automotive Engineering

JANUARY 2011

İSTANBUL TECHNICAL UNIVERSITY ★ INSTITUTE OF SCIENCE AND TECHNOLOGY

THE CORRELATION OF EXPERIMENTAL TESTS FOR PNEUMATIC VALVES OF VACUUM SYSTEM IN DIESEL ENGINES BY USING 1 D SIMULATION METHOD

M.Sc. Thesis by
Ömer CEYHUN
(503081712)

Date of submission : **20 December 2010**
Date of defence examination: **25 January 2011**

Supervisor (Chairman) : **Assis. Prof. Dr. Akın KUTLAR (ITU)**
Members of the Examining Committee : **Prof. Dr. Metin ERGENEMAN (ITU)**
Prof. Dr. İrfan YAVAŞLIOL(YTU)

JANUARY 2011

İSTANBUL TEKNİK ÜNİVERSİTESİ ★ FEN BİLİMLERİ ENSTİTÜSÜ

DİZEL MOTORLARINDA VAKUM İLE ÇALIŞAN PNÖMATİK VALFLERİN DENEYSEL TESTLERİNİN 1 BOYUTLU SİMÜLASYON METODU İLE KORELASYONU

YÜKSEK LİSANS TEZİ

Ömer CEYHUN
(503081712)

Tezin Enstitüye Verildiği Tarih : 20 Aralık 2010
Tezin Savunulduğu Tarih : 25 Ocak 2011

Tez Danışmanı : Yrd. Doç. Dr. O. Akın KUTLAR (İTÜ)
Diğer Jüri Üyeleri : Prof. Dr. Metin ERGENEMAN (İTÜ)
Prof. Dr. İrfan YAVAŞLIOL (YTÜ)

OCAK 2011

FOREWORD

I would like to express my deep appreciation and thanks for my advisor Assis. Prof. Dr. Osman Akın Kutlar for his time, support and critical view during this project. This work is supported by ITU Mechanical Engineering Faculty Automotive Engineering. Furthermore, I wish to thank Ford Otosan for implementing the vacuum models and carrying out the measurements.

December 2010

Ömer Ceyhun

(Mechanical Engineering)
(Automotive)

TABLE OF CONTENTS

Page

ABBREVIATIONS

AC	: Alternative Current
Atm	: Atmosphere
DTCY	: Duty Cycle
ECU	: Engine Control Unit
EGR	: Exhaust Gas Recirculation
EU	: European Union
EVRV	: Electronic Vacuum Regulating Valve
Ft	: Feets
HV	: High Vacuum
In-Hg	: Inches of Mercury
J/K	: Joule/Kelvin
Kn	: Knudsen
Mbar	: Milibar
Mm	: Milimeter
Ms	: Milisecond
MV	: Medium Vacuum
Nm	: Newton meter
NRV	: Non Return Valve
Psi	: Pound per square inch
PWM	: Pulse Width Modulation
Re	: Reynold number
RPM	: Revolution per Minute
RV	: Rough Vacuum
SI	: International System of Units
TBV	: Turbine Bypass Valve
TGV	: Turbocharger with Variable Geometry
UHV	: Ultra High Vacuum
VAC	: Vacuum
VGT	: Variable Geometry Turbine
VNT	: Variable Nozzle Turbine
VSV	: Vacuum Switching Valve
WG	: Westgate

LIST OF TABLES

Page

LIST OF FIGURES

Page

THE CORRELATION OF EXPERIMENTAL TESTS FOR PNEUMATIC VALVES OF VACUUM SYSTEM IN DIESEL ENGINES BY USING 1 D SIMULATION METHOD

SUMMARY

The aim of this thesis is to present the theoretical and experimental work related to the vacuum system used for controlling the actuation of pneumatic valves in internal combustion engines in order to obtain a physical model of this system. In this context, these valves control the turbocharger operation in a two-stage sequential turbocharged diesel engine and egr cooler.

First, an overview is given of the physical principles and relations in vacuum technology. Equations used for vacuum are all reviewed and extracted for further processes.

In the next step, some related information about engine vacuum system is shared. Today's vacuum applications on engines are researched. While the engine vacuum system is being explained, real engine datas are used.

Components like a solenoid valve, a reservoir, a pump and an actuator have been implemented in the simulation model which includes real experimental datas performed by different engine suppliers. After that, measurements have been done to verify the implementation of the model . Due to unfortunate circumstances in calibration, only a limited number of measurements could be done.

Related to the theoretical contents, model of the vacuum system is developed by using a 1D approach and compared with the real experimental results. The aim is to make the simulation results as similar as when comparing the test results. So that, the model can be used to predict the behaviour of the vacuum system. Finally, the model is used to optimize the performance of the system in terms of response times by using different variations which are effective. This will contribute the engineer to develop and predict the general behaviour of the system prior to prototype part production by simulating with CAE. In addition to this, design process of the system can be shortened as well. Therefore a consistent methodology has been established in order to reproduce the vacuum system model and can be used as a designing tool for complex applications devoted to engine controlling tasks and devices in engineering.

DİZEL MOTORLARINDA VAKUM İLE ÇALIŞAN PNÖMATİK VALFLERİN DENEYSEL TESTLERİNİN 1 BOYUTLU SİMÜLASYON METODU İLE KORELASYONU

ÖZET

Bu tezin amacı içten yanmalı motorlarda vakum ile tahrik edilen pnömatik valflerin fiziksel modelini elde etmek için vakum sistemi ile ilgili teorik ve deneysel çalışmalar sunmaktır. Bu çalışmada, bahsi geçen valfler çift sıralı turboyu ve egr soğutucuyu kontrol etmektedir.

İlk olarak, vakum teknolojisi ile ilgi fiziksel ilkelere ve bağıntılara genel bir bakış yapıldı. Vakum ile ilgili tüm denklemler incelenerek sonraki süreçler için çıkartıldı.

Diğer aşamada, motor vakum sistemi ile ilgili bilgilere değinildi. Motorlarda günümüzde kullanılan vakum sistemleri ile ilgili araştırmalar yapıldı. Motor vakum sistemi anlatılırken, gerçek motor bilgileri ve özellikleri kullanıldı.

Farklı motor imalatçıları tarafından yapılan gerçek test datalarını içeren simülasyon modelinde valf, rezervuar, pompa ve aktüatör gibi parça modelleri kullanımı uygulandı. Daha sonra modelin uygulamasını doğrulamak için çeşitli ölçümler alındı. Birtakım kalibrasyondaki şansız durumlar yüzünden, gerçek test datası alımlarında zorluklar yaşandı.

Teorik içerikle ilgili olarak, vakum sistemi modeli tek boyutlu simülasyon yaklaşımı kullanılarak geliştirildi ve gerçek deneysel sonuçlarla mukayese edildi. Amaç analiz sonuçlarını test sonuçlarına mümkün olduğunca yaklaştırmaktır. Böylece oluşturulan 1D model ile vakum sisteminin davranışı tahmin edilebilir. Son olarak model, sistem performansını tepki süresi bakımından sisteme etkiyen çeşitli varyasyonları kullanarak optimize etmede kullanılabilir. Bu, araştırma geliştirme yapan mühendise ve kişiye bir boyutlu bilgisayar analizi kullanarak sistemin daha geliştirilmesi ve performansını tahmin etmede yardımcı olmaktadır. Ayrıca buna ek olarak, sistem tasarım süreci de kısaltılabilinir. Dolayısıyla vakum sistemi modeli üretmek için tutarlı bir yöntem kurulmuş olup, bu kurulan metodoloji motor kontrol görevlerine adanmış kompleks uygulamalar için tasarım aracı olarak kullanılabilinir.

1. INTRODUCTION

The use of pneumatic valves is widely extended in internal combustion engines to control different systems. Some of these systems are designed to improve the engine antipollution capabilities by means of exhaust gas recirculation (EGR). Other systems are related to the engine performance improvement in turbocharged diesel units by using a variable geometry turbine (VGT). The waste-gate (WG), which is widely used in turbocharged engines to limit the energy delivery to the turbine, is also a pneumatic valve [1]. Another example is represented by the pneumatic braking system used in commercial vehicles. In all these systems, the main aim is to ensure a controlled pressure in the actuator chamber of the air valve according to the application requests. The utilization of a vacuum system is avoided in some engines for controlling tasks. In these cases, electrically driven valves are used instead.

1.1 Purpose of the Thesis

The two main objectives of this study are to model a vacuum system of a internal combustion engine with a 1D simulation program and correlate the model with the experimental results. During the design phase of the system for a new engine, this will contribute to develop with less experimental tests and cost. Besides those advantages, this will bring a gain in time saving in the development process.

1.2 Background

Literature search showed that vacuum systems have been extensively studied in the field of brake apply systems of vehicles. However, very few investigations have been carried out related to the analysis of a vacuum system in automotive applications coupled with the internal combustion engine. There are several reasons for explaining this fact. At the very beginning, studies in internal combustion engines were performed under steady state running conditions so there was no point in analyzing the vacuum system since the valves EGR, VGT, or WG remained in constant position. Later on, most studies under transient engine operation did not consider the vacuum system modeling and directly actuated to the valve position. Nowadays, transient modeling has great importance and it should be considered the influence that the valve movements have during the engine evolution. They also have a strong impact in controlling tasks, which is becoming a valuable topic in engine development. The vacuum system also plays an important role in engines with two turbocharging systems because the number of valves to control increases. These relatively new engine concepts with sequential parallel or series turbocharging are becoming more and more popular in automotive applications since a reduction in engine capacity is possible, which leads to an improvement on engine fuel consumption and emissions while maintaining the engine performance [1].

2. VACUUM TECHNOLOGY

A vacuum is a space from which air or other gas has been removed. All gas can not be removed. The amount removed depends on the application and is done for many reasons. A vacuum is defined as a diluted gas, or the corresponding state at which its pressure or density is lower than that of the ambient surrounding atmosphere. Because atmospheric pressure fluctuates locally over the Earth's surface and lessens as altitude above sea level increases, it is not possible to specify a general upper limit for the vacuum range. The necessity for creating and maintaining vacuum conditions is usually related to the need to reduce the number of density of gaseous molecules or their surface collison rate.

2.1 Basic Terms and Concepts in Vacuum Technology

2.1.1 Pressure p (mbar)

Pressure is defined in DIN Standard 1314 as the quotient of standardized force applied to a surface and the extent of this surface (force referenced to the surface area). Even though the Torr is no longer used as a unit for measuring pressure it is nonetheless useful in the interest of transparency to mention this pressure unit: 1 Torr is that gas pressure which is able to raise a column of mercury by 1 mm at 0 ^{o}C (Standard atmospheric pressure is 760 Torr or 760 mm Hg.). Pressure p can be more closely defined by way of subscripts. Any gas enclosed within a volume is always uniformly distributed. The individual gas particles are constantly moving back and forth at high-speed within the volume; upon striking the vessel wall, they exert a force *F* on surface *A* due to pulse transmission. The pressure *p* that is exerted on the wall is defined as [2];

$$p = \frac{F}{A} = \frac{N}{mm^2} \qquad \textbf{(2.1)}$$

2.1.2 Total pressure p_t

The total pressure in a vessel is the sum of the partial pressures for all the gases and vapors within the vessel.

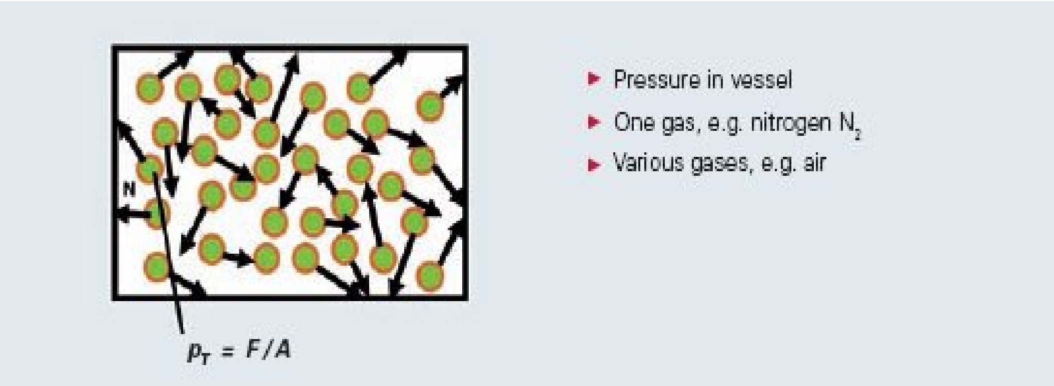


Figure 2.1 : Definition of total pressure

2.1.3 Partial pressure p_i

The partial pressure of a certain gas or vapor is the pressure which that gas or vapor would exert if it alone were present in the vessel. If the gas is made up of different types of gases, each of these gases will exert a pressure that corresponds to its concentration; this is called partial pressure. The sum of all partial pressures equals the total pressure [2].

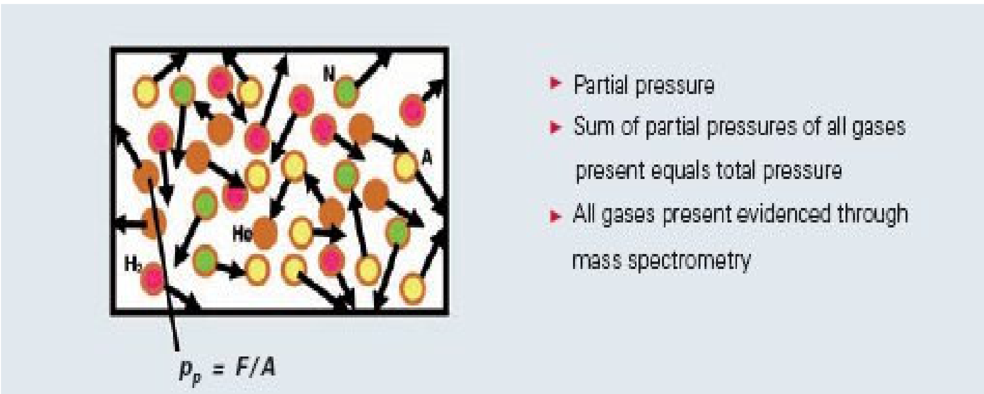


Figure 2.2 : Definition of partial pressure

Air is a good example of this: In addition to its main constituents of nitrogen, oxygen and water vapor, air also contains many trace gases.

Table 2.1: Total pressure and composition of air at 20 °C and 50 % relative humidity

Gas	Partial Pressure / mbar
Nitrogen	781.8
Oxygen	209.7
Water vapor	12
Argon	9.34
Carbon dioxide	$3.3 \cdot 10^{-1}$
Neon	$1.82 \cdot 10^{-2}$
Helium	$5.23 \cdot 10^{-3}$
Krypton	$1.15 \cdot 10^{-3}$
Hydrogen	$4.94 \cdot 10^{-3}$
Xenon	$8.7 \cdot 10^{-5}$
Total pressure	1,013

2.1.4 Saturation vapor pressure p_s

The pressure of the saturated vapor is referred to as saturation vapor pressure ps. Ps will be a function of temperature for any given substance.

2.1.5 Vapor pressure p_d

Partial pressure of those vapors which can be liquefied at the temperature of liquid nitrogen (LN2).

2.1.6 Standard pressure p_n

Standard Atmospheric Pressure (*atm*) is used as a reference for gas densities and volumes. The Standard Atmospheric Pressure is defined at sea-level at *273 ^{o} K (0 ^{o} C)* and is 1.01325 bar or 101325 Pa (absolute). The temperature of *293 ^{o} K (20 ^{o} C)* is also used [2].

2.1.7 Ultimate pressure p_{end}

The lowest pressure which can be achieved in a vacuum vessel. The so called ultimate pressure p_{end} depends not only on the pump's suction speed but also upon the vapor pressure p_d for the lubricants, sealants and propellants used in the pump. If a container is evacuated simply with an oilsealed rotary (positive displacement) vacuum pump, then the ultimate pressure which can be attained will be determined primarily by the vapor pressure of the pump oil being used and depending on the cleanliness of the vessel, also on the vapors released from the vessel walls and of course, on the leak tightness of the vacuum vessel itself [2].

2.1.8 Ambient (atmospheric) pressure p_{amb}

Atmospheric pressure is pressure in the surrounding air at - or "close" to - the surface of the earth. The atmospheric pressure vary with temperature and altitude above sea level [2].

2.1.9 Absolute pressure p_{abs}

Absolute pressure is simply the addition of the observed gage pressure plus the value of the local atmospheric pressure.

2.1.10 Gauge pressure (overpressure) p_e

A gauge is often used to measure the pressure difference between a system and the surrounding atmosphere. This pressure is often called the gauge pressure [2].

$$p_e = p_{abs} - p_{atm} \quad \textbf{(2.2)}$$

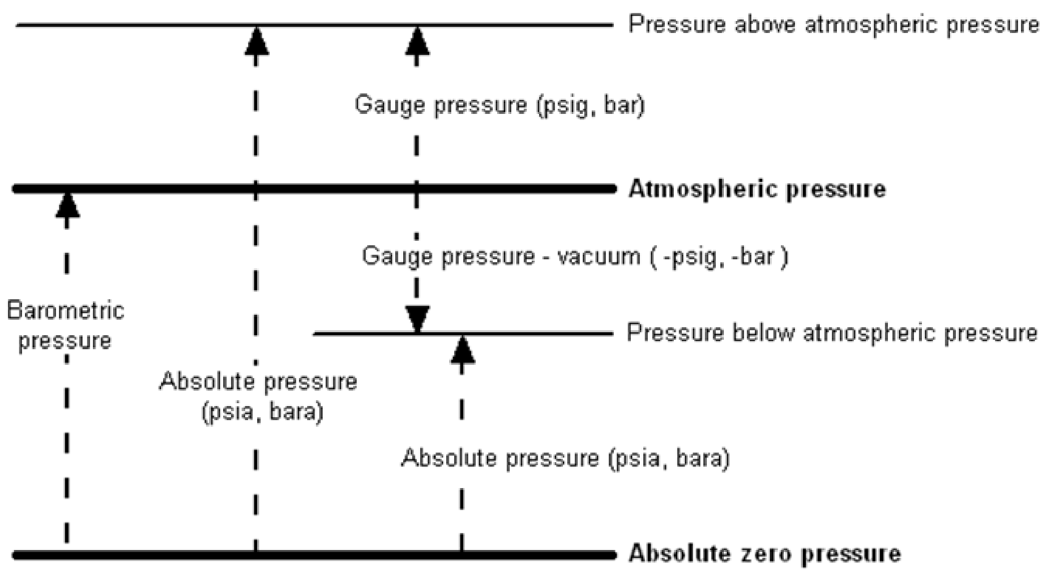


Figure 2.3 : Definition of gauge pressure

Here positive values for p_e will indicate overpressure or gauge pressure; negative values will characterize a vacuum.

Table 2.2: Conversion table for units of pressure

	Pa = N/m²	bar	mbar	µbar = dyn/cm²	Torr = mm Hg	micron µ = mTorr	atm	at	mm WS	psi = lbf/inch²	psf = lbf/ft²
Pa	1	$1 \cdot 10^{-5}$	$1 \cdot 10^{-2}$	10	$7.5 \cdot 10^{-3}$	7.5	$9.87 \cdot 10^{-6}$	$1.02 \cdot 10^{-5}$	0.102	$1.45 \cdot 10^{-4}$	$2.09 \cdot 10^{-2}$
bar	$1 \cdot 10^{5}$	1	$1 \cdot 10^{-3}$	$1 \cdot 10^{6}$	750	$7.5 \cdot 10^{5}$	0.987	1.02	$1.02 \cdot 10^{4}$	14.5	$2.09 \cdot 10^{3}$
mbar	100	$1 \cdot 10^{-3}$	1	1,000	0.75	750	$9.87 \cdot 10^{-4}$	$1.02 \cdot 10^{-3}$	10.2	$1.45 \cdot 10^{-2}$	2.09
µbar	0.1	$1 \cdot 10^{-6}$	$1 \cdot 10^{-3}$	1	$7.5 \cdot 10^{-4}$	0.75	$9.87 \cdot 10^{-7}$	$1.02 \cdot 10^{-6}$	$1.02 \cdot 10^{-2}$	$1.45 \cdot 10^{-5}$	$2.09 \cdot 10^{-3}$
Torr	$1.33 \cdot 10^{2}$	$1.33 \cdot 10^{-3}$	1.33	1,330	1	1,000	$1.32 \cdot 10^{-3}$	$1.36 \cdot 10^{-3}$	13.6	$1.93 \cdot 10^{-2}$	2.78
micron	0.133	$1.33 \cdot 10^{-6}$	$1.33 \cdot 10^{-3}$	1.33	$1 \cdot 10^{-3}$	1	$1.32 \cdot 10^{-6}$	$1.36 \cdot 10^{-6}$	$1.36 \cdot 10^{-2}$	$1.93 \cdot 10^{-5}$	$2.78 \cdot 10^{-3}$
atm	$1.01 \cdot 10^{5}$	1.013	1,013	$1.01 \cdot 10^{6}$	760	$7.6 \cdot 10^{5}$	1	1.03	$1.03 \cdot 10^{4}$	14.7	$2.12 \cdot 10^{3}$
at	$9.81 \cdot 10^{4}$	0.981	981	$9.81 \cdot 10^{5}$	735.6	$7.36 \cdot 10^{5}$	0.968	1	$1 \cdot 10^{-4}$	14.2	$2.04 \cdot 10^{3}$
mm WC	9.81	$9.81 \cdot 10^{-5}$	$9.81 \cdot 10^{-2}$	98.1	$7.36 \cdot 10^{-2}$	73.6	$9.68 \cdot 10^{-5}$	$1 \cdot 10^{-4}$	1	$1.42 \cdot 10^{-3}$	0.204
psi	$6.89 \cdot 10^{3}$	$6.89 \cdot 10^{-2}$	68.9	$6.89 \cdot 10^{4}$	51.71	$5.17 \cdot 10^{4}$	$6.8 \cdot 10^{-2}$	$7.02 \cdot 10^{-2}$	702	1	144
psf	47.8	$4.78 \cdot 10^{-4}$	0.478	478	0.359	359	$4.72 \cdot 10^{-4}$	$4.87 \cdot 10^{-4}$	4.87	$6.94 \cdot 10^{-3}$	1

2.1.11 Particle number density n (cm^3)

According to the kinetic gas theory the number n of the gas molecules, referenced to the volume, is dependent on pressure p and thermodynamic temperature T as expressed in the following:

$$p = \mathrm{n} \cdot \mathrm{k} \cdot \mathrm{T} \qquad \textbf{(2.3)}$$

n = particle number density

k = Boltzmann's constant (J/K)

At a certain temperature, therefore, the pressure exerted by a gas depends only on the particle number density and not on the nature of the gas. The nature of a gaseous particle is characterized, among other factors, by its mass m_T.

2.1.12 Gas density r (kg .m^3 / g. cm^3)

The product of the particle number density n and the particle mass m_T is the gas density r:

$$r = n \cdot m_T = kg.m^3 \qquad \textbf{(2.4)}$$

2.1.13 The ideal gas law

The following applies for gases: A volume of 22.414 liters (mol volume) at a temperature of 273.15 K (standard temperature = 0 °C) and a pressure of 101,325 pa (standard pressure) contains 6.02 times 10^{23} particles (Avogadro's number). The mass of the gas thus enclosed is its molecular weight in grams. The general gas equation describes the state of a gas as a function of pressure, temperature and volume.

General gas equation;

$$p \times V = \frac{m}{M} \times R \times T = n \times V \times k \times T \qquad \textbf{(2.5)}$$

Thus gas pressure;

$$p = n \times k \times T \qquad \textbf{(2.6)}$$

Where;

P = pressure [Pa; N / m^2]

V = volume [m^3]

M = Mass [kg]

M = molar mass [kg / kmol]

R = general gas constant R = 8.314510 kJ / (kmol K)

T = thermodynamic temperature [K]

N = molecular number density [1 / m^3]

K = Boltzmann's constant k = 1.380 · 10^{-23} J/K

2.1.14 Volume V (l, m3, cm3)

The term volume is used to designate;

a) the purely geometric, usually predetermined, volumetric content of a vacuum chamber or a complete vacuum system including all the piping and connecting spaces (this volume can be calculated).

b) the pressure-dependent volume of a gas or vapor which, for example, is moved by a pump or absorbed by an adsorption agent.

2.1.15 Volumetric flow (flow volume) q_v (l/s, m^3/h, cm^3/s)

The term "flow volume" designates the volume of the gas which flows through a piping element within a unit of time, at the pressure and temperature prevailing at the particular moment. Here one must realize that, although volumetric flow may be identical, the number of molecules moved may differ, depending on the pressure and temperature.

The volumetric flow rate can be calculated as the product of the cross- sectional area (A) for flow and the average flow velocity (v).

$$q_v = V \times A = m / s^2 .m^2 \quad \textbf{(2.7)}$$

2.1.16 Mass flow q_m (kg/h, g/s),

The *mass flow rate* of a system is a measure of the mass of fluid passing a point in the system per unit time. The mass flow rate is related to the volumetric flow rate as shown below;

$$q_m = q_v \times r \quad \textbf{(2.8)}$$

$$q_m = \frac{m}{t} = \frac{kg}{h} \quad \textbf{(2.9)}$$

2.1.17 PV flow q_{pV} (mbar . l . s^{-1}).

PV flow is the product of the pressure and volume of a quantity of gas flowing through a piping element, divided by time, i.e.:

$$q_{pv} = \frac{P \times V}{t} = \frac{m \times R \times T}{t \times M} \quad \textbf{(2.10)}$$

2.1.18 Pumping speed S (l/s, m3/h, cm3/s)

The pumping speed is the volumetric flow through the pump's intake port.

$$S = \frac{dV}{dt} \quad \textbf{(2.11)}$$

If S remains constant during the pumping process, then one can use the difference quotient instead of the differential quotient:

$$S = \frac{\Delta V}{\Delta t} \quad \textbf{(2.12)}$$

2.1.19 Mean free path

The gas molecules fly about and among each other, at every possible velocity, and bombard both the vessel walls and collide (elastically) with each other. This motion of the gas molecules is described numerically with the assistance of the kinetic theory of gases. The mean free path is the mean path length that a molecule traverses between two successive impacts with other molecules. It depends upon molecular diameter d_m and temperature T in accordance with the following equation [2];

Mean free path;

$$\bar{l} \times p = \frac{k \times T}{\Pi \times \sqrt{2} \times d_m^{\ 2}} \quad \textbf{(2.13)}$$

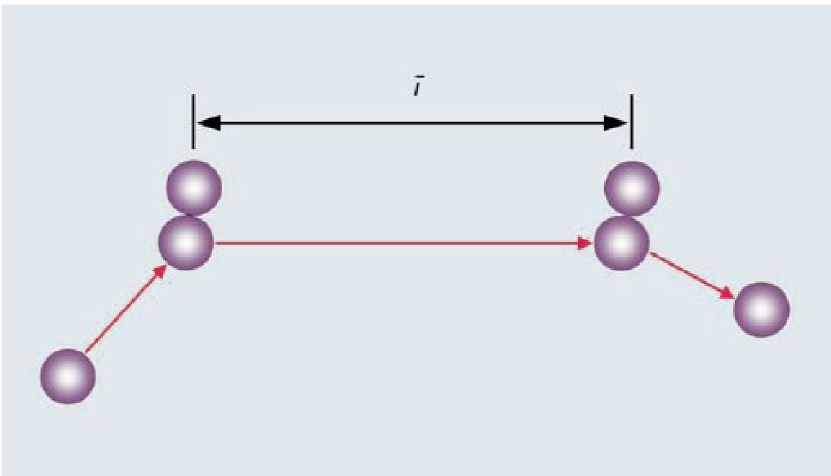


Figure 2.4 : Mean free paths

Table 2.3: Mean free paths of various gases at 0 °C

Gas	$\bar{l}p/(m \cdot Pa)$	Gas	$\bar{l}p/(m \cdot Pa)$
H_2	$11.5 \cdot 10^{-3}$	Xe	$3.6 \cdot 10^{-3}$
N_2	$5.9 \cdot 10^{-3}$	Hg	$3.1 \cdot 10^{-3}$
He	$17.5 \cdot 10^{-3}$	CO	$6.0 \cdot 10^{-3}$
Ne	$12.7 \cdot 10^{-3}$	CO_2	$4.0 \cdot 10^{-3}$
Ar	$6.4 \cdot 10^{-3}$	HCl	$4.4 \cdot 10^{-3}$
Air	$6.65 \cdot 10^{-3}$	NH_3	$4.3 \cdot 10^{-3}$
Kr	$4.9 \cdot 10^{-3}$	Cl_2	$2.8 \cdot 10^{-3}$

At atmospheric pressure, the gas molecules are very close together and as they are in constant motion, the distance between molecule-to-molecule collisions is very short. This distance is known as mean free path. As the vacuum pump removes molecules, the distance between collisions becomes greater and greater. As molecules are removed there are fewer other molecules for a given molecule to collide with the distance becomes longer and longer as the pressure is reduced [2].

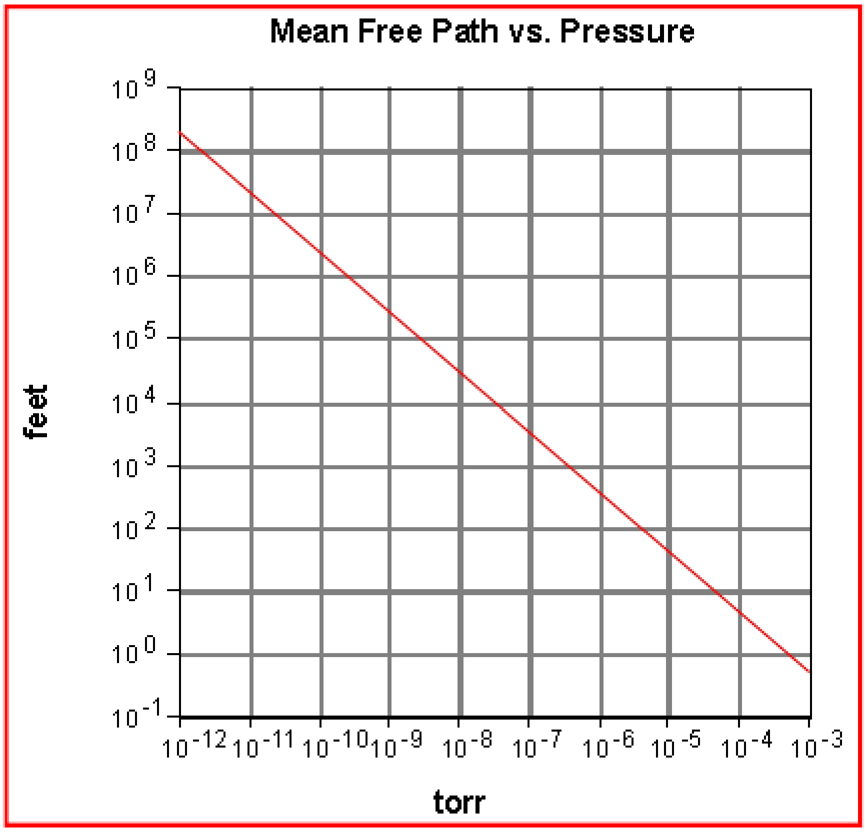


Figure 2.5 : Mean free paths of various gases at 0 °C

2.1.20 Knudsen Number

The Knudsen number (*Kn*) is a dimensionless number defined as the ratio of the molecular mean free path length to a representative physical length scale. This length scale could be, for example, the radius of a body in a fluid [2].The Knudsen number is a dimensionless number defined as:

$$K_n = \frac{\lambda}{L} \tag{2.14}$$

where;

- λ = mean free path [L^1]

- L = representative physical length scale [L^1].

For an ideal gas, the mean free path may be readily calculated so that:

$$K_n = \frac{k_B T}{\sqrt{2} \cdot \Pi \cdot \sigma^2 \cdot p \cdot L} \qquad \textbf{(2.15)}$$

where;

- k_B is the Boltzmann constant (1.3806504(24) × 10^{-23} J/K in SI units), [M^1 L^2 T^{-2} θ^{-1}]
- T is the thermodynamic temperature, [θ^1]
- σ is the particle hard shell diameter, [L^1]
- p is the total pressure, [M^1 L^{-1} T^{-2}].

2.1.21 Conductivity

Vacuum chambers are connected to a vacuum pump via piping. Flow resistance occurs as a result of external friction (gas molecules / wall surface) and internal friction (gas molecules / gas molecule „viscosity”). This flow resistance shows itself in the form of the volume flow rate, or pumping speed. In vacuum technology, it is customary to use the reciprocal, the conductivity of piping *L*, instead of flow resistance *W* [3]. This is expressed in [l / s] or [m^3 /h].

Parallel connection conductivities;

$$L_1 + L_2 + L_3 + ... = L_{gas} \qquad \textbf{(2.16)}$$

And if connected in series, the reciprocals are added:

Series connection conductivities;

$$\frac{1}{L_1} + \frac{1}{L_2} .... + \frac{1}{L_N} = \frac{1}{L_{gas}} \qquad \textbf{(2.17)}$$

2.1.22 Outgassing (mbar . lt)

The term outgassing refers to the liberation of gases and vapors from the walls of a vacuum chamber or other components on the inside of a vacuum system. This quantity of gas is also characterized by the product of p .V, where V is the volume of the vessel into which the gases are liberated, and by p, or better Δp, the increase in pressure resulting from the introduction of gases into this volume [2].

2.1.23 Pump throughput q_{pV}

The pumping capacity (throughput) for a pump is equal either to the mass flow through the pump intake port:

$$q_m = \frac{m}{t} \qquad \textbf{(2.18)}$$

or to the pV flow through the pump's intake port:

$$q_{pv} = \frac{P \times V}{t} \qquad \textbf{(2.19)}$$

It is normally specified in mbar . l . s^{-1}. Here p is the pressure on the intake side of the pump. If p and V are constant at the intake side of the pump, the throughput of this pump can be expressed with the simple equation;

$$q_{pv} = p \times S \qquad \textbf{(2.20)}$$

where S is the pumping speed of the pump at intake pressure of p. (The throughput of a pump is often indicated with Q, as well.) The concept of pump throughput is of major significance in practice and should not be confused with the pumping speed. The pump throughput is the quantity of gas moved by the pump over a unit of time, expressed in mbar . l/s; the pumping speed is the transportation capacity which the pump makes available within a specific unit of time, measured in m^3/h or l/s [2].

2.2 Gas Laws and Models

2.2.1 Continuum theory

Gas is pourable (fluid) and flows in a way similar to a liquid. The continuum theory and the summarization of the gas laws which follows are based on experience and can explain all the processes in gases near atmospheric pressure. Only after it became possible using ever better vacuum pumps to dilute the air to the extent that the mean free path rose far beyond the dimensions of the vessel were more far-reaching assumptions necessary; these culminated in the kinetic gas theory. The kinetic gas theory applies throughout the entire pressure range; the continuum theory represents the (historically older) special case in the gas laws where atmospheric conditions prevail.

Summary of the most important gas laws (continuum theory):

Boyle-Mariotte Law

$$p \times V = const \quad \textbf{(2.21)}$$

for T = constant (isotherm)

Gay-Lussac's Law (Charles' Law)

$$\mathrm{V} = \mathrm{V0}(1 + \mathrm{b} \cdot \mathrm{t}) \quad \textbf{(2.22)}$$

for p = constant (isobar)

Amonton's Law

$$\mathrm{p} = \mathrm{p0}(1 + \mathrm{g} \cdot \mathrm{t}) \quad \textbf{(2.23)}$$

for V = constant (isochor)

Poisson's Law

$$p \times V^k = const(adiabatic) \quad \textbf{(2.24)}$$

Avogadro's Law

$$\frac{m_1}{V_1} \div \frac{m_2}{V_2} = M_1 \div M_2 \quad \textbf{(2.25)}$$

Ideal gas Law

$$p \times V = \frac{m}{M} \times R \times T = \nu \times R \times T \quad \textbf{(2.26)}$$

Also: Equation of state for ideal gases (from the continuum theory)

Van der Waals' Equation

$$(p + \frac{a}{V_m{}^2}) \times (V_m - b) = R \times T \quad \textbf{(2.27)}$$

a, b = constants (internal pressure, covolumes)

V_m = Molar volume

Also: Equation of state for real gases;

Clausius-Clapeyron Equation

$$L = T \times \frac{dp}{dT} \times (V_{m,v} - V_{m,l}) \quad \textbf{(2.28)}$$

L = Enthalpy of evaporation,

T = Evaporation temperature,

Vm,v, Vm,l = Molar volumes of vapor or liquid

2.2.2 Kinetic gas theory

With the acceptance of the atomic view of the world - accompanied by the necessity to explain reactions in extremely dilute gases (where the continuum theory fails) the kinetic gas theory was developed. Using this it is possible not only to derive the ideal gas law in another manner but also to calculate many other quantities involved with the kinetics of gases such as collision rates, mean free path lengths, monolayer formation time, diffusion constants and many other quantities.

Kinetic gas theory is the theory that gases are made up of a large number of small particles (atoms or molecules), all of which are in constant, random motion. The rapidly moving particles constantly collide with each other and with the walls of the container. Kinetic theory explains macroscopic properties of gases, such as pressure, temperature, or volume, by considering their molecular composition and motion. Essentially, the theory posits that pressure is due not to static repulsion between molecules but due to collisions between molecules moving at different velocities [4].

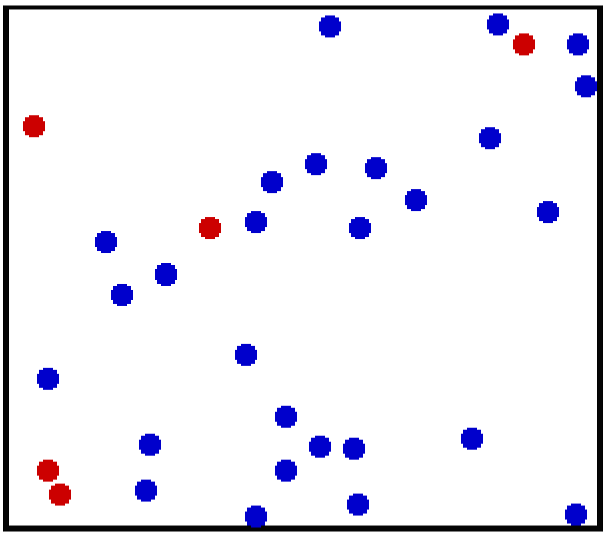


Figure 2.6 : Mean free paths of various gases at 0 °C

Model concepts and basic assumptions:

1. Atoms/molecules are points.
2. Forces are transmitted from one to another only by collision.
3. The collisions are elastic.
4. Molecular disorder (randomness) prevails.

2.3 The Pressure Ranges in Vacuum Technology and Their Characterization

It is common in vacuum technology to subdivide its wide overall pressure range which spans more than 16 powers of ten into smaller individual regimes. These are generally defined as follows:

- Rough (low) vacuum (RV) 1000 - 1 mbar
- Medium vacuum (MV) 1 – 10^{-3} mbar
- High vacuum (HV) 10^{-3} – 10^{-7} mbar
- Ultrahigh vacuum (UHV) 10^{-7} - (10^{-18}) mbar

The pressure range of vacuum comprises the interval of 0 – 1 bar. A distinction is made between the following pressure ranges [5]:

Table 2.4: Pressure ranges / Molecular number density

Pressure Range	Pressure / mbar	Molecular Number Density / cm^{-3}
Low vacuum	10^{3} – 10^{0}	$2.65 \cdot 10^{19}$ – $2.65 \cdot 10^{16}$
Medium vacuum	10^{0} – 10^{-3}	$2.65 \cdot 10^{16}$ – $2.65 \cdot 10^{13}$
High vacuum	10^{-3} – 10^{-7}	$2.65 \cdot 10^{13}$ – $2.65 \cdot 10^{9}$
Ultra high vacuum	10^{-7} – 10^{-12}	$2.65 \cdot 10^{9}$ – $2.65 \cdot 10^{4}$

2.4 Types of Flow

2.4.1 Viscous or continuum flow

This will be found almost exclusively in the rough (low) vacuum range. The character of this type of flow is determined by the interaction of the molecules. Consequently internal friction, the viscosity of the flowing substance, is a major factor. The flow in a viscous flow can be either laminar or turbulent flow. If vortex motion appears in the streaming process, one speaks of turbulent flow. If various layers of the flowing medium slide one over the other, then the term laminar flow or layer flux may be applied. Laminar flow in circular tubes with parabolic velocity distribution is known as Poiseuille flow. This special case is found frequently in vacuum technology. Viscous flow will generally be found where the molecules mean free path is considerably shorter than the diameter of the pipe: l << d.

A characteristic quantity describing the viscous flow state is the dimensionless Reynolds number Re. Re is the product of the pipe diameter, flow velocity, density and reciprocal value of the viscosity (internal friction) of the gas which is flowing. Flow is turbulent where Re > 2200, laminar where Re < 2200.

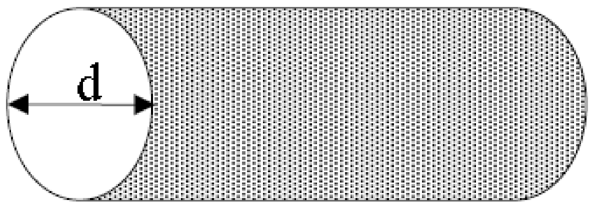


Figure 2.7 : Pressure ranges / Molecular number density

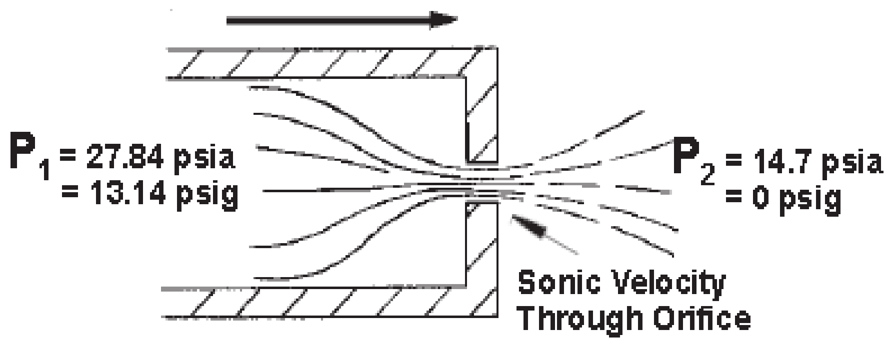
$$K_n = \frac{\bar{l}}{d} \qquad \textbf{(2.29)}$$

Kn < 0.01 Viscous Flow - gas flow determined by gas-gas collisions.The phenomenon of choked flow may also be observed in the viscous flow situation. It plays a part when venting and evacuating a vacuum vessel and where there are leaks [6].

2.4.2 Choked flow

Choked flow of a fluid is a fluid dynamic condition caused by the Venturi effect. When a flowing fluid at a certain pressure and temperature flows through a restriction (such as the hole in an orifice plate or a valve in a pipe) into a lower pressure environment, under the conservation of mass the fluid velocity must increase for initially subsonic upstream conditions as it flows through the smaller cross-sectional area of the restriction. At the same time, the Venturi effect causes the static pressure to decrease. Choked flow is a limiting condition which occurs when the mass flow rate will not increase with a further decrease in the downstream pressure environment while upstream pressure is fixed [7]. A greatly misunderstood and misapplied notion is that of “choked flow”, also referred to as “critical flow”. In gas flow through an orifice there is an occasion where the gas velocity reaches sonic conditions. This occurs for air flow when the absolute pressure ratio is .528, i.e. when the downstream absolute pressure (P2) is 52.8% of the upstream absolute pressure (P1).

For air flow through an orifice with an inlet air temperature of 68°F (20°C) the choked (sonic) velocity is 1129 ft/sec (344,11 m/sec= sound velocity). Once sonic velocity is achieved in orifice air flow (P2/P1 = .528), it is easy to "assume" that the mass flow rate is constant for all pressure ratios less than .528; i.e. P2/P1 ≤ .528. For example, when P2 is 14.7 psia and P1 is 27.84 psia, sonic velocity occurs through the orifice. As P1 further increases there is no further increase in the velocity of the air flowing through the orifice [7].



Conditions for the onset of sonic velocity in orifice air flow.

Figure 2.8 : Sonic velocity in orifice air flow

The mass flow rate through an orifice is a function of three basic parameters.

- Velocity
- Density
- Orifice Area

When the air velocity reaches sonic velocity (P2/P1 ≤ .528) further increases in P1 (upstream pressure) do not cause any further increase in the air velocity through the orifice. Consequently it is wrongly concluded that the mass flow rate also does not increase [7]. As the air pressure (P1) increases, the density of the air also increases; and since the mass flow rate is also a function of density, the mass flow rate increases linearly with pressure (P1).

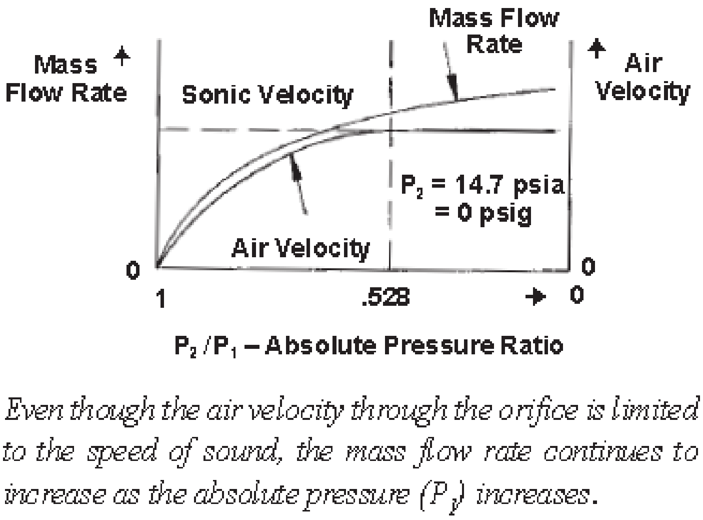


Figure 2.9 : Mass flow rate change in choked flow

The parameter that becomes "choked" or "limited" is the velocity of the air. It is more accurate to use the term "choked velocity" rather than "choked flow" when the absolute pressure ratio of air through an orifice is ≤ .528.

In the case of vacuum conditions on the outlet of an orifice and where the inlet is at ambient atmospheric pressure, both the air velocity and the mass flow rate become choked (limited) when sonic velocity is achieved through the orifice [7].

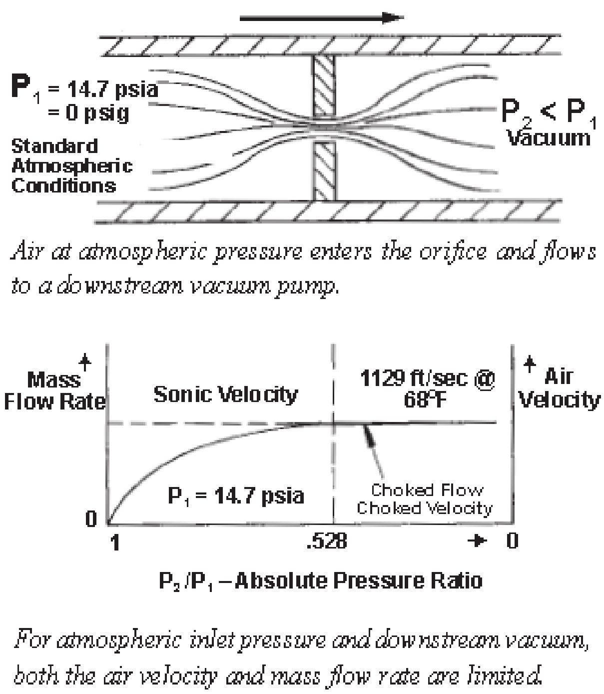


Figure 2.10 : Choked flow in vacuum conditions

Table 2.5: Sonic velocity conditions

Sonic Velocity Conditions – Air Flow			
Inlet Pressure		Outlet Pressure For Sonic Velocity	
Gage Pressure psig	Absolute Pressure psia	Absolute Pressure psia	Gage Pressure psig
100	114.7	≤ 60.56	≤ 45.86
90	104.7	≤ 55.28	≤ 40.58
80	94.7	≤ 50.00	≤ 35.30
70	84.7	≤ 44.72	≤ 30.02
60	74.7	≤ 39.44	≤ 24.74
50	64.7	≤ 34.16	≤ 19.46
40	54.7	≤ 28.88	≤ 14.18
30	44.7	≤ 23.60	≤ 8.90
20	34.7	≤ 18.32	≤ 3.62
15	29.7	≤ 15.68	≤ .98
14.7	29.4	≤ 15.52	≤ .82
10	24.7	≤ 13.08	≤ -1.62
5	19.7	≤ 10.40	≤ -4.30
1	15.7	≤ 8.29	≤ -6.47
0	14.7	≤ 7.76	≤ -6.94

Temperature 68°F

2.4.3 Molecular flow

Molecular flow prevails in the high and ultrahigh vacuum ranges. In these regimes the molecules can move freely, without any mutual interference. Molecular flow is present where the mean free path length for a particle is very much larger than the diameter of the pipe: l >> d.

Kn > 1.0 Molecular Flow - gas flow determined by gas-wall collisions [6].

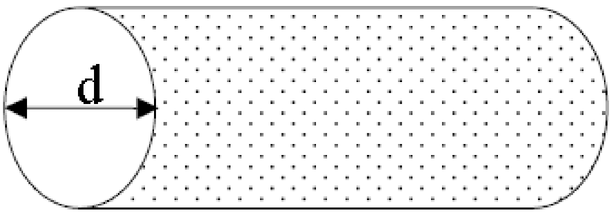


Figure 2.11 : Molecular Flow

2.4.4 Knudsen flow (Transitional)

The transitional range between viscous flow and molecular flow is known as Knudsen flow. It is prevalent in the medium vacuum range: l ~ d. The product of pressure p and pipe diameter d for a particular gas at a certain temperature can serve as a characterizing quantity for the various types of flow.

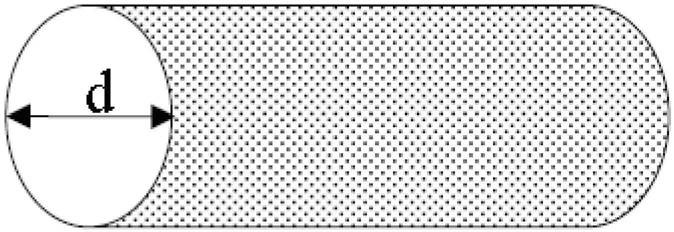


Figure 2.12 : Knudsen Flow. 1 > Kn > 0.01 Transition Flow

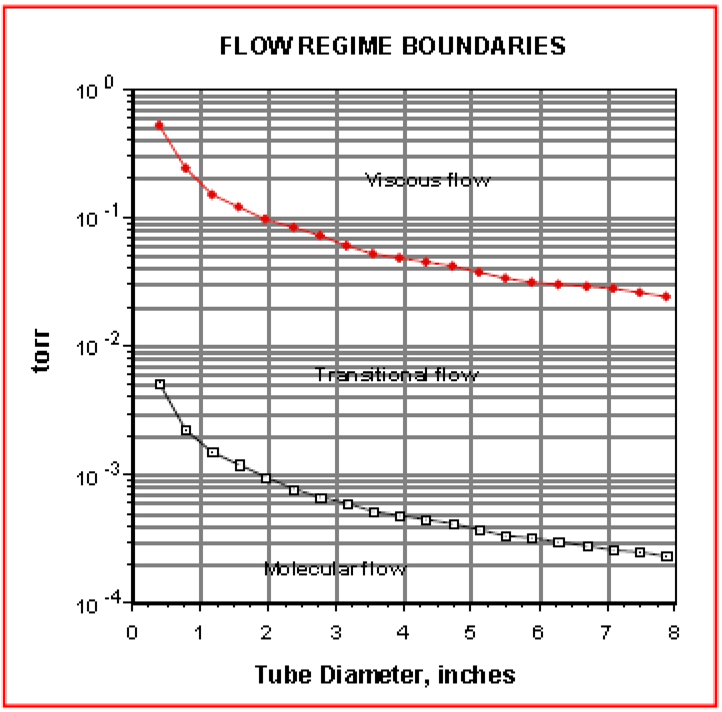


Figure 2.13 : Flow regimes as a function of pressure and chamber dimensions

3. ENGINE VACUUM SYSTEMS

An engine generates vacuum when running, this vacuum is used to operate several accessories from the temperature control system (heater, air conditioner) to the emission control systems;

- To decrease brake pedal force to make driver feel more comfortable during braking.
- To actuate engine pneumatic valves to contribute engine emission reduction.
- To operate air conditioner heater.

Engine vacuum system layout can be shown in general as;

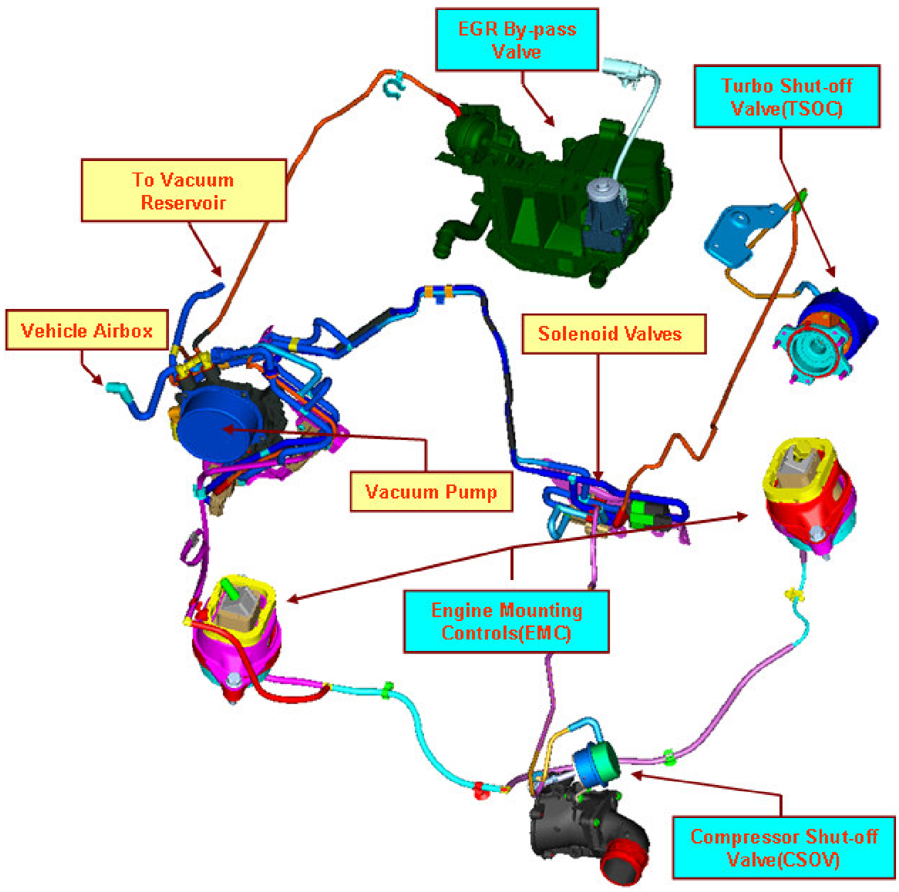


Figure 3.1 : Engine vacuum system schematic view

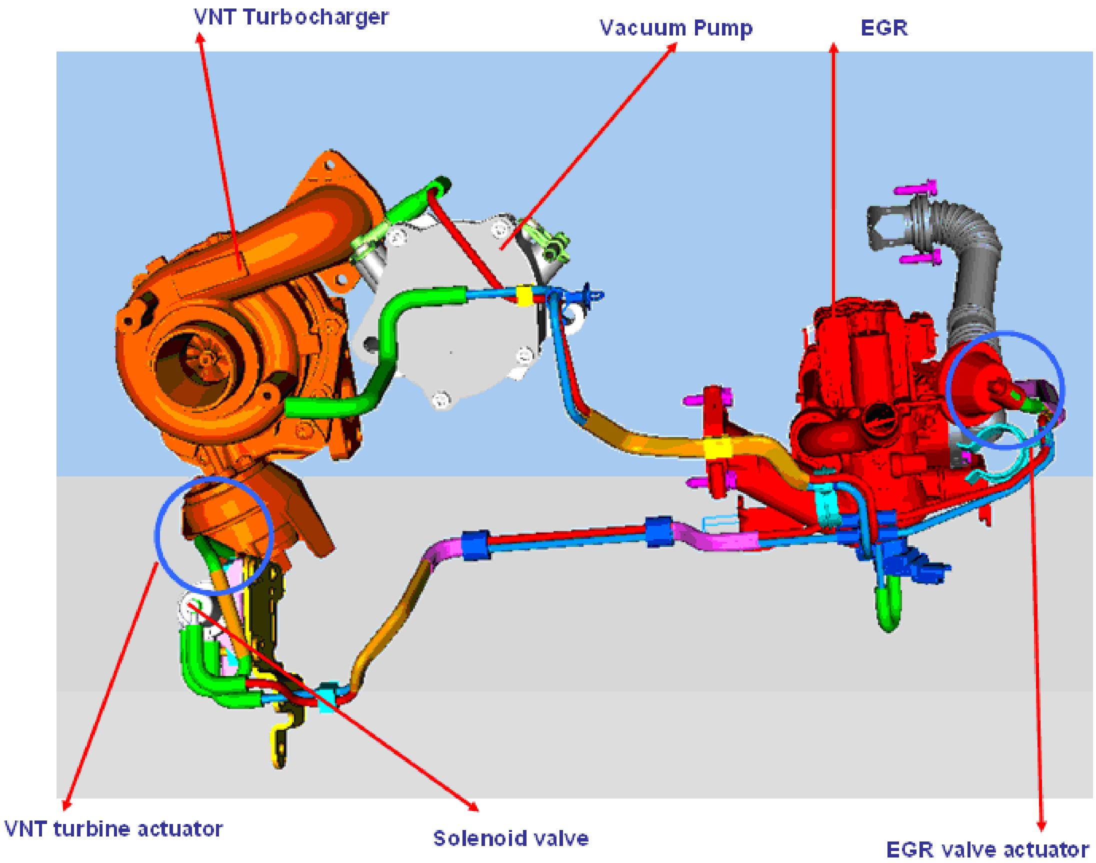


Figure 3.2 : A simple engine vacuum system

3.1 Vacuum Pumps

A vacuum pump is a device that removes gas molecules from a sealed volume in order to leave behind a partial vacuum. Vacuum pump converts the mechanical input energy of a rotating shaft into pneumatic energy by evacuating the air contained within a system. The internal pressure level thus becomes lower than that of the outside atmosphere. The amount of energy produced depends on the volume evacuated and the pressure difference produced [8].

Mechanical vacuum pumps use the same pumping mechanism as air compressors, except that the unit is installed so that air is drawn from a closed volume and exhausted to the atmosphere. A major difference between a vacuum pump and other types of pumps is that the pressure driving the air into the pump is below atmospheric and becomes vanishingly small at higher vacuum levels. Other differences between air compressors and vacuum pumps are:

- The maximum pressure difference produced by pump action can never be higher than 1 atm since this represents a perfect vacuum.
- The mass of air drawn into the pump on each suction stroke, and hence the absolute pressure change, decreases as the vacuum level increases.

3.1.1 Classification of vacuum pumps

Vacuum pumps are used to reduce the gas pressure in a certain volume and thus the gas density. Consequently consider the gas particles need to be removed from the volume. Basically differentiation is made between two classes of vacuum pumps:

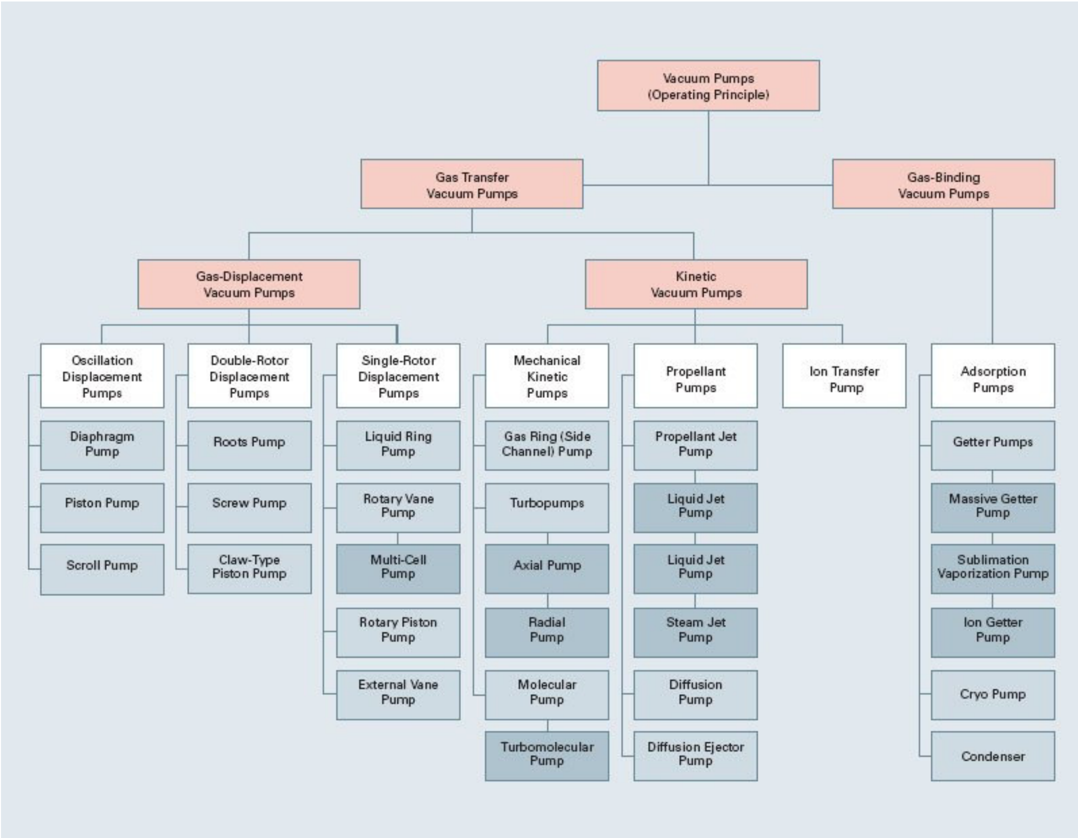
a) Vacuum pumps where - via one or several compression stages - the gas particles are removed from the volume which is to be pumped and ejected into the atmosphere (compression pumps). The gas particles are pumped by means of displacement or pulse transfer.

b) Vacuum pumps where the gas particles which are to be removed condense on or are bonded by other means (e.g. chemically) to a solid surface, which often is part of the boundary forming volume itself.

A classification which is more in line with the state-of-the-art and practical applications makes a difference between the following types of pumps, of which the first three classes belong to the compression pumps and where the two remaining classes belong to the condensation and getter pumps:

1. Pumps which operate with periodically increasing and decreasing pump cham-ber volumes (rotary vane and rotary plunger pumps; also trochoid pumps).
2. Pumps which transport quantities of gas from the low pressure side to the high pressure side without changing the volume of the pumping chamber (Roots pumps, turbomolecular pumps).
3. Pumps where the pumping effect is based mainly on the diffusion of gases into a gas-free high speed vapor jet (vapor pumps).
4. Pumps which pump vapors by means of condensation (condensers) and pumps which pump permanent gases by way of condensation at very low temperatures (cryopumps).
5. Pumps which bond or incorporate gases by adsorption or absorption to surfaces which are substantially free of gases (sorption pumps) [8].

Figure 3.3 : Classification of vacuum pumps



3.1.2 Vacuum pump selection

The first major step in selecting the right vacuum pump is to compare application vacuum requirements with the maximum vacuum ratings of commercial pumps. At low levels, there is a wide choice of pumps. But as vacuum level increases, the choice narrows, sometimes to the point where only one type of pump may be available.

To calculate a system's vacuum needs, consider all work devices to be driven. The working vacuum of the devices can be determined by calculations based on handbook formulas, theoretical data, catalog information, performance curves, or tests made with prototype systems. Once you know the vacuum required, you can begin looking for pumps that can accommodate application requirements. The maximum vacuum rating for a pump is commonly expressed for either continuous or intermittent duty cycles, and can be obtained from pump manufacturers. Because the maximum theoretical vacuum at sea level is 1 atm, actual pump capabilities are based on and compared to this theoretical value. Depending on pump design, the vacuum limit ranges 0.95-1 atm or about 93% or 98% of the maximum theoretical value. For some pump types, the maximum vacuum rating will be based on this practical upper limit. For others, where heat dissipation is a problem, the maximum vacuum rating might also take into account allowable temperature rise [8].

3.1.3 Vacuum at high altitudes

Atmospheric pressure determines the maximum vacuum force that can be achieved. And standard atmospheric pressure at sea level is 1 atm. But what happens at locations a mile above sea level? The maximum vacuum that can be achieved in locations above sea level will be less than 1 atm. The force will be limited by the ambient atmospheric pressure. Vacuum pumps have maximum vacuum ratings based on sea level conditions and must be re-rated for operation at higher elevations.

First, determine the local atmospheric pressure. A rule of thumb is that for every 304.8 m. of altitude above sea level, atmospheric pressure drops by 0.03 bar. Using rounded-off figures, for a city at an elevation of 1524 m, the atmospheric pressure is about 0.846 bar.

To adjust a pump rating, think of that rating as a percentage of atmospheric pressure at sea level. If a pump is rated for 0.846 bar , it can achieve 83.4% of a sea level perfect vacuum. At a 1524 m elevation, that same pump can achieve 83.4% of 0.846 bar- or a vacuum of 0.706 bar.

3.1.4 Vacuum pumps in automotive

Vacuum may be used to power, or provide assistance to mechanical devices. In diesel engine motor vehicles, a pump fitted on the engine (usually on the camshaft) is used to produce vacuum. In petrol engines, instead, vacuum is obtained as a side effect of the operation of the engine and the flow restriction created by the throttle plate. Petrol engines have a throttle butterfly, which restricts airflow into the engine, and controls engine speed, the engine draws the air into the cylinders past this butterfly and this creates a vacuum due to the restriction not allowing enough air to enter the intake manifold during low engine speeds. At high engine speeds with a wide-open throttle there is no vacuum in the intake manifold as there is minimal restriction. A diesel engine has no throttle butterfly or restriction to airflow, so NO vacuum is created in the intake manifold; this is due to a diesel engine's speed being controlled by the amount of diesel fuel entering the cylinder for a given amount of air. Therefore, a diesel engine requires an external vacuum pump to create the vacuum for operation of brake boosters etc. In automotive, the mostly used vacuum pumps are rotary vane pumps, which are a kind of positive displacement pump and working in low vacuum conditions (1-1000 mbar).

The function of a vacuum pump in automotive;

- To create the vacuum required from servo brake circuit,
- To supply vacuum to the auxiliary circuits (vacuum actuators) such as turbo shut off valves & west gate valve, compressor shut off valve, EGR by pass valve and engine active mounts' valve.

Gasoline engines create vacuum as a byproduct of normal operating. During engine operation, when the pistons inside the cylinders move down on the intake stroke, the combustion chamber area (volume) inside the cylinders is greatly. This action decreases the pressure inside the combustion chamber creating a partial void or vacuum (an area with fewer air molecules per square inch than the surrounding outside air). A mixture of air and fuel from the carburetor/fuel injectors and intake manifold rushes into the cylinders through the intake valve to fill this partial void. The continuous movement of the pistons within the cylinders creates a need for a constant supply of air and fuel to fill the partial void created inside the cylinders on the intake strokes. The continuous flow of air and fuel through the intake manifold creates a low pressure (vacuum) inside the manifold. The low pressure (vacuum) in the intake manifold exists as long as the engine is in operation. This is how vacuum is generated in an internal combustion (gasoline) engine.

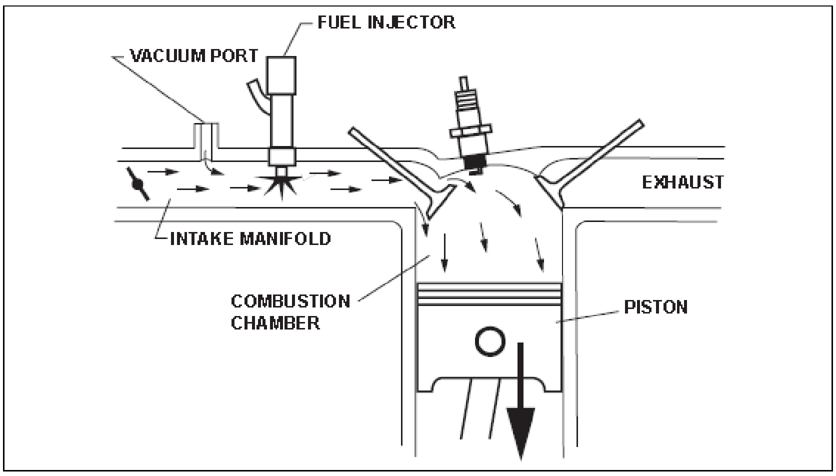


Figure 3.4 : Vacuum in gasoline engine

In a gasoline-operated engine/vehicle the vacuum that is generated in the intake manifold is used to operate a number of vacuum-actuated or vacuum-controlled devices.

A gasoline engine that uses a carburettor with butterfly valves is capable of providing enough suction pressure and does not require a pump. However, a gasoline engine that uses a fuel injection system instead of a carburettor does not create enough pressure and also needs a vacuum pump. Cars fitted with a petrol engine in combination with an automatic gearbox which comply with the EU 4 Standard have an electric vacuum pump. This is used for assisting the brake servo unit. In this engine-gearbox combination, the throttle valve is opened particularly wide when the engine is started from cold, as well as when it is idling with a drive position engaged and the brakes applied. This is intended to reduce the pressure drop in intake manifold. The reasons for this greater opening of the throttle valve is the heating-up phase of the catalytic converter after a cold engine start, as required by the EU 4 emission standard, and compensating for the greater torque friction (resistance in torque converter) which exists under the conditions mentioned above.

The diesel vacuum pump is an essential component for power brakes in diesel engine vehicles. By extracting air from the brake booster, it helps reduce the effort required to depress the brake pedal. Some pumps use a vane and rotor mechanism to create suction pressure, while others use a rubber diaphragm and plunger rod. Since both pump types operate continuously while the engine is running, they will eventually wear out and need to be replaced. Changing a diesel vacuum pump is a simple task that can be done by most people with a spanner set or an adjustable wrench.

Most drivers do not have the luxury of using a braking system powered by pressurized air or electric motor. The system used in most vehicles has brakes which are operated by hydraulic pressure, created when the brake pedal is depressed. A significant force needs to be applied to the pedal to build up this pressure, which making driving in busy traffic exhausting work because of frequent stopping. To reduce this effort, a component called a brake booster is installed in most vehicles.

The brake booster is a drum-shaped component which has two internal sections that are separated by a rubber diaphragm. When the brakes are not being used, both sections are kept at low pressure by suction provided by either the engine or a pump. When the brake pedal is depressed, a valve opens that allows air into the rear section of the booster. This extra pressure helps push the master brake cylinder piston forward and reduces the effort required to operate the brakes. When the brake pedal is released, the valve closes and air inside the rear section is removed through the vacuum hose.

A diesel engine does not create enough suction pressure by itself for the brake booster to function. Instead, engine power is used to drive a diesel vacuum pump to create the required pressure.

The diesel vacuum pump is a small component that weights around two pounds. The mounting flange and cover are made from die cast aluminum or steel. They are sealed together with a gasket and several bolts to form an airtight unit. The top of the pump is disc-shaped and measures several inches in diameter. There is a large air inlet for connecting the vacuum hose from the brake booster, and an exhaust port on the rear of the flange. A small oil inlet may also be present for adding engine oil to the pump. The pump is driven mechanically through either a gear drive or a port on the rear of the mounting flange.

Most problems in automobile vacuum systems result from disconnected or leaking hoses, bad connectors, defective motor diaphragms or valves. Pinched vacuum hoses or clogged valves may also restrict vacuum flow. The first step in checking a vacuum system is to visually check all vacuum hoses, lines and connectors for cuts, cracks and splits. Check for collapsed or pinched vacuum hoses. Vacuum hoses have a tendency to deteriorate because of harsh conditions within the engine compartment.

3.1.5 Rotary vane pumps- working principle

Rotary vane vacuum pumps are the most common pumps that are used in automotive. A rotary vane pump is a positive-displacement pump that consists of vanes mounted to a rotor that rotates inside of a cavity. In some cases these vanes can be variable length and/or tensioned to maintain contact with the walls as the pump rotates. The most simple vane pump is a circular rotor rotating inside of a larger circular cavity. The centers of these two circles are offset, causing eccentricity [9].

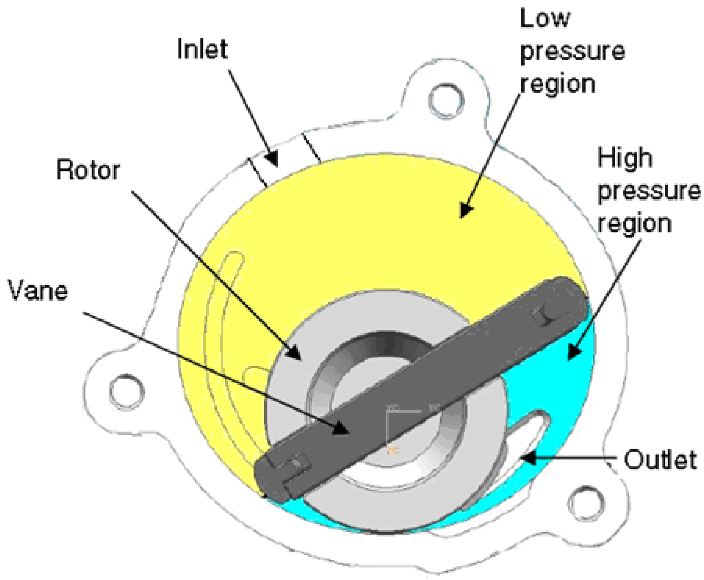


Figure 3.5 : Vacuum pump operating principle

Vanes are allowed to slide into and out of the rotor and seal on all edges, creating vane chambers that do the pumping work. On the intake side of the pump, the vane chambers are increasing in volume. These increasing volume vane chambers are filled with fluid forced in by the inlet pressure. Often this inlet pressure is nothing more than pressure from the atmosphere. On the discharge side of the pump, the vane chambers are decreasing in volume, forcing fluid out of the pump. The eccentrically mounted rotor compresses the gas and sweeps it toward the discharge port. When gas pressure exceeds atmospheric pressure, the exhaust valve opens and gas is expelled [9].

Rotary vane vacuum pumps can be mostly classified into 2 groups in terms of vane number ;

- Single vane vacuum pump
- Multi vane vacuum pump

According to driving type, vacuum pumps are mainly driven by camshaft, belt, gear or electric motor.

3.1.6 Design guide of vacuum pump

Vacuum pump is a device which gets drive from engine cam shaft, belt, electric motor etc. The main function of vacuum pump is to evacuate the air from the tank, thus creating vacuum, which can be used for brake application. In addition to this, in new generation engines, it also creates the vacuum in the auxiliary tank which can be used for some engine components such as turbo charger waste gate actuation, egr bypass valve, compressor shut off valves etc [9].

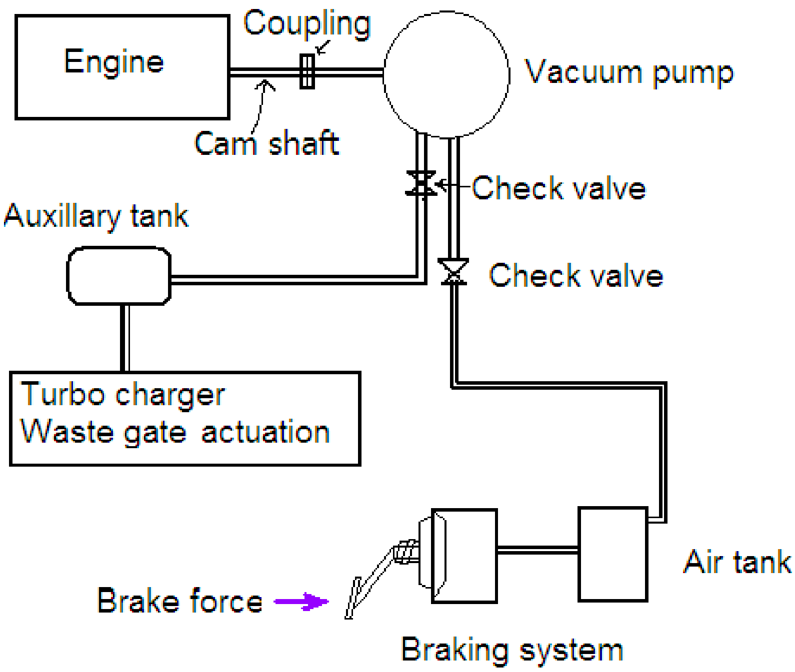


Figure 3.6 : Schematic view of cam operated vacuum pump

The common type used for automotive market is generally cam shaft driven vacuum pump. The vacuum pump is driven by the cam shaft by means of a coupling. The rotating parts inside the vacuum pump are lubricated by the engine oil. The oil also ensures good sealing between the stator and the moving parts by forming a thin film.

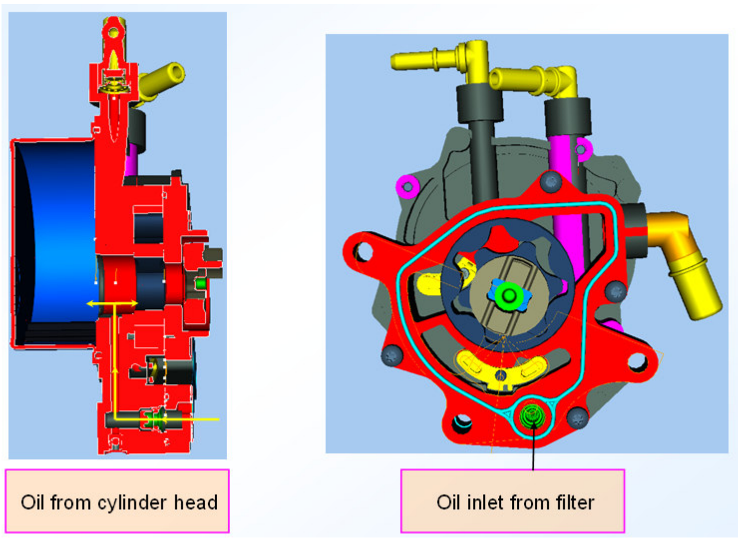


Figure 3.7 : Vacuum pump oil supply

It has an eccentrically mounted rotor which guides the vane. The rotor rotates around a unique generated profile. The non return valve mounted on the vacuum pump body sucks the air from air-tank and creates vacuum which will be used for braking application.

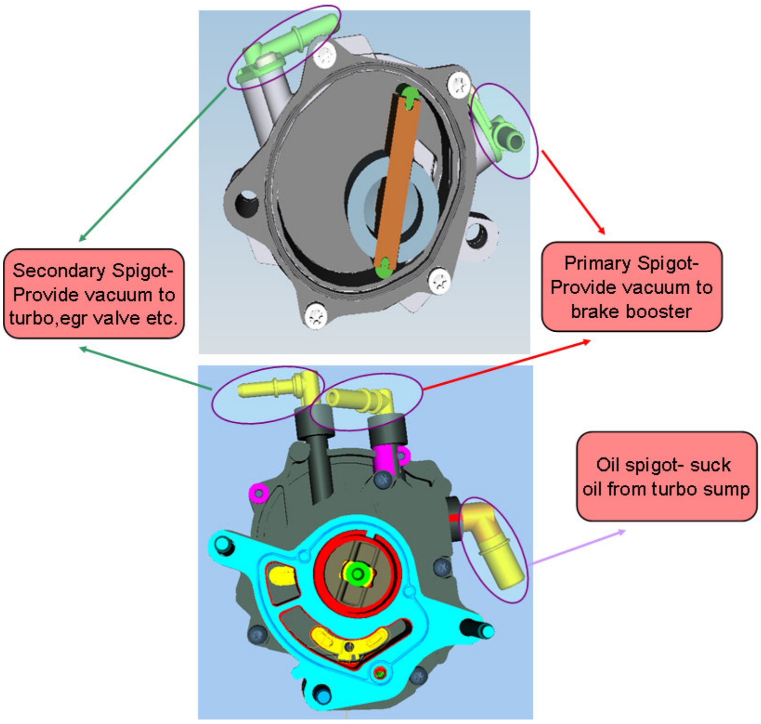


Figure 3.8 : Vacuum pump air-oil connectors

The vane on the rotor separates the vacuum chamber into low pressure and high pressure region. This pressure difference is due to the rotation of the vane which causes the air to enter from the air tank through the check valve.

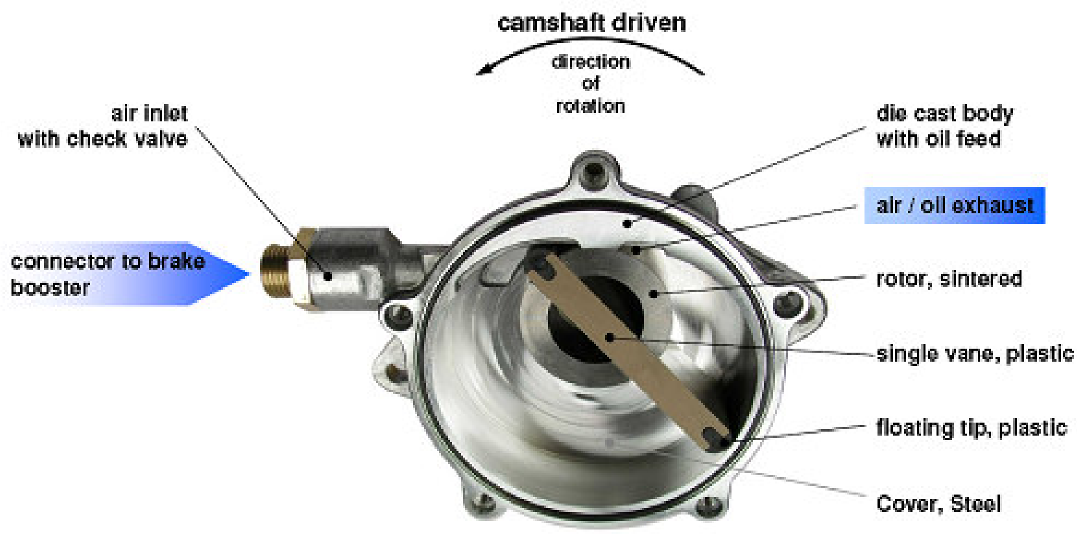


Figure 3.9 : Parts of a cam operated vacuum pump

After the work is done, the air-oil mixture is forced through the reed valve opening to atmosphere.

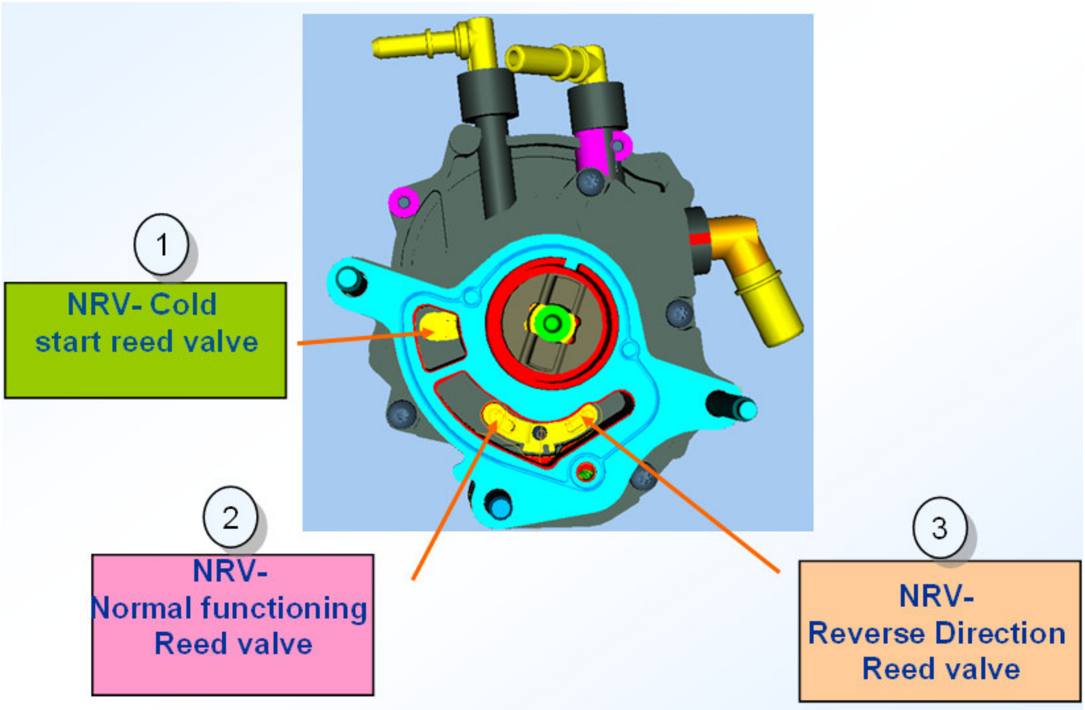


Figure 3.10 : Vacuum pump air oil mixture outlet valves

3.2 Vacuum Pump for Brakes

The brake booster is designed to create a greater braking force from a minimum pedal effort using a difference in atmospheric pressure and the engine's manifold vacuum. It increases the pedal force 2 to 4 times depending on the size of the diaphragm. The brake booster is located between the pedal and the master cylinder. When pressure is applied to the brake pedal, pressure is exerted on the booster air valve. With pressure created by the booster the master cylinder is applied.

The brake booster consists of the body, booster piston, piston return spring, reaction mechanisms and control valve mechanism. The body is divided into a constant pressure chamber and a variable pressure chamber. The chambers are seperated from each other by a diaphragm. The control valve mechanism regulates the pressure inside the variable pressure chamber [10].

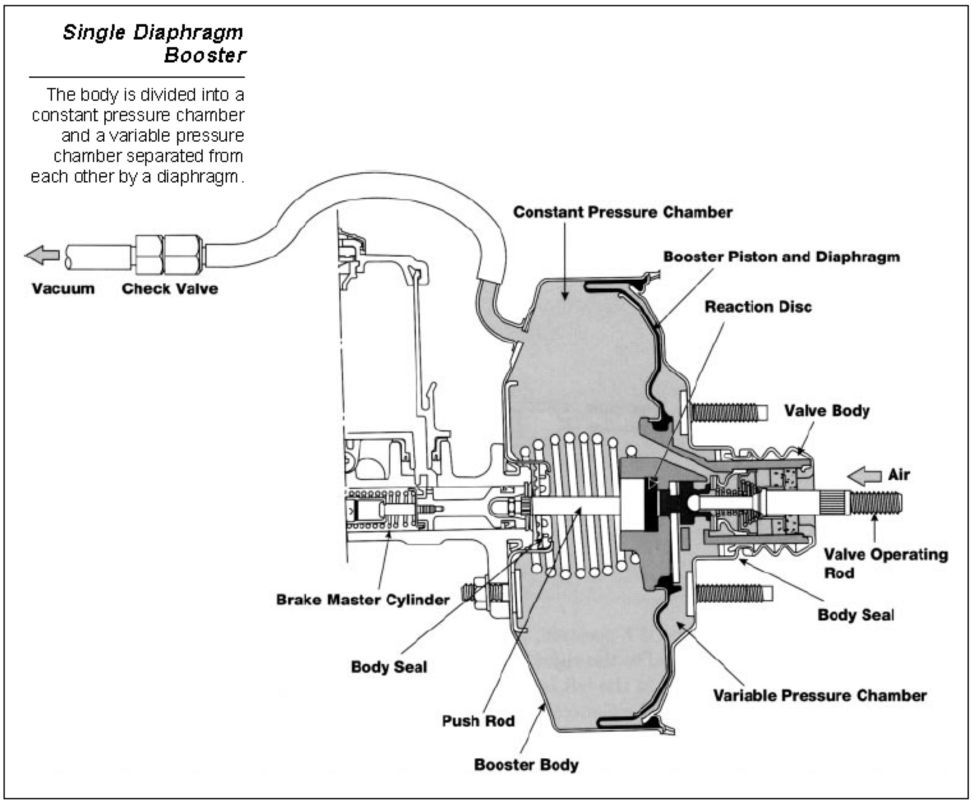


Figure 3.11 : Single brake booster

The basic principle of the brake booster is pressure differential. When vacuum is applied to both sides of the piston, the piston is pushed to the right by the spring and remains there [10].

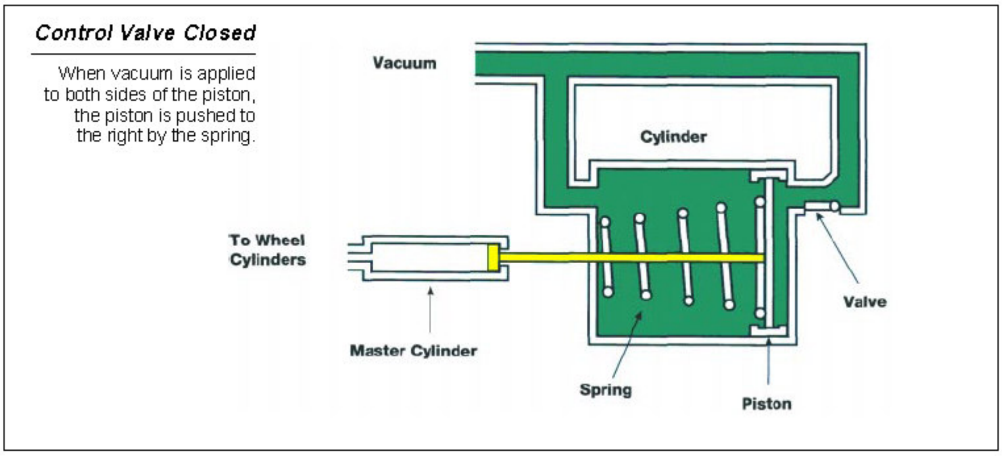


Figure 3.12 : Basic principle of the brake booster

When atmospheric air is allowed into chamber B, the piston starts to compress the spring, due to the difference in pressure, and moves to the left. This caueses the piston rod to move the piston of the master cylinder, generating hydraulic pressure [10].

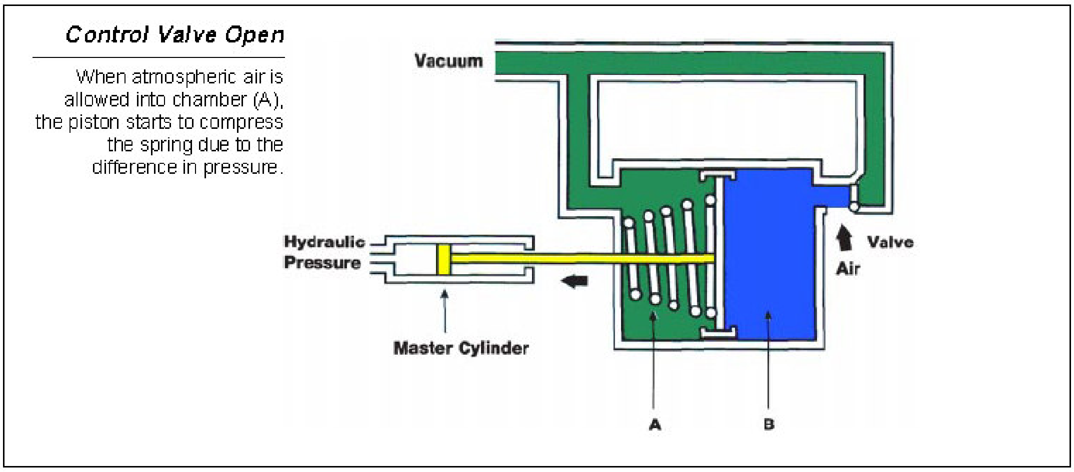


Figure 3.13 : Brake booster control valve open

In the off position, the air valve (connected to the valve operating rod) is pulled to the right by the air valve return spring. The control valve is pushed to the left by the control valve spring. This causes the air valve to contact the control valve. Therefore, the atmospheric air that passes through the air cleaner element is prevented from entering the variable pressure chamber.

The piston's vacuum valve is seperated from the control valve in this position, providing an opening between passage A and B. Since there is always vacuum in the constant pressure chamber, the opening allows vacuum into the variable pressure chamber. As a result, the piston is pushed to the right by the piston return spring [10].

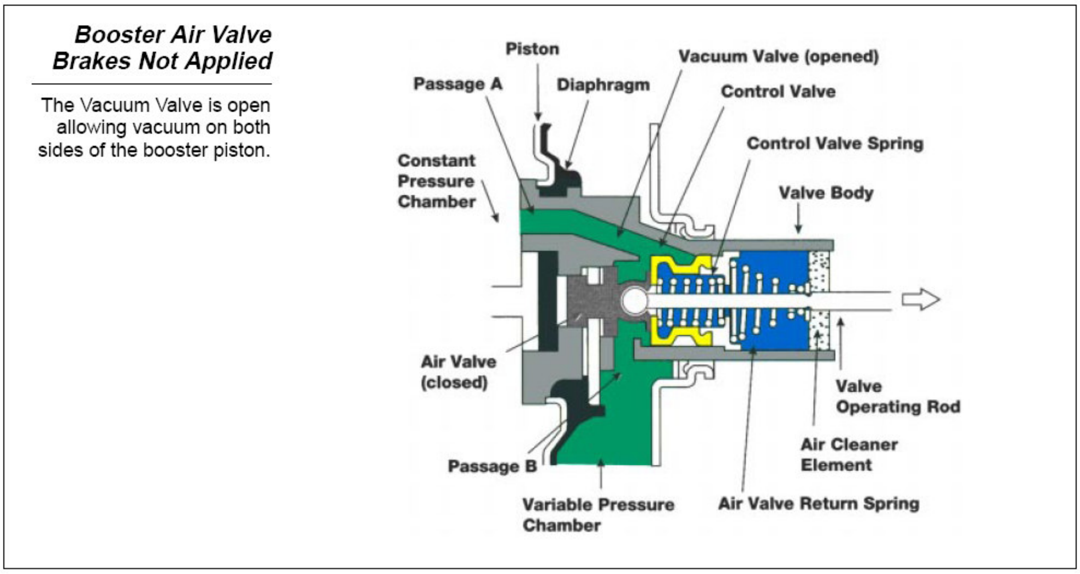


Figure 3.14 : Brake booster when brakes are not applied

In the on position, when the brake pedal is depressed, the valve operating rod pushes the air valve to the left. The control valve which is pushed against the air valve by the control valve spring, moves to the left untill it touches the vacuum valve. This blocks off the opening between passage A and B [10].

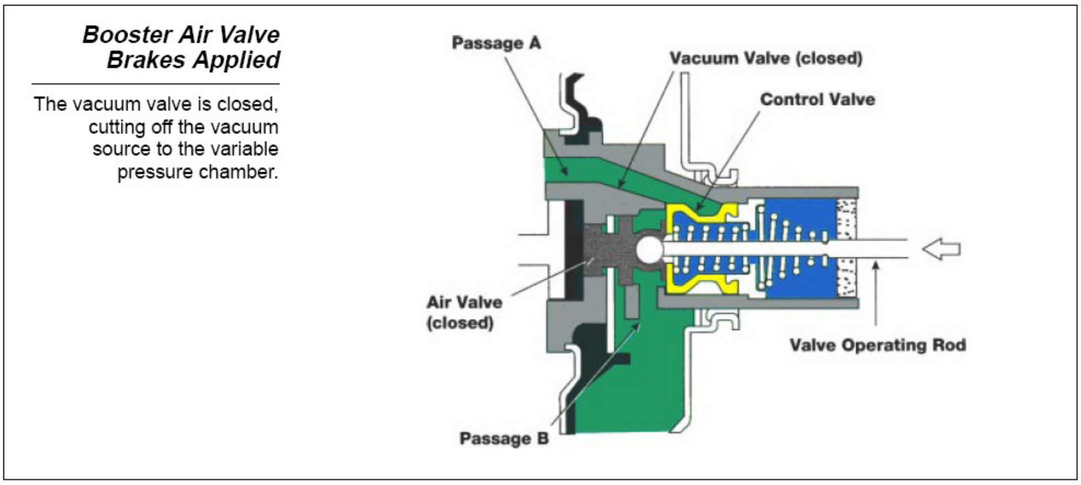


Figure 3.15 : Brake booster when brakes are applied-1

As the air valve moves to the left, it moves away from the control valve. This allows atmospheric pressure to enther the variable pressure chamber through passage B. The pressure difference between the constant pressure chamber and the variable pressure chamber causes the piston to move to the left. This, in turn, causes the reaction disc to move the booster push road to the left and exert braking force.

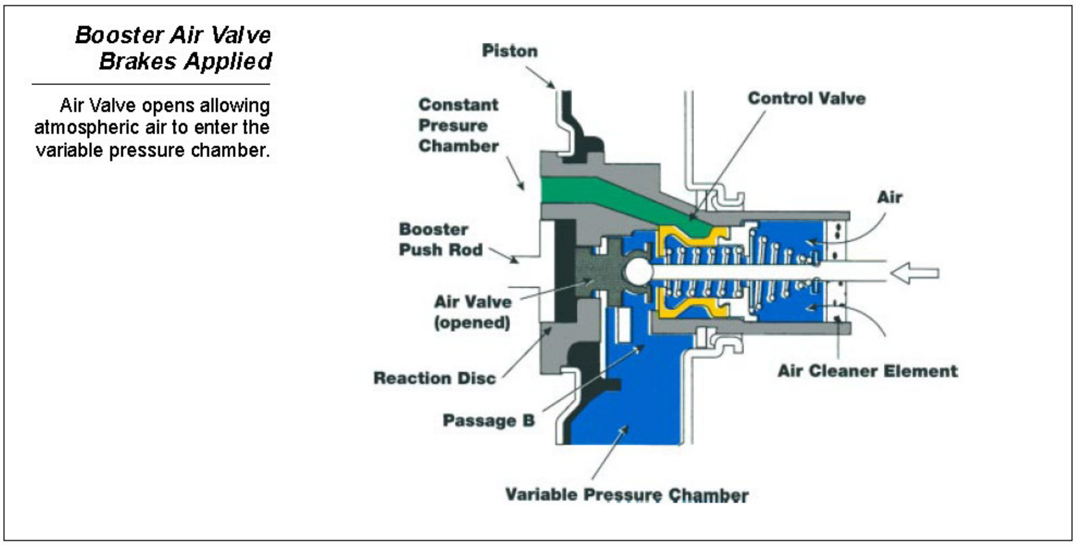


Figure 3.16 : Brake booster when brakes are applied-2

When the brake pedal is released, the valve operating rod and the air valve are moved to the right by the air valve return spring and reaction force of the master cylinder. This movement causes the air valve to contact the control valve, blocking atmospheric pressure from the variable pressure chamber. At the same time, the air valve also retracts the control valve spring. This control valve moves away from the vacuum valve, connecting passage A with passage B. This allows atmospheric pressure from the variable pressure chamber to flow into the constant pressure chamber. The pressure difference is eliminated between the two chambers and the piston is pushed back to the right by the diaphragm/piston return spring. The booster returns to the released position [10].

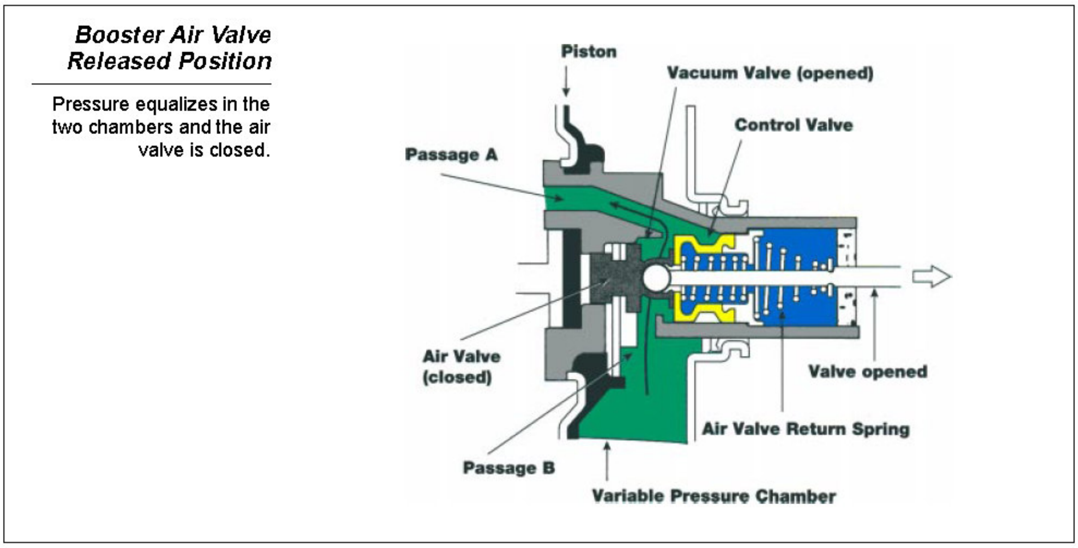


Figure 3.17 : Brake booster released position

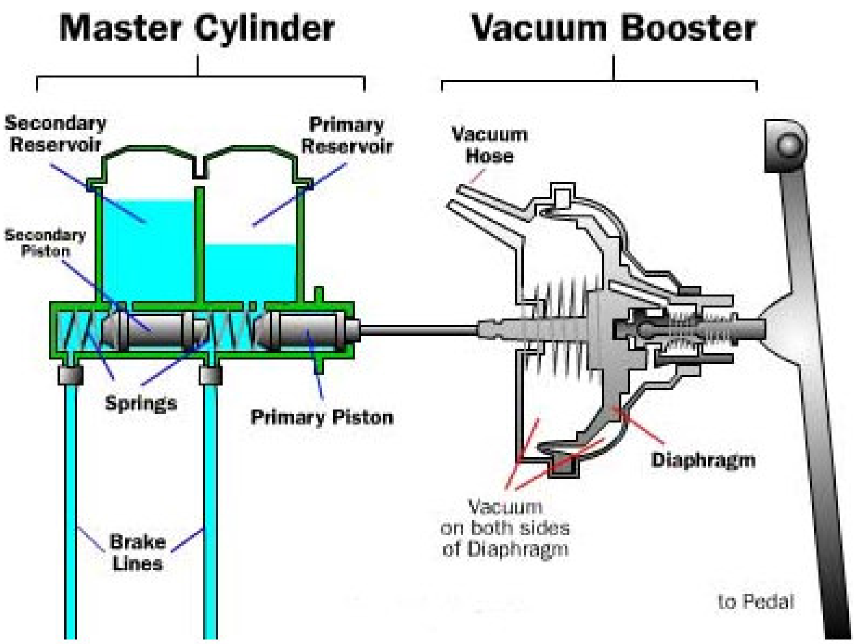


Figure 3.18 : Vacuum booster and master cylinder view

3.3 Vacuum Actuators

A vacuum motor/actuator is a device that uses vacuum to perform a mechanical function. An automobile engine uses vacuum motors / actuators to perform many functions. Typical functions performed by vacuum motors/actuators are:

- Opening and closing the heater and air conditioning duct doors
- Raising the EGR valve pintle off it's seat to recirculate exhaust gas back into the system
- Activating the vacuum advance unit to advance distributor spark advance timing

A vacuum motor/actuator is a partially sealed container (on most vehicles container is cylindrical in shape). One side of the container is sealed, and is equipped with a vacuum port. The other side of the container is covered by a rubber diaphragm, or a piston a spring, with an attached shaft or a lever. When engine vacuum is applied to the container's vacuum port, a partial vacuum (lower than atmospheric pressure) is created in the sealed section of the container. As a result, atmospheric pressure exerts a force on the side of the piston or diaphragm that is open to the atmosphere, causing it to move in the direction of the applied vacuum. This mechanical action is utilized to activate whatever is attached to the shaft or lever on the diaphragm. The amount of force applied by atmospheric pressure to the diaphragm is calculated by multiplying the pressure difference between atmospheric pressure and the pressure within the container times the surface area of the diaphragm or piston.

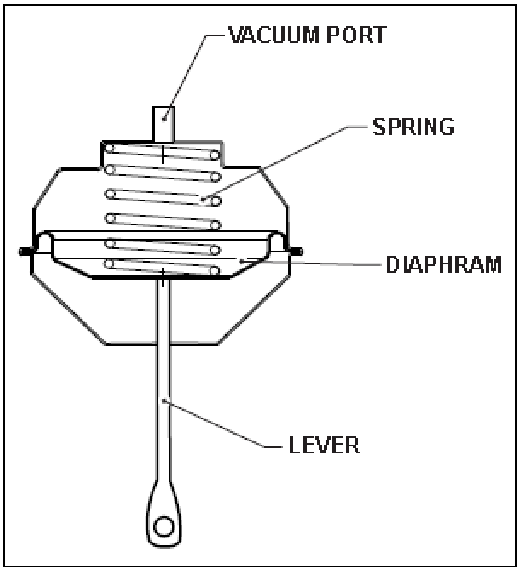


Figure 3.19 : Typical vacuum actuator

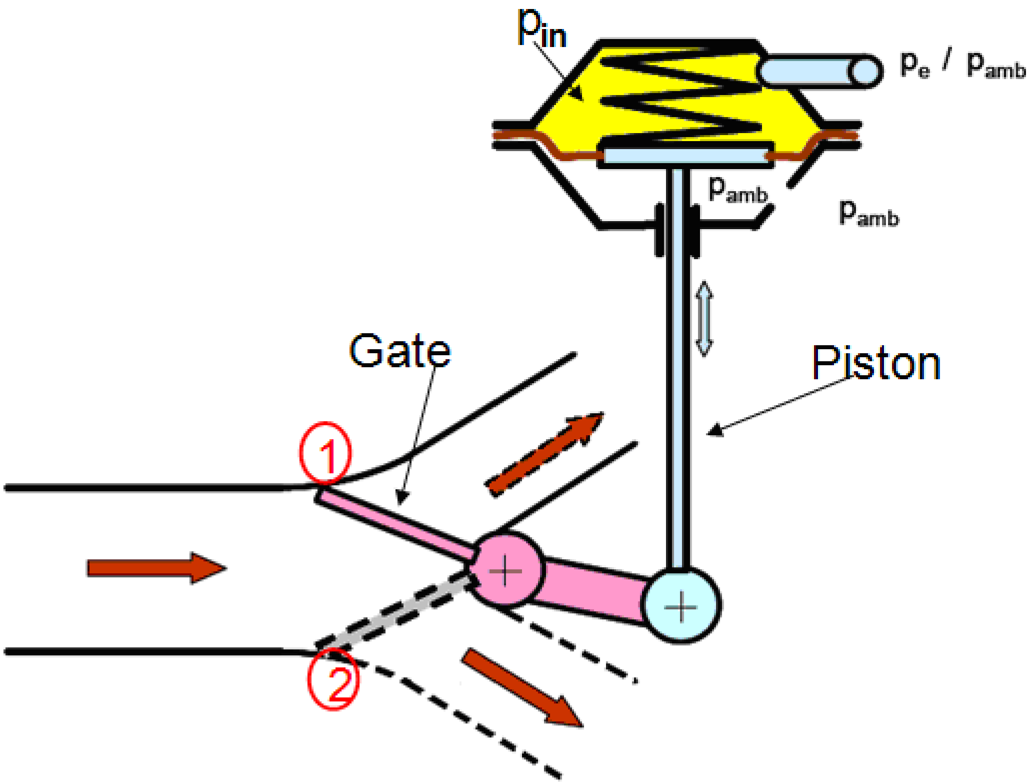


Figure 3.20 : View of a vacuum actuator

3.4 Electronic Vacuum Regulating Valves (EVRV)

The part is used to control a pilot pressure at the inlet port of a pneumatic actuator that is created by the vacuum circuit of the vehicle. The part is basically used to control a pneumatic actuator that is connected to the turbocharger with variable geometry (TVG). Additional pneumatic components that may also be controlled by this part are: exhaust gas re-circulation valve (EGR), metering device, RAA blade (intake air cooler blade). The part is connected with the vacuum system of the vehicle. Its continuous control permits regulating the vacuum at the inlet port of the pneumatic actuator, also called pilot pressure.

Since the air flow is not allowed to pulsate, the electric control valve must maintain a stable position irrespective of the prevailing control conditions.

In case of a failure of the electrical supply system the valve must be in the closed position: the pilot pressure at the output port of the part is between the atmospheric pressure and the residual control vacuum $\Delta P_{C0}\%$.

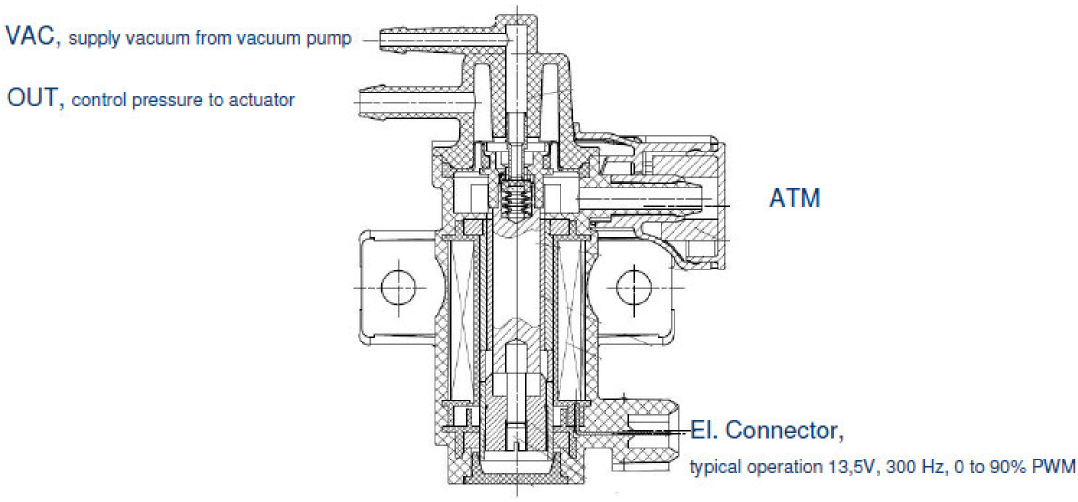


Figure 3.21 : Electronic vacuum regulating valve

The supply vacuum is supplied to the VAC port. If no current is applied, bellow closes the cap nozzle.

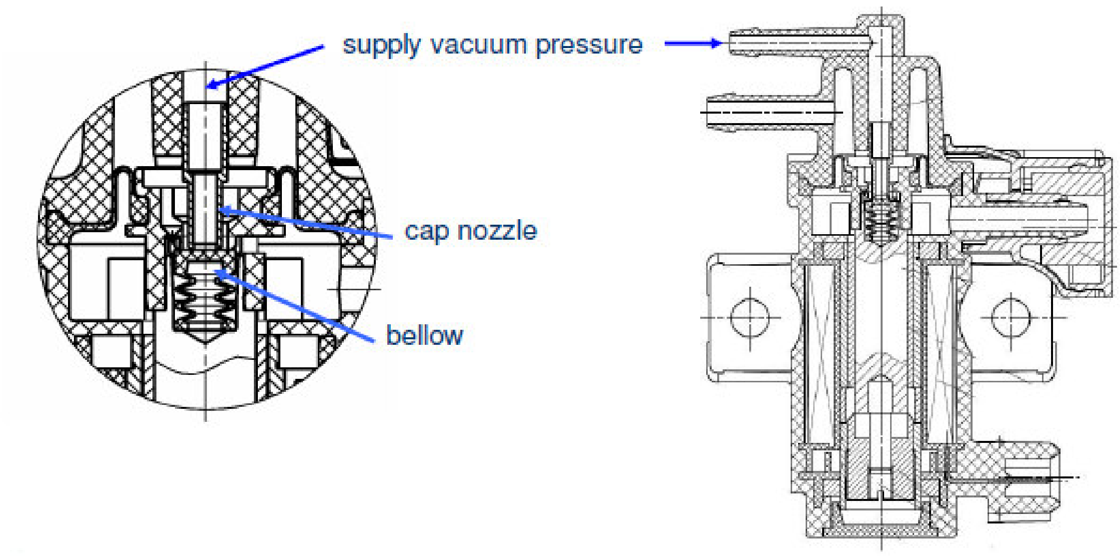


Figure 3.22 : EVRV when no current applied

Caused by current supply the whole plunger moves downwards and opens the closed cap nozzle. The vacuum pressure increases until vacuum pressure on diaphragm is equal to magnetic force. The control pressure spreads into cap and connected actuator. When balance is reached plunger moves back into neutral position.

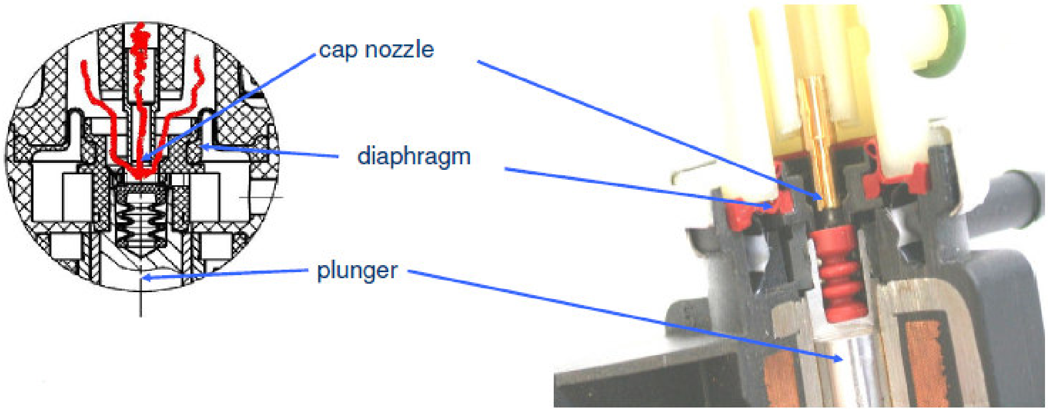


Figure 3.23 : EVRV evacuation phase

Without or less current no or less magnetic force is available. The force on diaphragm due to control pressure still exists. Caused by this the whole plunger moves upwards and the cap nozzle is closed. Due to this the plunger moves further and opens the ventilation borings. The control pressure decreased until balance between control pressure and magnetic force is reached.

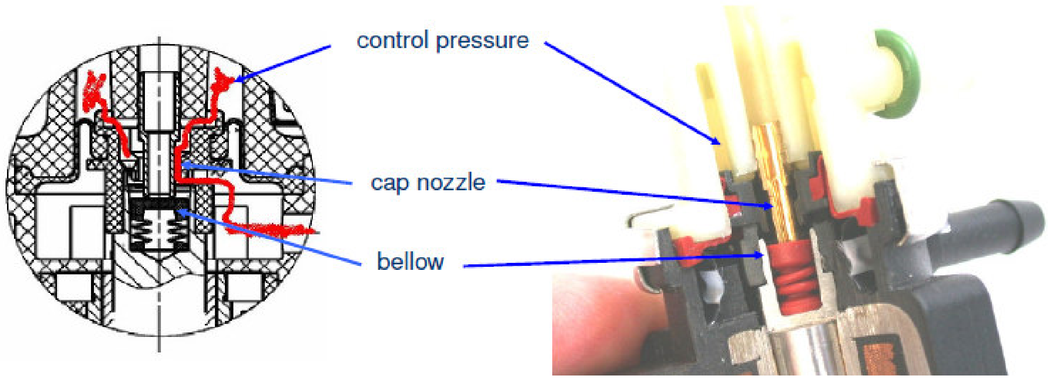


Figure 3.24 : EVRV ventilation phase

3.5 Solenoid Valve (VSV)

A solenoid is an electromechanical device which allows for an electrical device to control the flow of a gas or liquid. The electrical device causes a current to flow through a coil located on the solenoid valve. This current flow in turn results in a magnetic field which causes the displacement of a metal actuator. Solenoid valves may have two or more ports: in the case of a two-port valve the flow is switched on or off; in the case of a three-port valve, the outflow is switched between the two outlet ports.

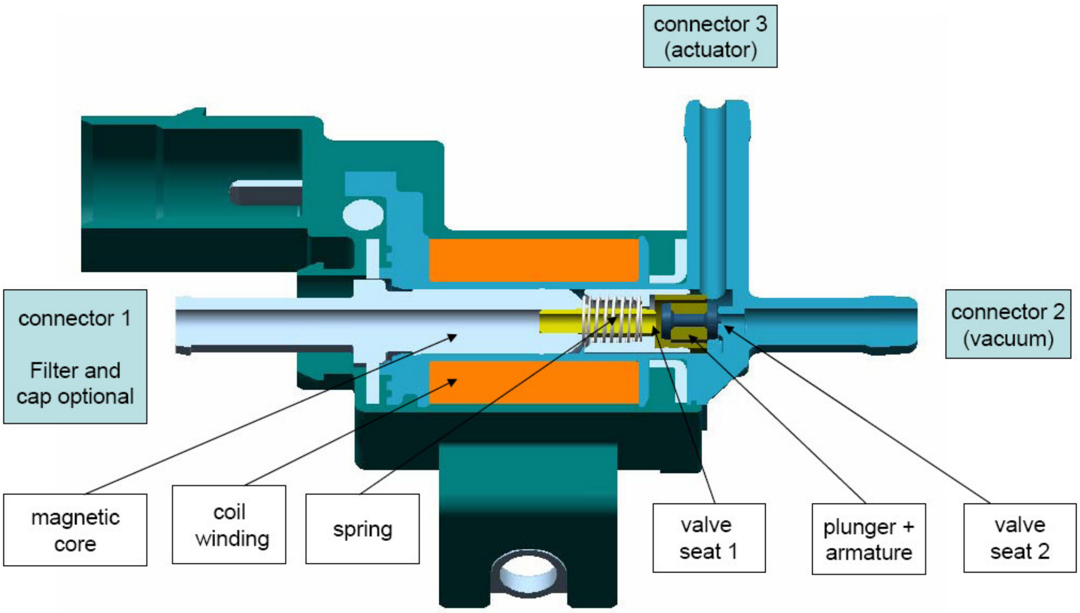


Figure 3.25 : View of a vacuum switching valve-3 way

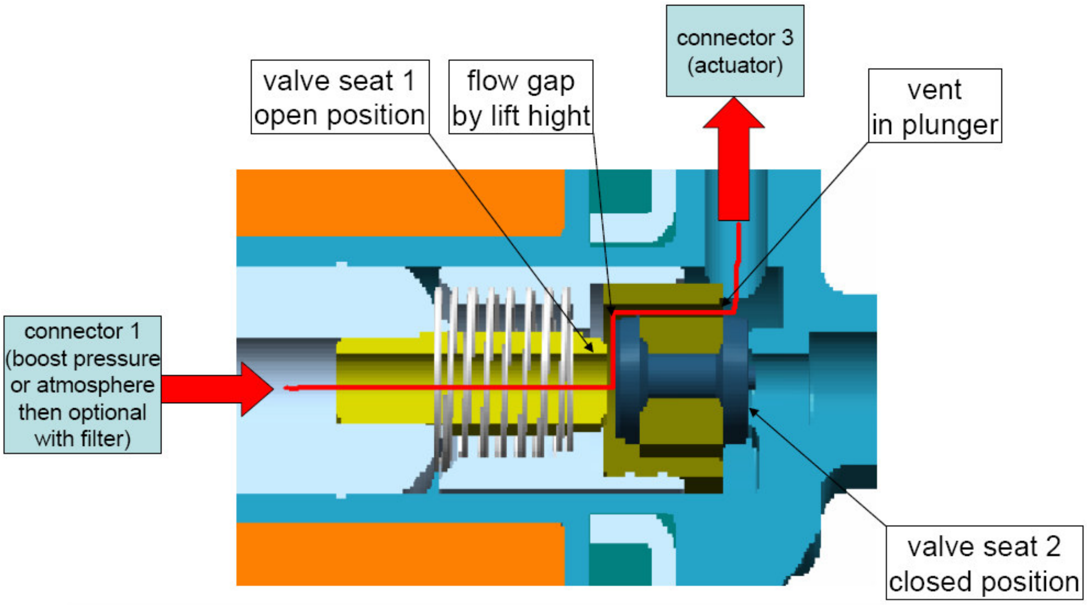


Figure 3.26 : VSV ventilation phase

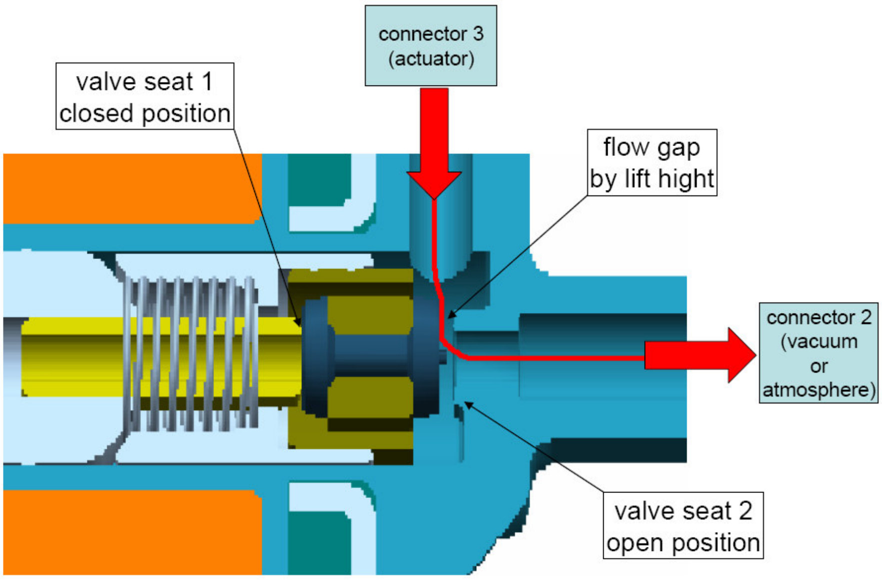


Figure 3.27 : VSV evacuation phase

3.6 Vacuum for EGR Systems

The Exhaust Gas Recirculation (EGR) system is used to reduce Oxides of Nitrogen (NOx) emissions. NOx is created when nitrogen and oxygen in the atmosphere mix at temperatures above 2500F° (1371° C). During combustion, temperatures in the cylinders can exceed 3500 F° (1927° C), providing ideal conditions for the formation of NOx. The EGR system reduces the formation of NOx by lowering the combustion temperature. This is accomplished by recirculating exhaust gases through an EGR valve back into the combustion chambers.

Depending on the make, model and year of vehicle, the EGR valve may be controlled by one of two methods [11]:

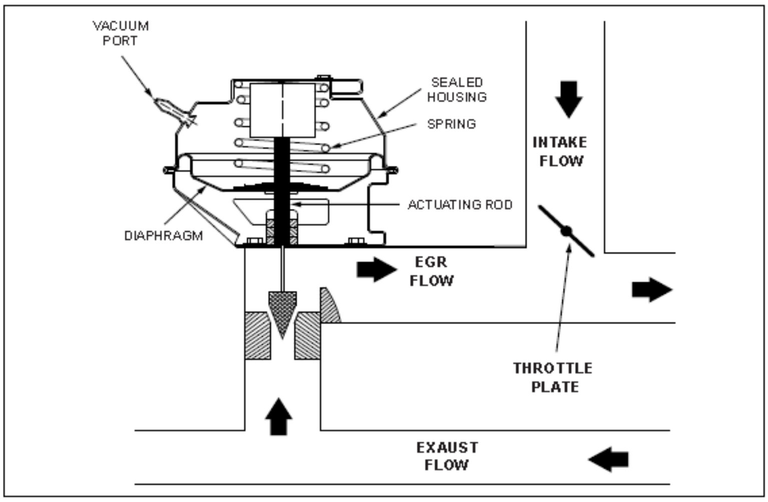


Figure 3.28 : Vacuum controlled egr valve-1

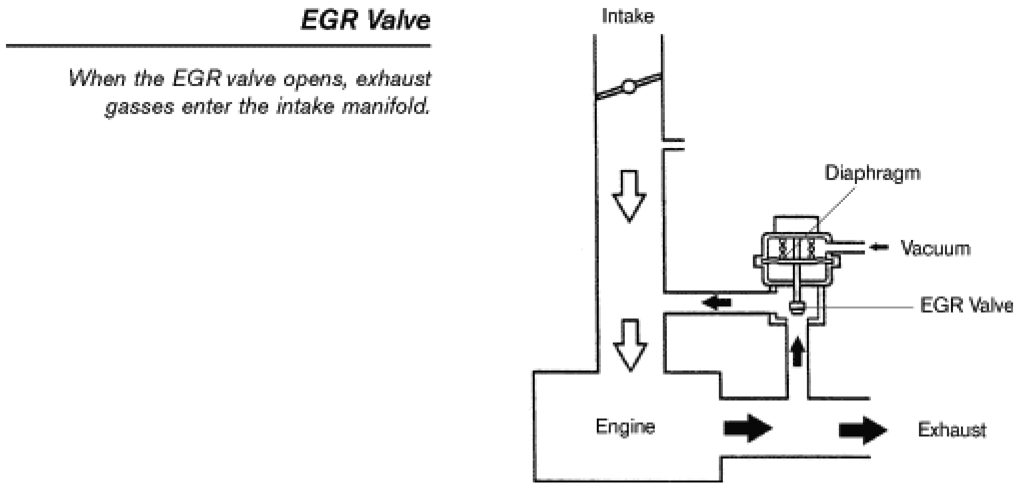


Figure 3.29 : Vacuum controlled egr valve-2

Ported Vacuum - The EGR valve is actuated by ported vacuum from above the carburetor's throttle plates. The amount of exhaust gas that is recirculated back into the combustion chamber depends on the amount of vacuum that reaches the EGR valve. At idle and at wide open throttle a negligible amount of vacuum reaches the valve and the valve stays closed. As the throttle plates gradually open, vacuum reaches the valve and the valve starts to open. A low vacuum to the EGR valve will cause the valve to open slightly. As the vacuum to the valve increases, the EGR valve continues to open until it is fully opened.

EGR Control System - The control system monitors engine operation and modulates the amount of EGR based on engine operating conditions such as coolant temperature, ambient air temperature, exhaust back pressure and engine speed or load.

To improve drivability before the engine has reached normal operating temperature, most vehicles use a thermal switching device, which shuts off the EGR system while the engine is cold [11].

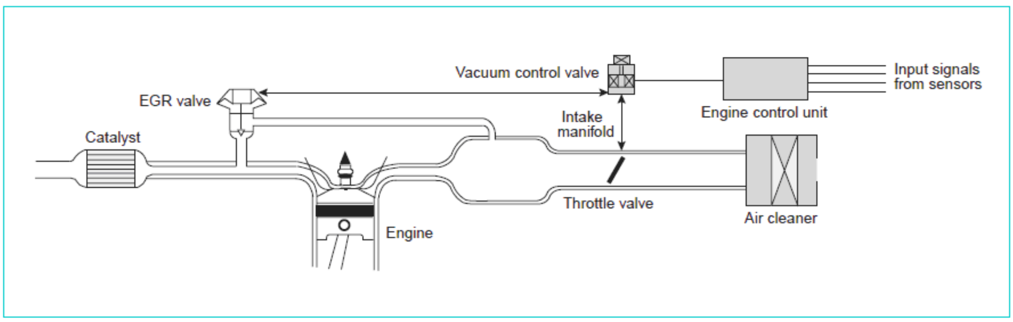


Figure 3.30 : A conventional vacuum diaphragm for egr system

3.7 Vacuum for Turbo System

A turbocharger is a small radial fan pump driven by the energy of the exhaust gases of an engine. A turbocharger consists of a turbine and a compressor on a shared shaft. The turbine converts exhaust heat to rotational force, which is in turn used to drive the compressor. The compressor draws in ambient air and pumps it in to the intake manifold at increased pressure, resulting in a greater mass of air entering the cylinders on each intake stroke.

The objective of a turbocharger is the same as a supercharger; to improve the engine's volumetric efficiency by solving one of its cardinal limitations. A naturally aspirated automobile engine uses only the downward stroke of a piston to create an area of low pressure in order to draw air into the cylinder through the intake valves. Because the pressure in the atmosphere is no more than 1 atm, there ultimately will be a limit to the pressure difference across the intake valves and thus the amount of airflow entering the combustion chamber. Because the turbocharger increases the pressure at the point where air is entering the cylinder, a greater mass of air (oxygen) will be forced in as the inlet manifold pressure increases. The additional airflow makes it possible to maintain the combustion chamber pressure and fuel/air load even at high engine revolution speeds, increasing the power and torque output of the engine.

Because the pressure in the cylinder must not go too high to avoid detonation/knocking and physical damage, the intake pressure must be controlled by venting excess gas. The control function is performed by a west gate, which routes some of the exhaust flow away from the turbine. This regulates air pressure in the intake manifold.

3.7.1 Parallel turbos

Some engines, such as V-type engines, utilize two identically sized but smaller turbo, each fed by a separate set of exhaust streams from the engine. The two smaller turbo produce the same (or more) aggregate amount of boost as a larger single turbo, but since they are smaller they reach their optimal RPM, and thus optimal boost delivery, more quickly. Such an arrangement of turbo is typically referred to as a parallel twin-turbo system. The first production automobile with parallel twin turbochargers was the Maserati Biturbo of the early 1980s. Later such installations include the Nissan

GT-R, Mitsubishi 3000GT VR-4, the Nissan 300ZX, the Audi B5 S4, and the BMW twin-turbo 3.0 liter inline 6 cylinder cars (E90, E81, E60) [12].

3.7.2 Sequential turbos

Some carmakers combat lag by using two small turbo. A typical arrangement for this is to have one turbo active across the entire rev range of the engine and one coming on-line at higher RPM. Early designs would have one turbocharger active up to a certain RPM, after which both turbochargers are active. Below this RPM, both exhaust and air inlets of the secondary turbo are closed. Being individually smaller they do not suffer from excessive lag and having the second turbo operating at a higher RPM range allows it to get to full rotational speed before it is required. Such combinations are referred to as sequential twin turbo. Porsche 959 first used this technology back in 1985. Sequential twin-turbo are usually much more complicated than a single or parallel twin-turbo systems because they require what amounts to three sets of pipes-intake and waste gate pipes for the two turbochargers as well as valves to control the direction of the exhaust gases. Many new diesel engines use this technology to not only eliminate lag but also to reduce fuel consumption and reduce emissions [12].

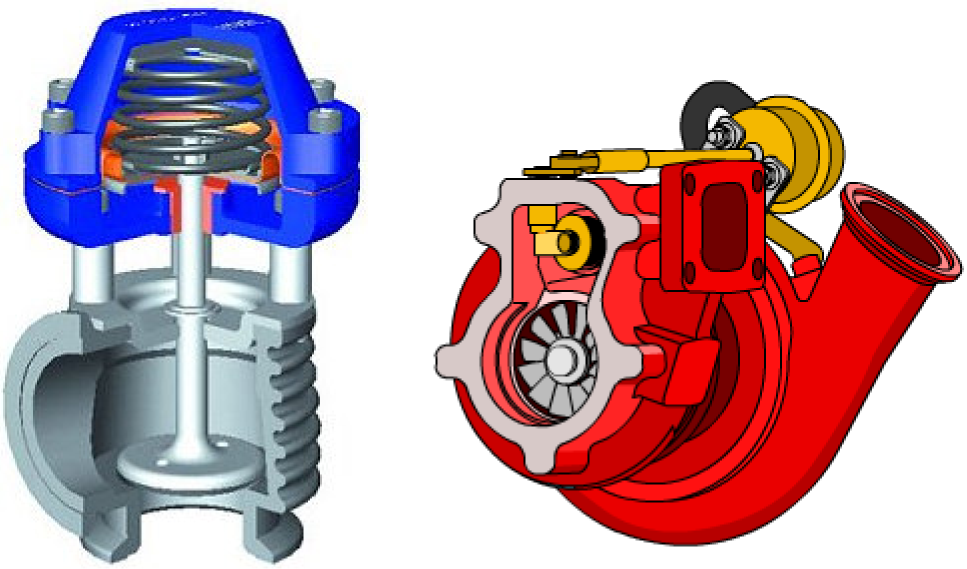


Figure 3.31 : Turbo west gate cut away

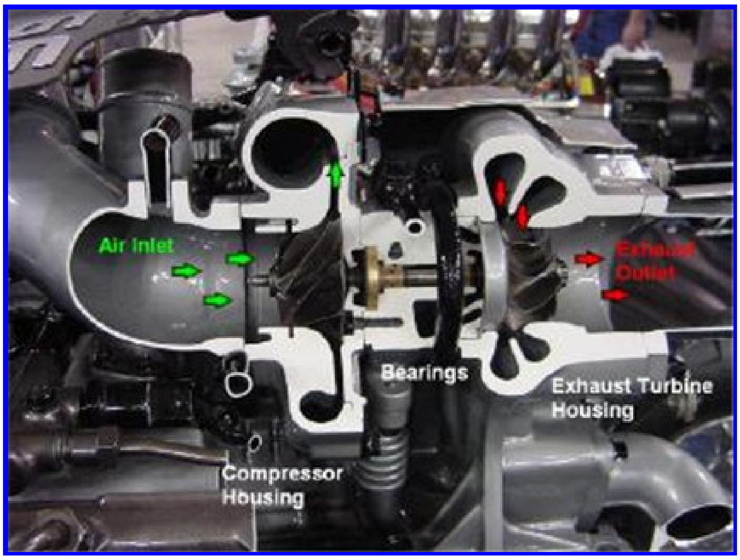


Figure 3.32 : Turbocharger view

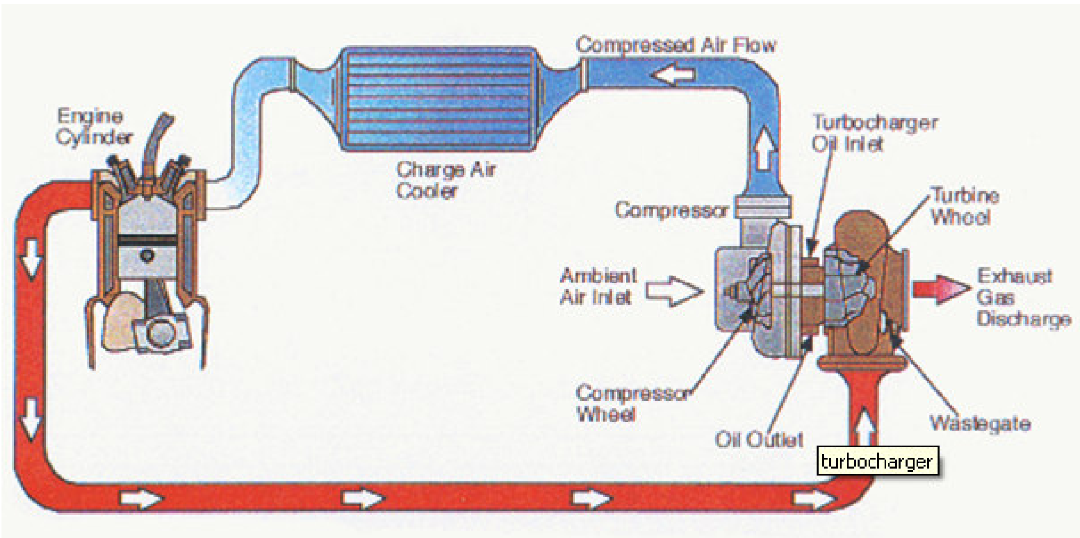


Figure 3.33 : Turbocharger working principle

Applications that require a good response at low engine speeds benefit from the use of a small turbo. However, even though they may be designed to spin at up to 250,000 rpm and withstand exhaust gas temperatures of over 1000°C, there is a danger that a small turbine can overspeed and overboost at higher engine speeds.

In order to prevent this from happening, some turbochargers are fitted with a wastegate or turbine bypass: as the pressure reaches the maximum preset level, a valve opens to allow some of the exhaust gas to bypass the turbine and flow straight into the exhaust system.

The simplest form of wastegate control is a pneumatic actuator. The sensor port on the actuator is connected directly to the compressor outlet and, as pressure rises in the top part of the actuator above the diaphragm, it acts against the pressure of a spring to move a rod, thereby opening the turbine bypass valve (wastegate).

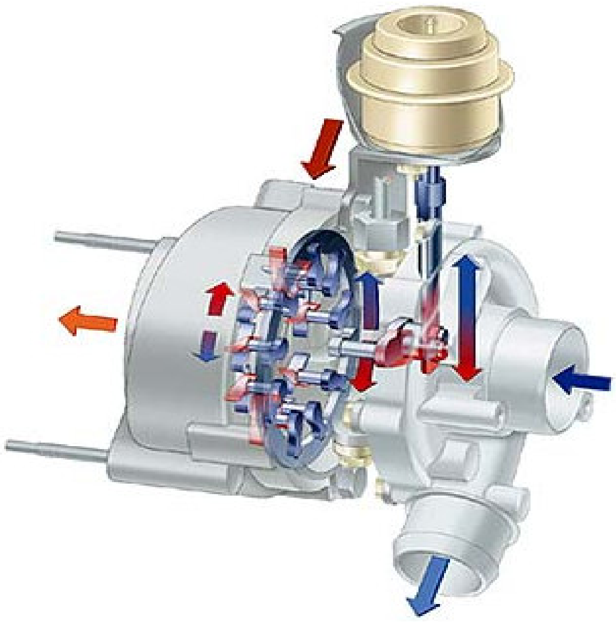


Figure 3.34 : VNT turbocharger

Vacuum pumps can be used in turbocharger systems by providing vacuum to west gate actuators and also to actuators for variable nozzle turbines. Moreover, according to the types of turbosystems, compressor shut off and turbine shut off valves are also actuated with vacuum pumps in generally.

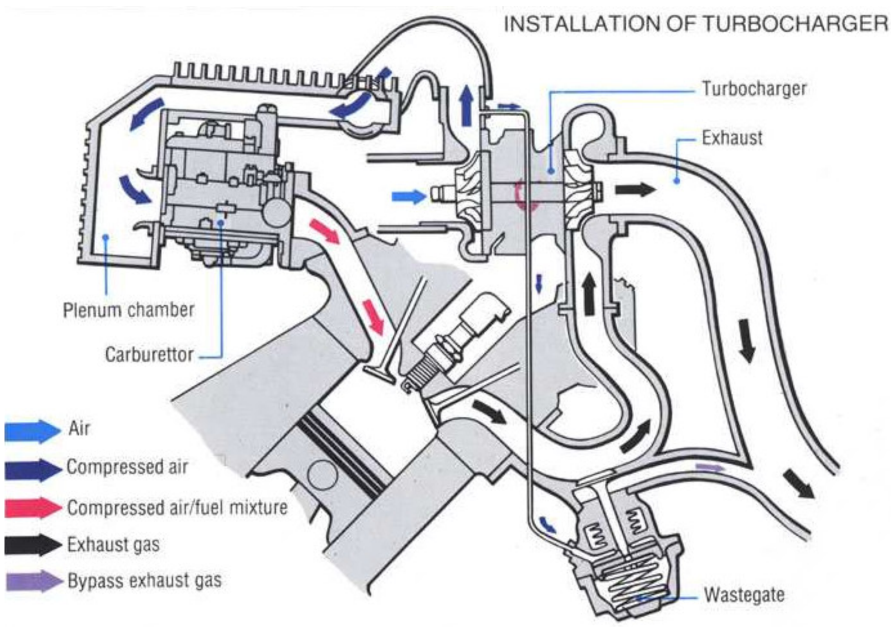


Figure 3.35 : Gas flow in turbocharger

4. DW10F PREMIUM ENGINE VACUUM SYSTEM CAPABILITY 1D-THEORITICAL ANALYSIS

4.1 Introduction

In this thesis, Ford 2.0 lt 204 Hp 4-cylinder diesel engine vacuum system capability is analyzed in terms of response times on the engine vacuum actuators and designed by optimazing vacuum reservoirs in different altitudes and capacities.

For this, FLOWMASTER/CAE 1d flow analysis program will be used to get the results by inputs from real component tests and engine tests. At the end of the analysis, it is purposed to demonstrate the capabilities of the vacuum system performance to define the targets.

Engine vacuum targets are defined by the system engineer according to program requirements. On this engine, it is required to open/close three actuators which are used for turbine by pass valve, waste gate valve and egr bypass valve. Those all will be controlled by vacuum with electronic vacuum regulated valves by appyling signals from ECU.

By considering the operation points, each have different actuating engine rpm. EGR bypass valve is generally activated at low engine speeds where the exhaust gases are relatively cold with compare to higher speeds where there is no need to send the gases to egr cooler. While wastgate valve, which is used in secondary small turbo, is operating at higher engine speeds, turbin bypass valve is used in a wide range of engine speed to prevent/allow gases to enter secondary turbo.

Key performance requirements:

- Turbine By-Pass Valve Response Time: 500 ms
- Waste Gate valve response time: 1000 ms
- EGR By-Pass valve response time: NOT CRITCAL

4.2 Governing Equations

In the analysis, the flow is assumed as compressible, transient flow where the density is not constant with the time changes. Compressible flow is the flow in which density changes are significant. Pressure changes normally occur throughout a fluid flow, and these pressure changes, in general, induce a change in the fluid density. In a compressible flow, the density changes that result from these pressure changes have a significant influence on the flow. The changes in the flow that result from the density changes are often termed compressibility effects. All fluids are compressible. However, compressibility effects are more frequently encountered in gas flows than in liquid flows.

The main difference between compressible flow and almost incompressible flow is not the fact that compressibility has to be considered. Rather, the difference is in two phenomena that do not exist in incompressible flow.. The first phenomenon is the very sharp discontinuity (jump) in the flow in properties. The second phenomenon is the choking of the flow. Choking is when downstream variations don't effect the flow. Though choking occurs in certain pipe flows in astronomy, there also are situations of choking in general (external) flow. Choking is referred to as the situation where downstream conditions, which are beyond a critical value(s), doesn't affect the flow.

An important dimensionless parameter in compressible flows is the Mach number, M.

$$M = \frac{V}{a} \qquad \textbf{(4.1)}$$

where a is the speed of sound and V is the velocity of the flow. For a gas, the speed of sound is given by

$$a = \sqrt{k \times R \times T} \qquad \textbf{(4.2)}$$

where R is the gas constant, $k = c_p/c_v$, c_p and c_v being the specific heats at constant pressure and constant volume respectively, and T is the temperature. If $M < 0.3$ in a flow, the density changes in the flow will usually be negligible; that is, the flow can be treated as incompressible. Compressible flows are, therefore, as a rough guide, associated with Mach numbers greater than 0.3.

Applying the continuity equation :

$$-\rho \times Q = \frac{d}{dt}(\rho \cdot V) \qquad \textbf{(4.3)}$$

Where ρ is the mass density of air:

$$= \rho \cdot \frac{\partial V}{\partial t} + V \cdot \frac{\partial \rho}{\partial t} \qquad \textbf{(4.3a)}$$

$$= \rho \cdot \frac{\partial V}{\partial t} + V \cdot (\frac{\rho}{P})\frac{\partial P}{\partial t} \qquad \textbf{(4.3b)}$$

The term $\frac{\partial V}{\partial t}$ is the rate of change of volume V, with respect to time.

Assuming that the entire volume is rigid, i.e. $\frac{dV}{dt} = 0$

Therefore:

$$-Q = \frac{V}{P} \cdot \frac{dP}{dt} \qquad \textbf{(4.3c)}$$

$$dt = -\frac{V}{Q} \cdot \frac{dP}{P} \qquad \textbf{(4.3d)}$$

Integrating both sides of the equation with the initial condition $P = P_0$ at t =0 gives:

$$t = \frac{V}{Q} \cdot \ln\frac{P_o}{P} \qquad \textbf{(4.4)}$$

Time [t]	:	Sec
Booster volume [V]	:	Liter
Initial pressure [P_0]	:	Mbar
End pressure [P]	:	Mbar
Pumping speed [Q]	:	l/sec

In the above equation P_0 is atmospheric pressure (1 atm in most cases) and P is the final vacuum level (in atm) to be reached [13].

The vacuum level is influenced by surrounding atmospheric pressure. This can be extracted from the below physical formula;

$$p(h) = p_0 \cdot \exp(-M \cdot g \cdot h / R \cdot T) \quad \textbf{(4.5)}$$

M =29 g/mol (molar mass air)

g = 9,81 m/s2 (gravity)

h = height over see level (m)

R = 8,3 J/molK (molar gas constant)

T =Temperature in Kelvin

Table 4.1: Vacuum vs altitude

Barometric pressure			The reading on the vacuum gauge at 1,013.25 mbar				
mmHg	mbar	Equivalent m above sea level	60 -kPa	75 -kPa	85 -kPa	90 -kPa	99 -kPa
593	790.6	2,000	37.7	52.7	62.7	67.7	76.7
671	894.6	1,000	48.1	63.1	73.1	78.1	87.1
690	919.9	778	50.7	65.7	75.7	80.7	89.7
700	933.3	655	52.0	67.0	77.0	82.0	91.0
710	946.6	545	53.3	68.3	78.3	83.3	92.3
720	959.9	467	54.7	69.7	79.7	84.7	93.7
730	973.3	275	56.0	71.0	81.0	86.0	95.0
740	986.6	200	57.3	72.3	82.3	87.3	96.3
750	999.9	111	58.7	73.7	83.7	88.7	97.7
760	1,013.25	0	60.0	75.0	85.0	90.0	99.0

The actuator consists of a spring and membrane inside a canister. When the spring fits in the canister without being compressed when the stroke is zero, the spring doesn't have any pretension. That means that the actuator will start to move as soon as pressure is applied [14].

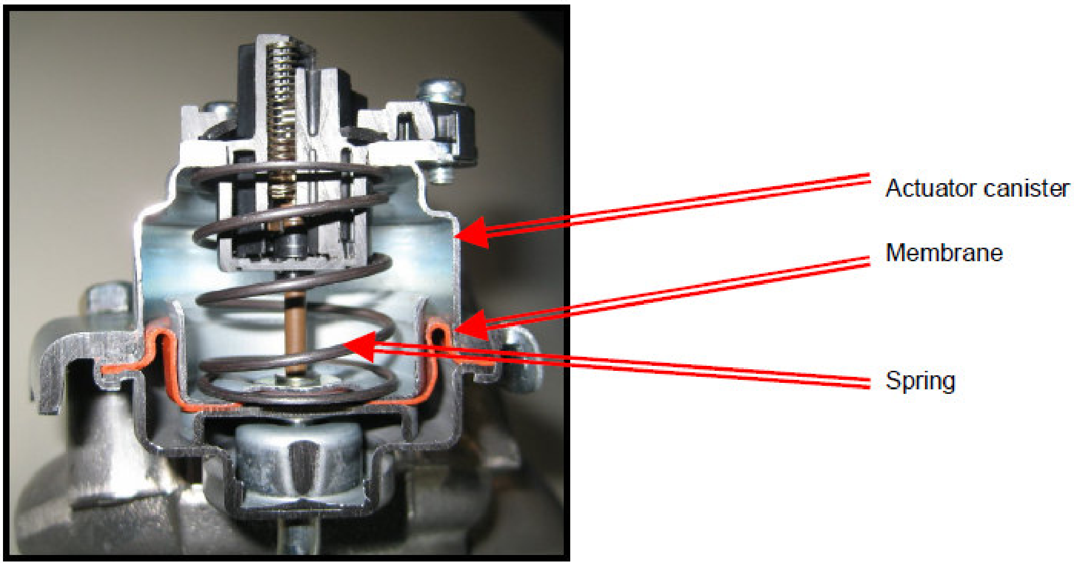


Figure 4.1 : Vacuum actuator parts

The actuator will work according to the law of Hooke ($F=k*L$) with F being the force applied (pressure * membrane area), k being the spring stiffness and L being the change of length of the spring. The spring stiffness determines the relation between the pressure needed to move the actuator rod over a certain distance.

The stiffer the spring, the more pressure is needed to move the actuator rod. This principle of an actuator without pre-tension is not used on turbochargers because of gas pulsations that will cause the waste gate valve to move uncontrolled, which is undesirable.

When the spring is compressed inside of the actuator canister at zero stroke, the actuator has a certain pre-tension. When pressure is applied to the actuator the actuator rod will not move directly. After a certain pressure, the pre-tension is reached. After this point the actuator rod will start to move. The pressure when the actuator rod start to move is corresponding with the strength of the pre-compressed spring. After the start of movement the stroke of the actuator rod will be linear in relation to the pressure.

The spring force is a static force. The exhaust gas force on the valve depends on the engine speed (exhaust gas forces increase with the engine speed). The needed actuator force is the maximum needed force which is the result of the maximum appearing exhaust gas force and the spring force [14].

The formulas below give an overview how to determine the needed actuator force;

$$F_{gas} = P_{gas} \times A_{w/G} \tag{4.6}$$

$$P_{gas} = P_{pulsation} + (P_{turbine-out} - P_{turbine-in}) \tag{4.7}$$

$$A_{W/G} = \frac{1}{4} \times \Pi \times D_{w/G}{}^{2} \tag{4.8}$$

$$F_{spring} = C_{spring} \times S_{act-rod} \tag{4.9}$$

$$Lever - ratio = \frac{l_{external}}{l_{internal}} \tag{4.10}$$

$$F_{Act} = F_{Gas} + F_{Spring} \tag{4.11}$$

$$F_{Act\_on\_W/G} = F_{Act} \times Lever - ratio \tag{4.12}$$

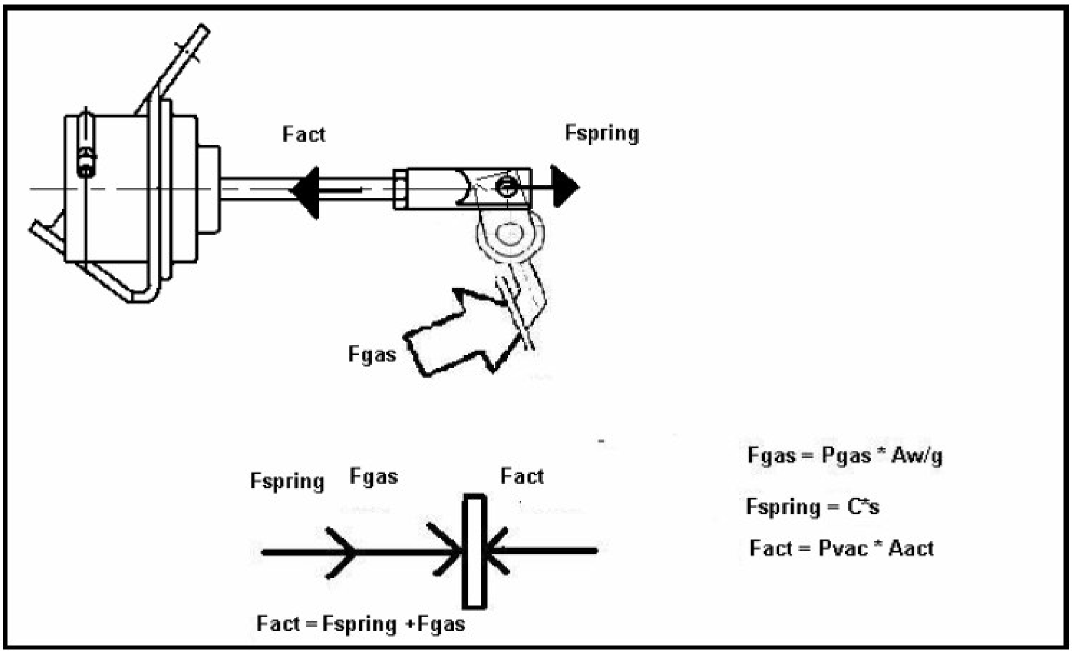


Figure 4.2 : Forces on vacuum actuator

Explanation abbreviations in formulas:

- F= Force (Newton)
- P= Pressure (atm)
- A=Area of membrane in the actuator (cm^2)
- D= Diameter of membrane in the actuator (cm)
- C = spring-stiffness (N/mm)
- S= displacement of valve (mm)
- L = length (mm)

The pre-tension and the maximum actuator stroke define the spring stiffness that is needed in order to supply the pre-tension.

$$C_{spring} = \frac{F_{spring}}{S_{actuator-rod}} \quad \textbf{(4.13)}$$

4.3 Flowmaster Vacuum Pump Model

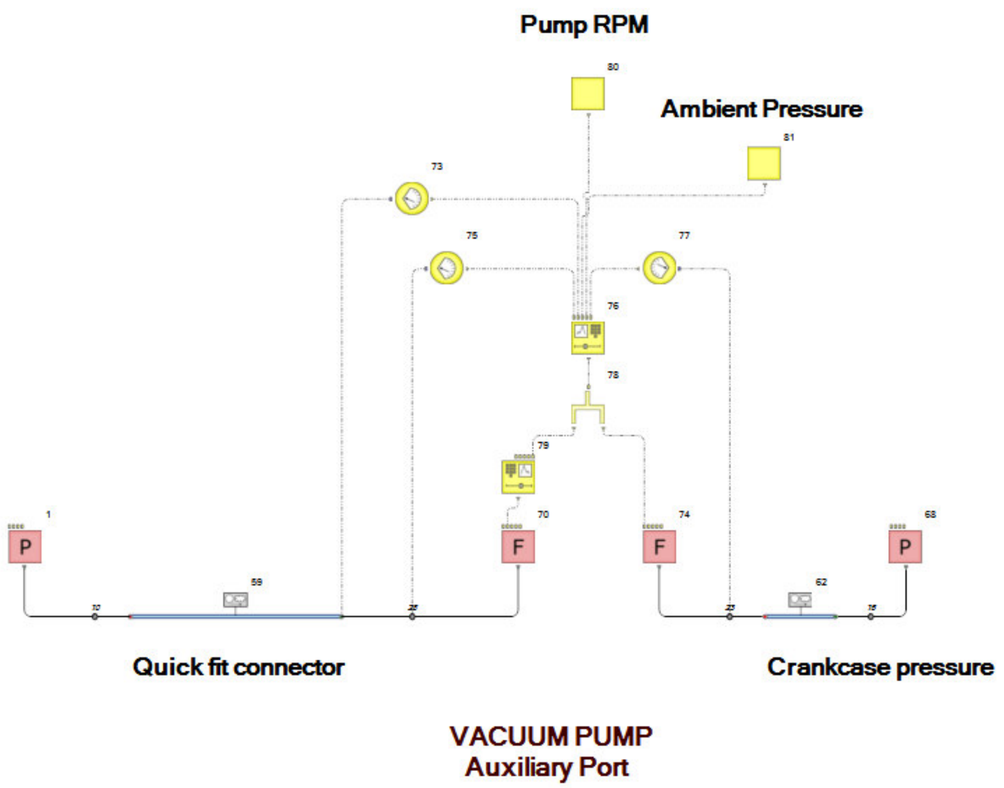


Figure 4.3 : Vacuum pump model

4.3.1 Vacuum pump performance data from rig test

1) Rig test results provided by supplier Pierburg.

2) The vacuum pump is a sliding vane pump with twin ports. The main port supplies the brake booster. The servo (or "auxiliary") port supplies the engine vacuum harness system.

3) The data given below shows curves of volume flow rate versus pump speed at the servo (secondary-to engine) port, for a fixed depression in the brake booster port.

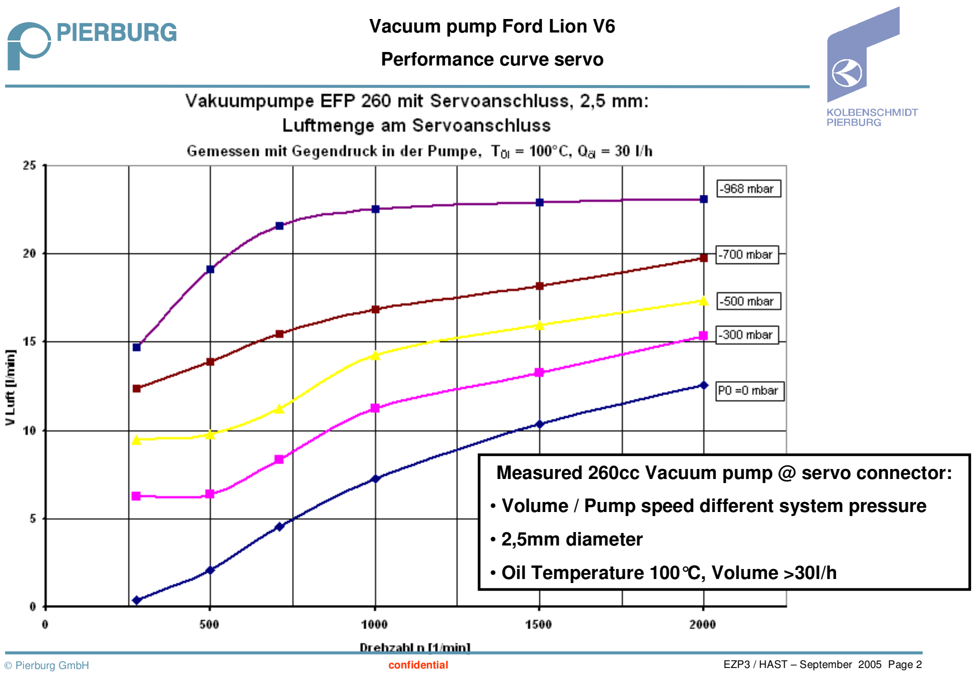


Figure 4.4 : Vacuum pump secondary port performance

4.3.2 Assumptions

1. The volume of the vacuum pump of the engine will be 260 cc
2. The oil temp for the rig test is 100 °C. This is considered a worse case for pump performance.
3. Vacuum pump performance for the primary port which is used for servo brakes is not considered.
4. Pump speed is half of the engine speed.

4.4 Vacuum Pump Rig Test Correlation – Step 1

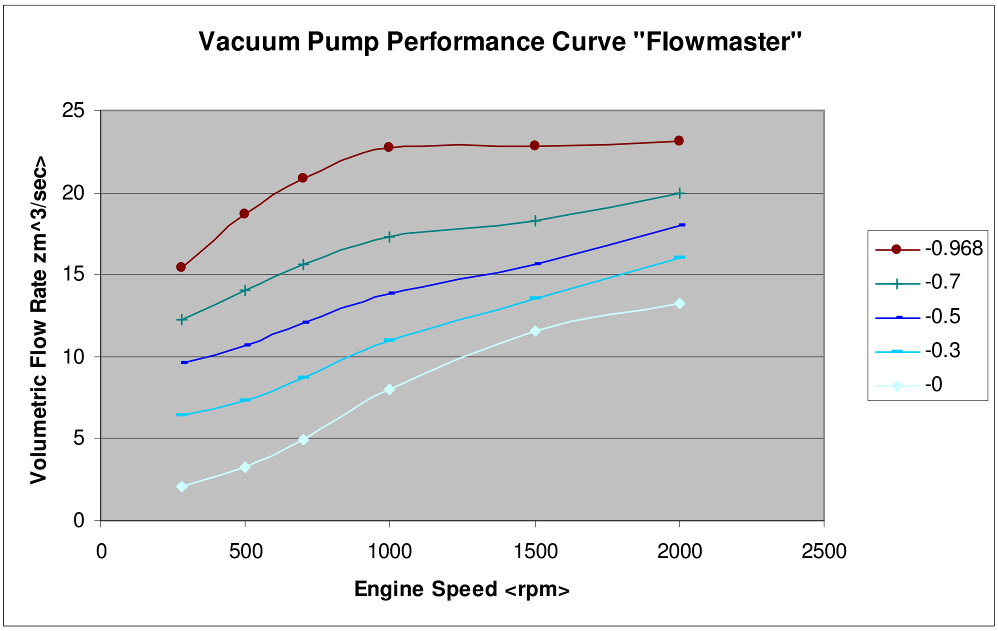


Figure 4.5 : Vacuum Pump secondary spigot performance data obtained from flowmaster pump model given above

4.5 Vacuum Pump Rig Test Correlation – Step 2

1) The test set-up given below was built by supplier Pierburg for the performance measurement of the vacuum pump

2) The brake port plugged for this test and initial pressure inside the reservoir set to the ambient pressure.

3) The volume of the reservoir set to 500 cc which is the equivalent total inner volume of the vacuum harness system.

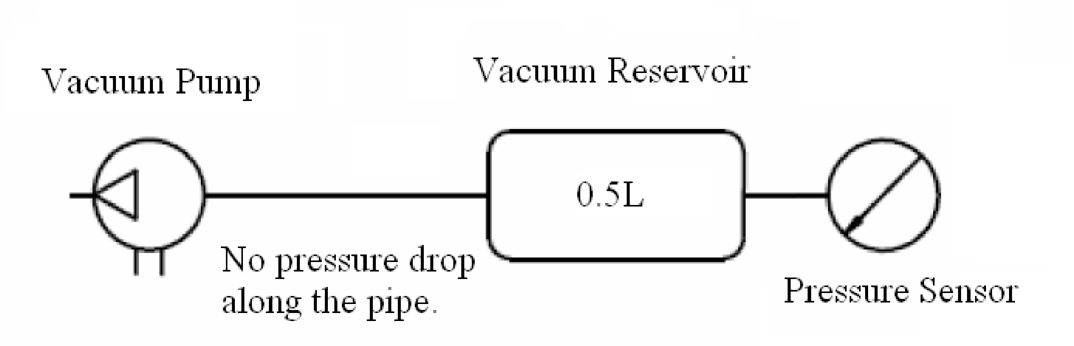


Figure 4.6 : Vacuum pump performance test set up

The vacuum pump drained air from a reservoir with 500 cc volume at 325 rpm according to the curve given below:

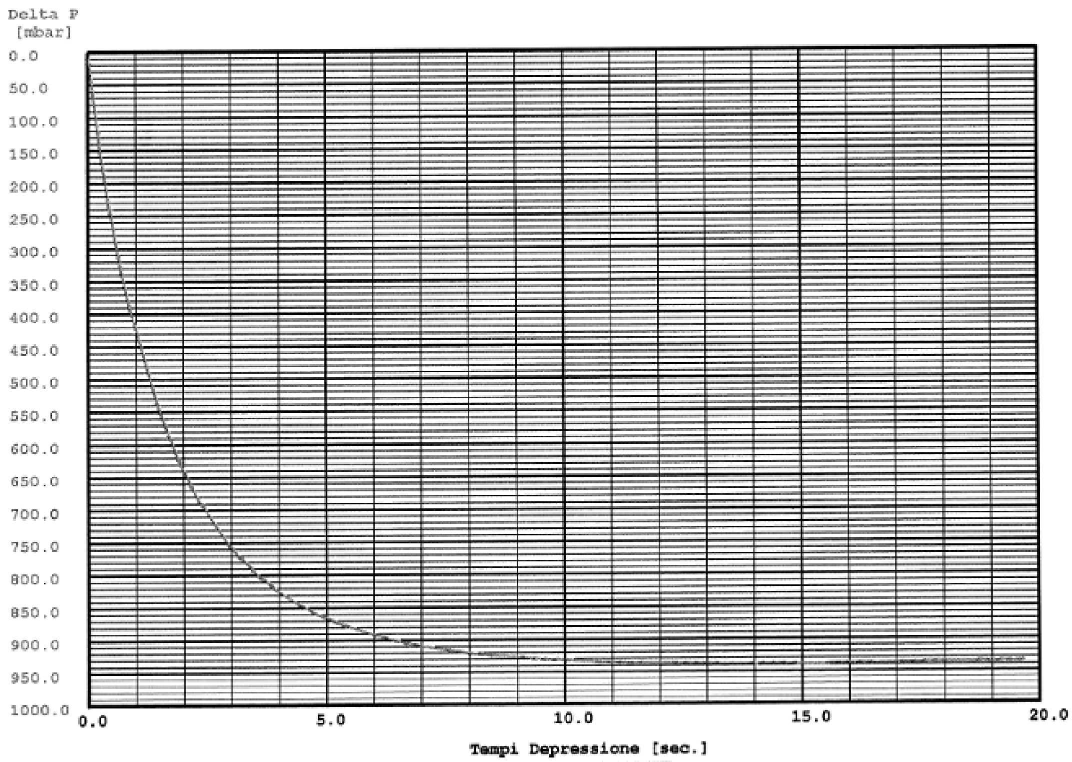


Figure 4.7 : Vacuum pump performance measurement @ 325rpm

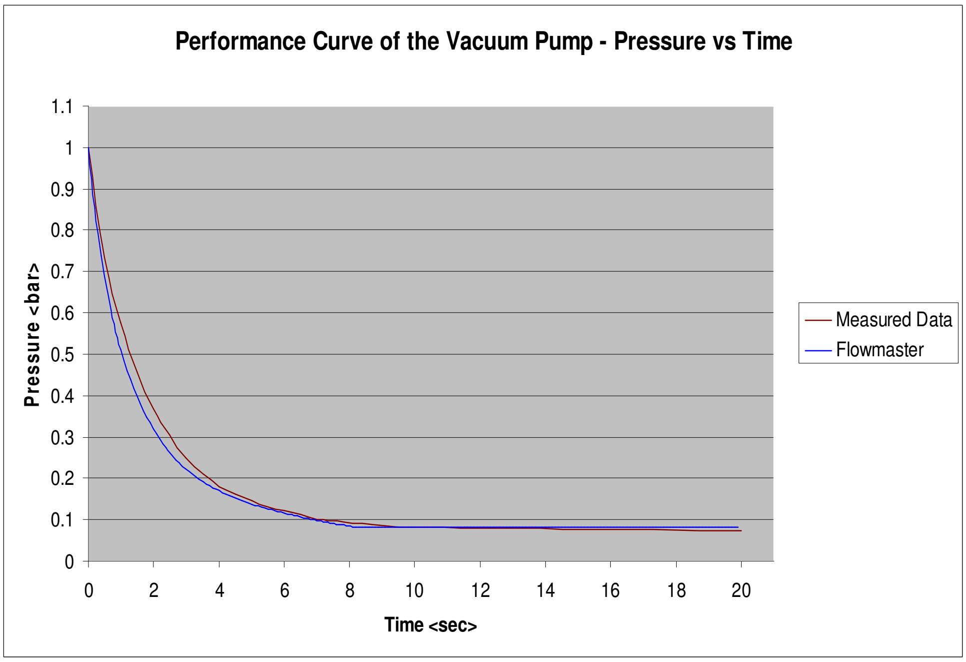


Figure 4.8 : Flowmaster vs. test data

4.6 Evrv (Electronic Vacuum Regulated Valve) Model:

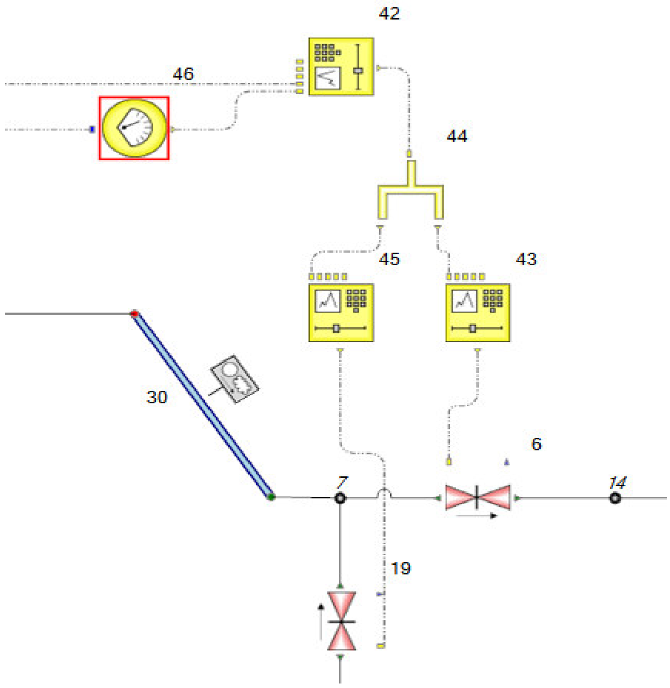


Figure 4.9 : Electronic vacuum regulated valve model

The EVRV valve modeled as shown above to simulate the regulation of the vacuum at different engine speeds. And the EVRV fine tuned with the help of the test set-up given below. There is 10 ms lag between command and demand of the EVRV. The lag will be neglected for these analysis.

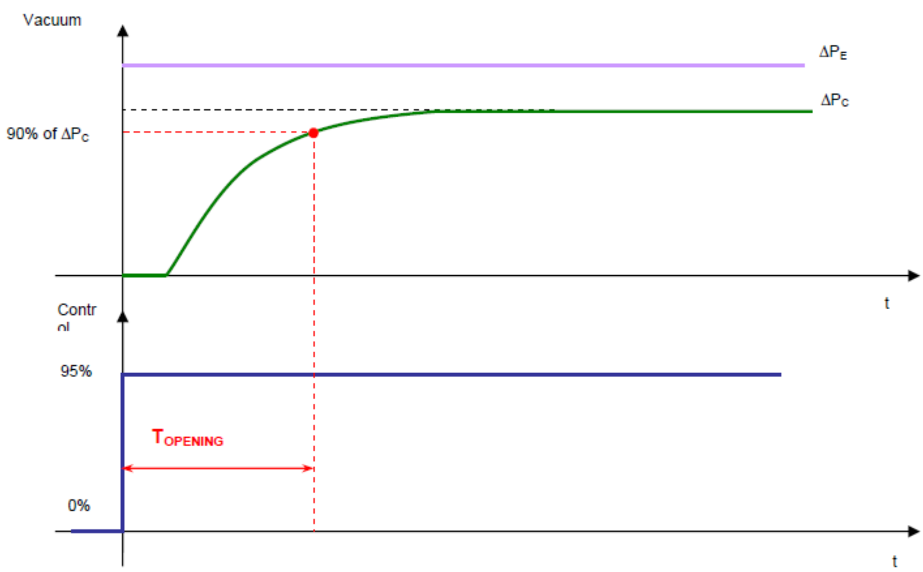


Figure 4.10 : Actuation of the EVRV

Blue line represents the command sent to the EVRV and green line represents the actual vacuum level at the out port (port to actuator) of the EVRV.

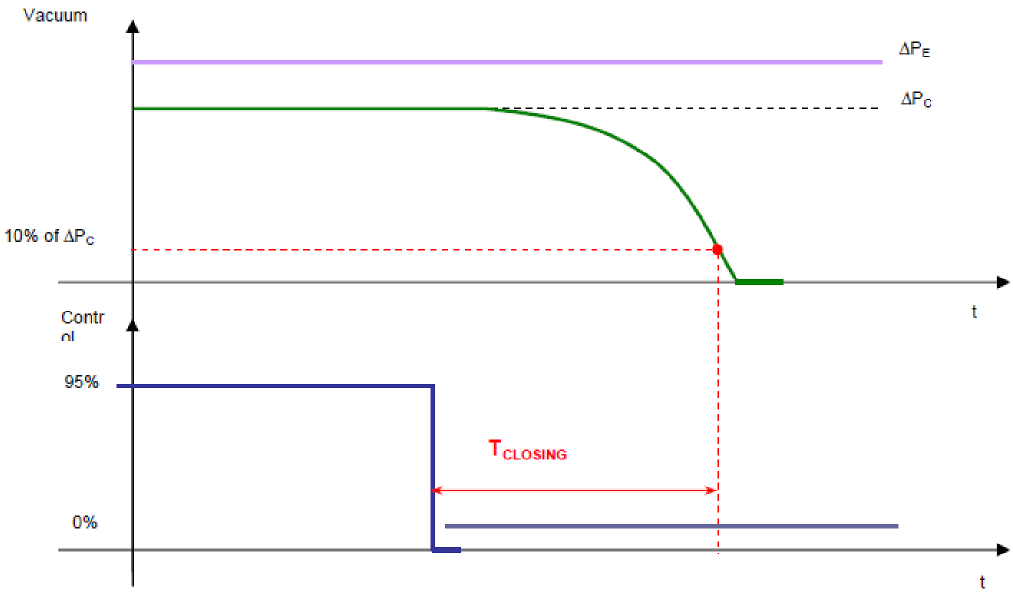


Figure 4.11 : De-actuation of the EVRV

4.6.1 Evrv rig test correlation:

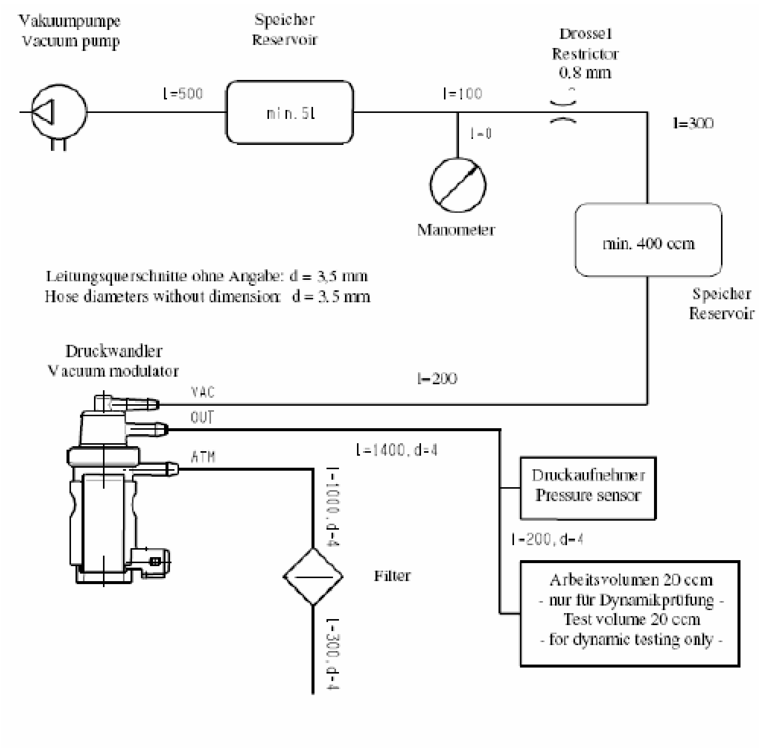


Figure 4.12 : EVRV response time measurement set-up

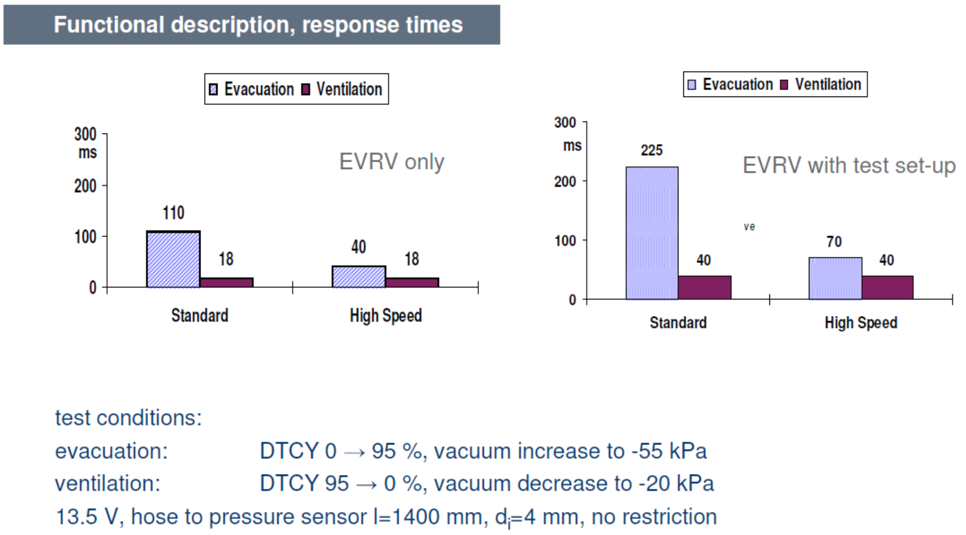


Figure 4.13 : EVRV response times

The test set-up given above constructed in Flowmaster and several analyses run in order to fine tune the EVRV according to the data given above.

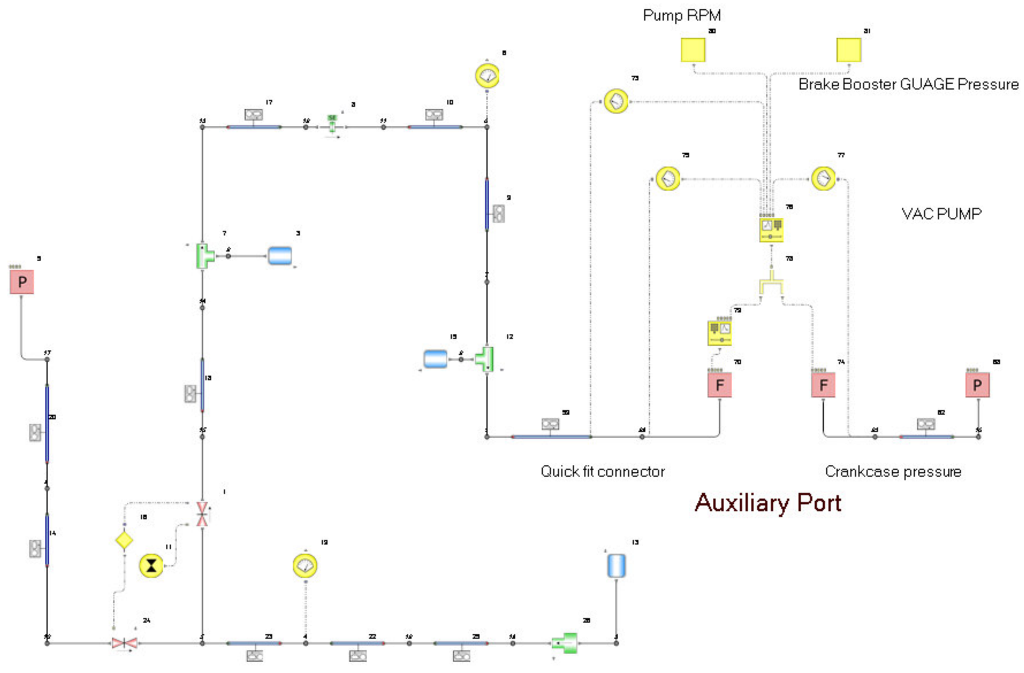


Figure 4.14 : Flowmaster model of the test set-up for fine tuning the EVRV

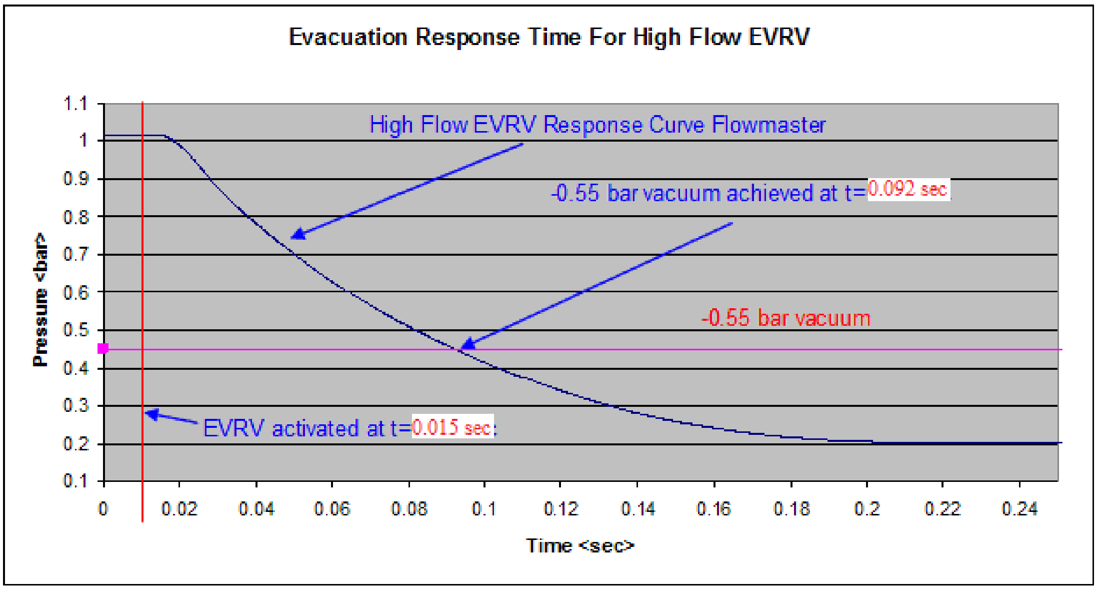


Figure 4.15 : The response time of the EVRV measured as 77 ms

4.7 Actuator Model:

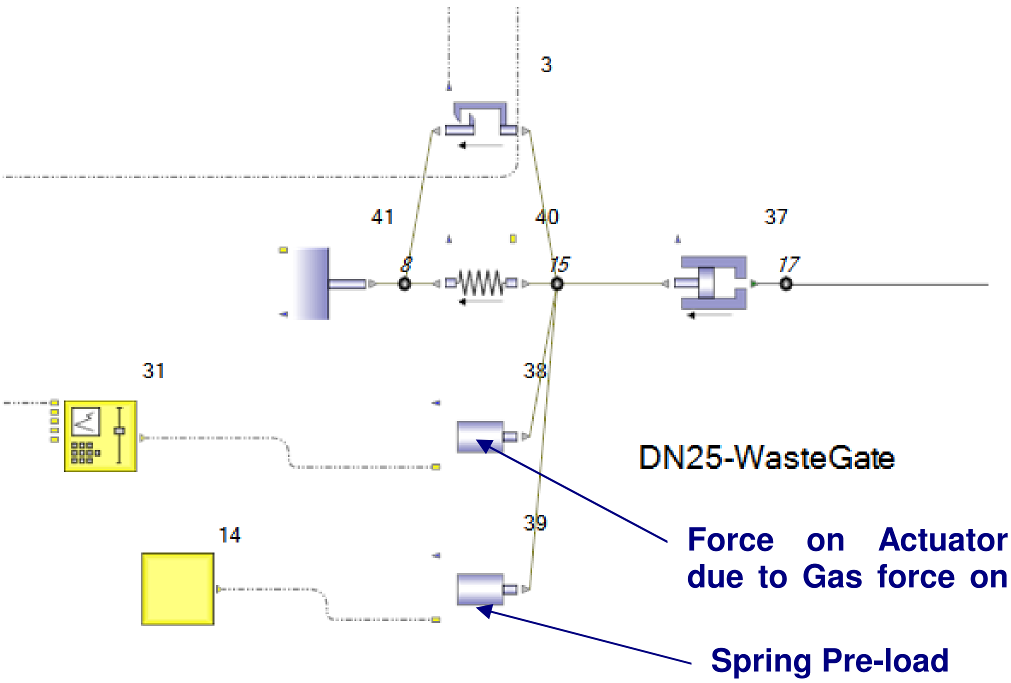


Figure 4.16 : Vacuum actuator 1D CFD model in Flowmaster

Electro-mechanical components and compressible analysis components of Flowmaster assembled to simulate the pneumatic actuator.

Table 4.2: DATA of the selected actuators for TBV and WG

	DN80=TBV	DN25=WG
Initial volume (cc)	235	80
Effective Area (cm^2)	46	18
Spigot Diameter (mm)	4	3
Stroke(mm)	22	14
Spring Rate(N/mm)	6	5.5
Max Gas Force on Flap Valve (N)	100	49

Force on the flap valves also modeled. Force on flap valve calculated by supplier BorgWarner for TBV and Waste Gate valves. See figures given below for forces on flap valves.

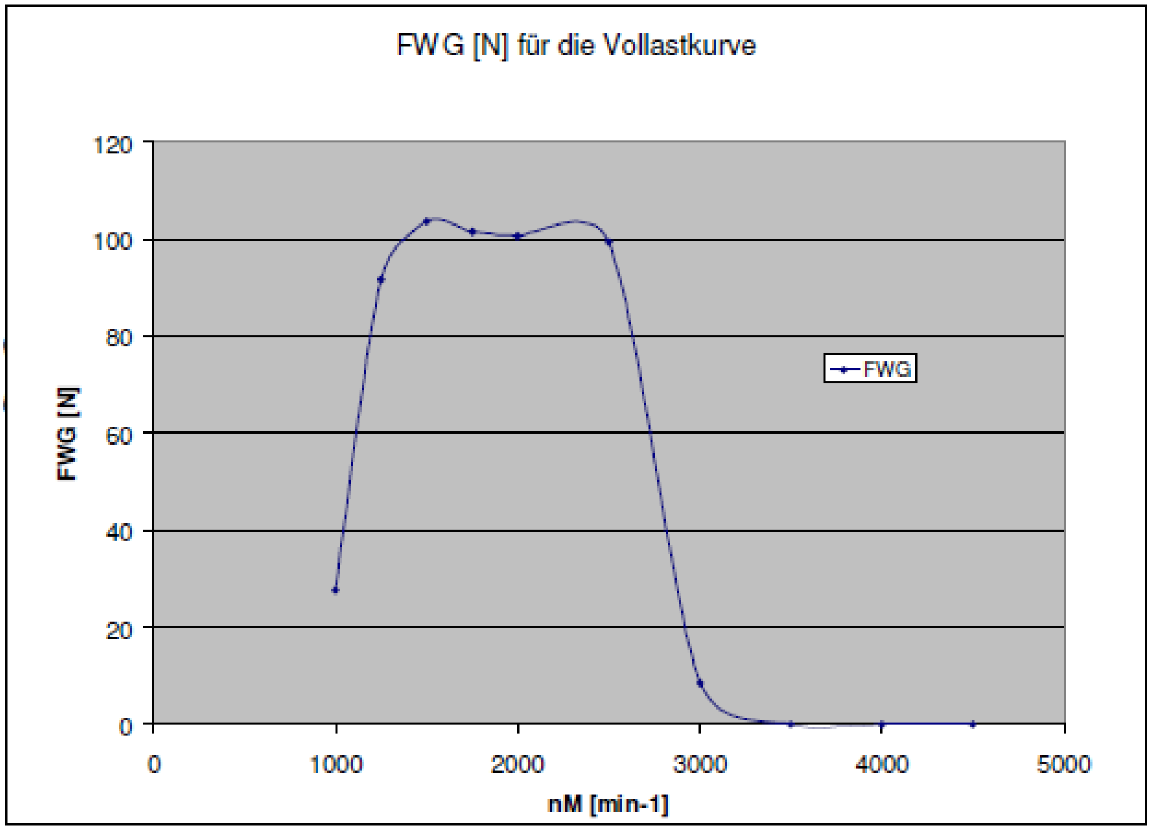


Figure 4.17 : Gas force on turbin bypass valve

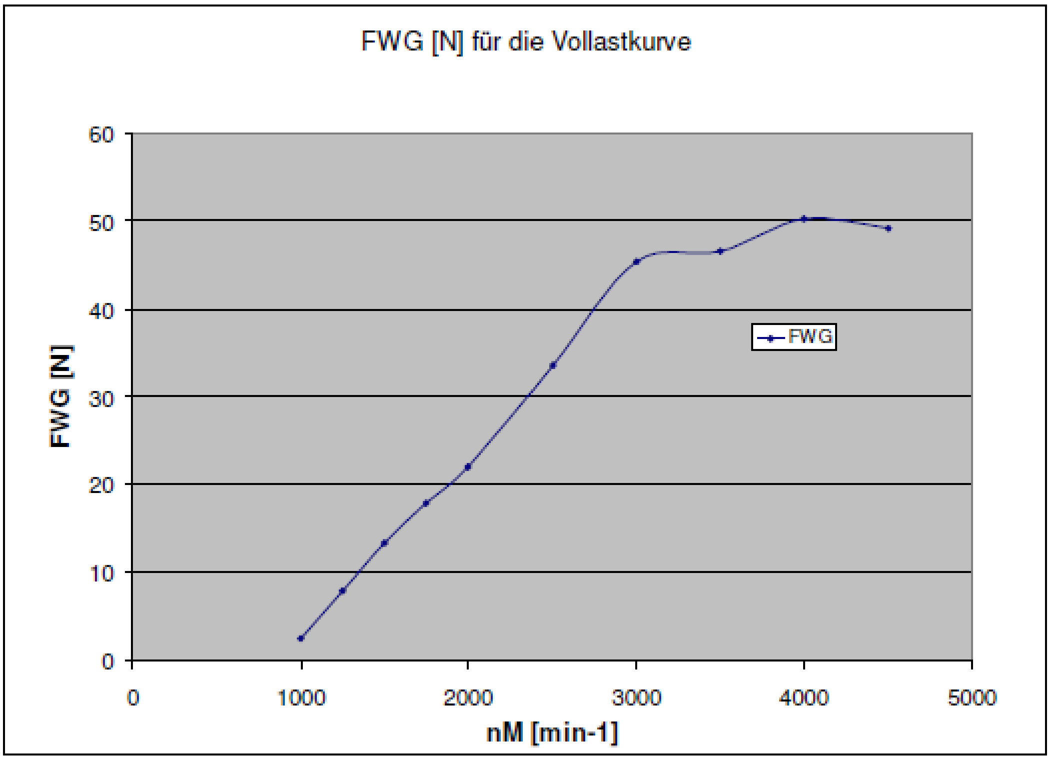


Figure 4.18 : Gas force on waste gate valve

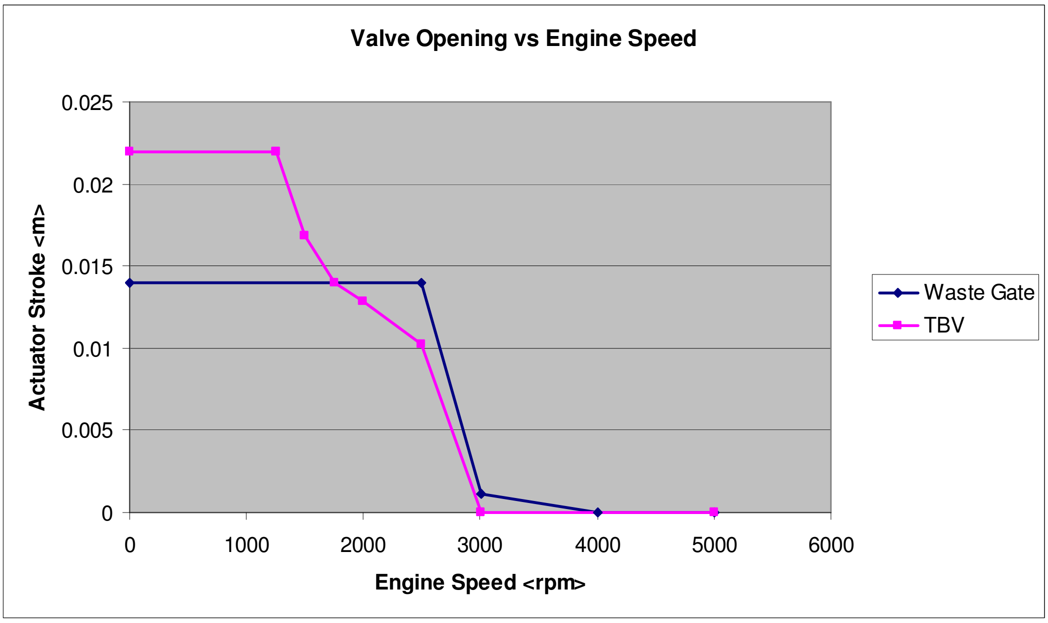


Figure 4.19 : Actuator opening vs. engine speed

Note: In Flowmaster neutral position of the actuator taken as reference point, and when vacuum applied to the actuator the stroke goes to -22 mm beginning from 0 mm. 22mm stroke in the curve given above equals to –22mm stroke in Flowmaster.

4.7.1 Correlation of actuator model:

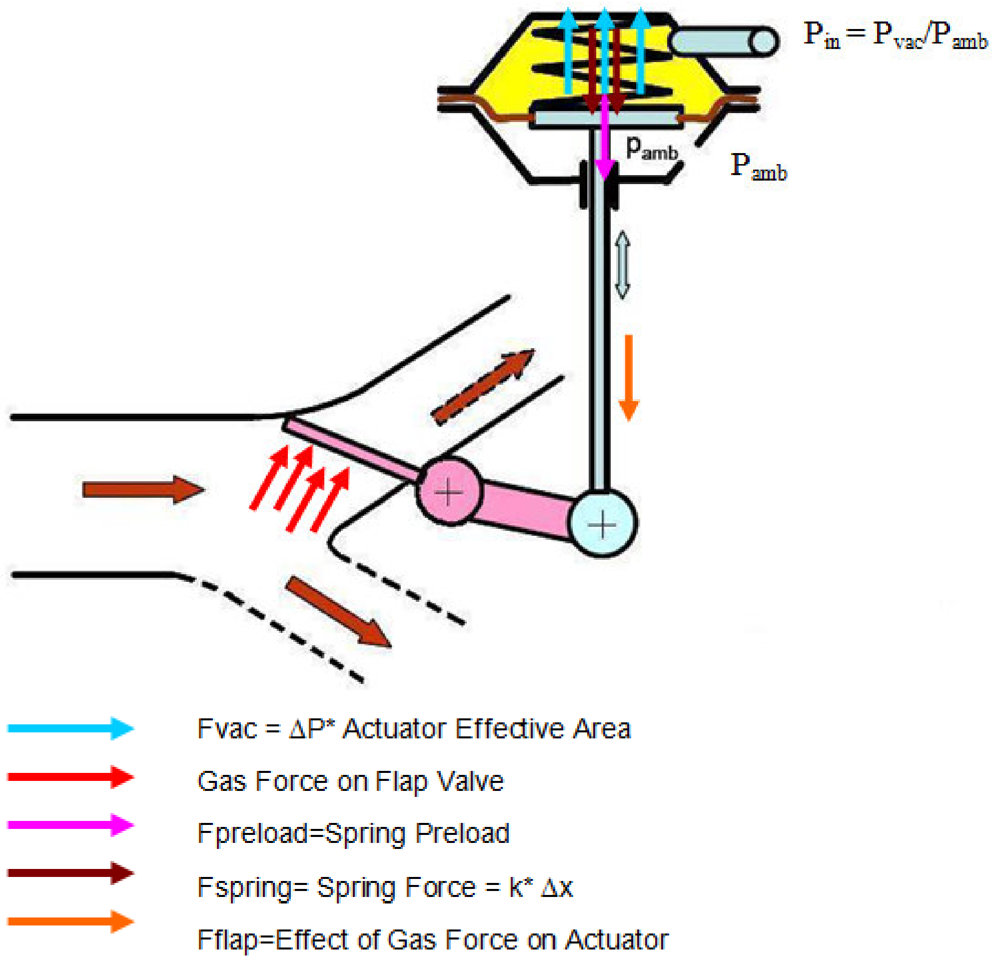


Figure 4.20 : Forces that affect on actuator

The equation given below must be satisfied for all actuators during steady state condition:

$$\text{Fvac} \geq \text{Spring Preload} + \text{Spring Force} + \text{Effect of Gas Force on Actuator} \qquad \textbf{(4.14)}$$

<u>At 1200 rpm</u>:

Fflap = 79.2N

Fpreload = 12N

Spring Rate=k= 6 N/mm

Δx = 22mm

Fspring= 6*22 = 132N

ΔP = Pin-Pamb= 506-1000= 494 mbar =4.94 x 10^{-4} Pa

Actuator Effective Area= 46 cm^2=4.6*10^{-3} m^2

Fvac = 4.94* 10^4 * 4.6* 10^{-3}= 232.76 N

From Equation 4.14:

232.76 >= 132 + 12 + 79.2= 223N

As it seen above the equation 1 is satisfied for at 1200 rpm.

4.8 Flowmaster Model of the Engine Vacuum Harness System:

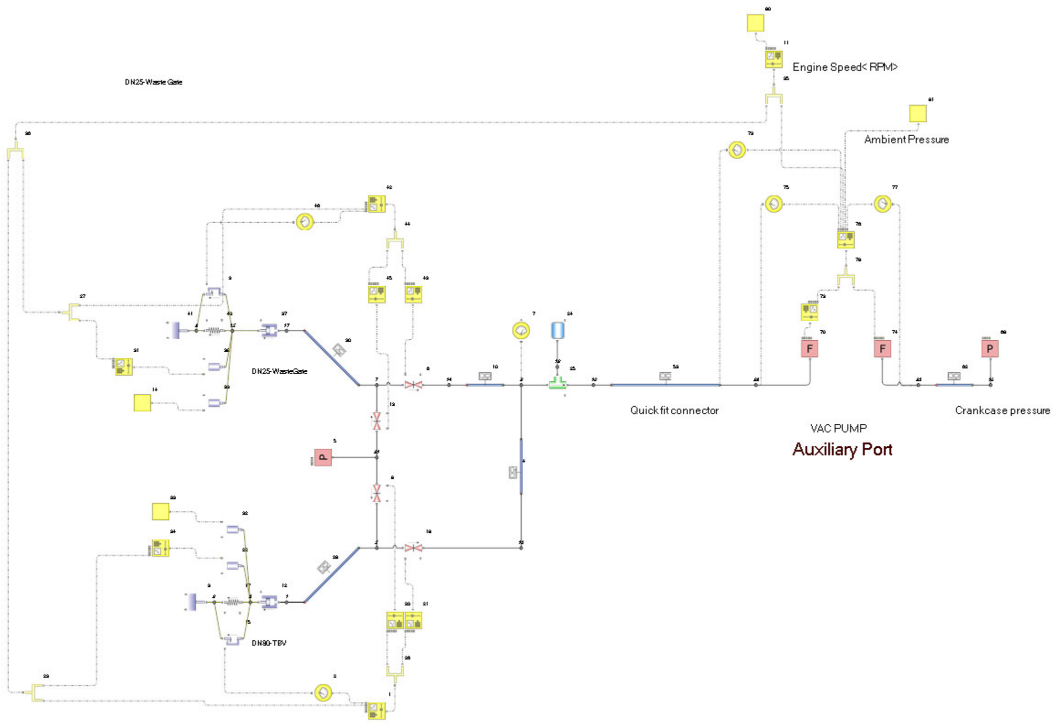


Figure 4.21 : 1D CFD model in flowmaster

4.9 Analysis

1) Vacuum Pump performance data of the rig test was used.

2) EVRV response time correlated to rig test given above and only high flow EVRV used for the modeling due to big sizes of the actuators.

3) Vacuum harness pipe lengths, connectors modelled on the initial design of the engine vacuum system.

4) Actuator specifications and force on flap valve provided by BorgWarner.

5) Spring rate, spring pre-load on the actuator and force on flap valve were modeled.

6) Pump Speed = ½ *Engine Speed

7) Engine speed was calculated from the 1-2, 2-3 and 1-3 gear shifts at engine speeds which are considered as the worst case scenarios for the switching.

8) There may not be a 1-3 gear-shift but it was added to the analysis because 1-3 gear shift covers whole range of the TBV actuator.

Table 4.3: Calculated engine speed data

Gear-Shift	Initial Speed <rpm>	Final Speed <rpm>	Time for Gear-Shift <sec>	Gear-Shift
1-->2	2203	1200	0.4	**1-->2**
2-->3	1961	1200	0.4	**2-->3**
1-->3	3000	1000	0.4	**1-->3**

9) Brake booster assumed as fully evacuated at the start of transition (0.95 bar vacuum) which is the best case,

10) Sea level (1 bar ambient)

11) Actuators represented by compressible cylinders at correct actuator volume.

12) Assumes no restrictor in auxiliary vac pump port

4.9.1 "1-2" gear-shift

Initial engine Speed: 2200rpm

Final engine Speed: 1200 rpm

Atmospheric pressure: 1 bar

Altitude: Sea level

Initially the analysis run with the 2200 rpm until the system reaches to steady state condition which is 2.6 seconds after the start. And then the engine speed reduced from 2200 rpm to 1200 rpm in 0.4 seconds which is assumed the time required during gear shift for the engine speed to achieve the calculated speed at higher gear.

Conclusion:

The response time for this transition is: 0.385 sec

Required vacuum level: 494 mbar

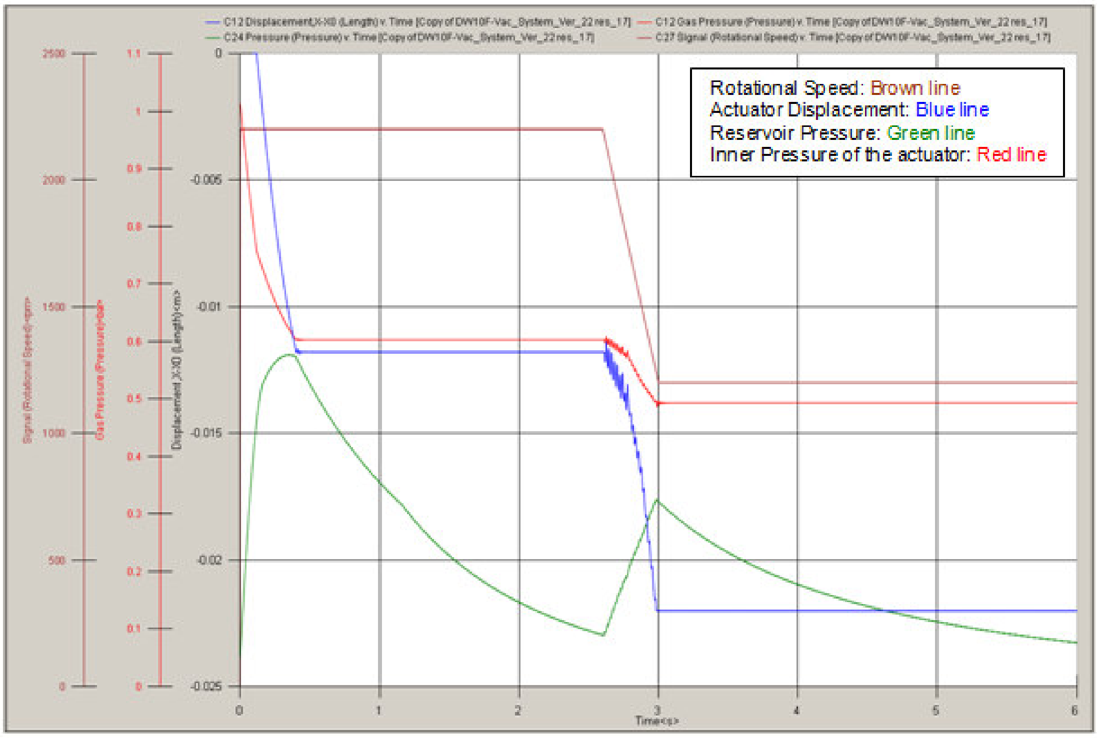


Figure 4.22 : Flowmaster results for 1-2 Gear-Shift

4.9.2 "2-3" gear-shift

Initial engine Speed: 1961rpm

Final engine Speed: 1200 rpm

Atmospheric pressure: 1 bar

Altitude: Sea level

Initially the analysis run with the 1961 rpm until the system reaches to steady state condition which is 2.6 seconds after the start. And then the engine speed reduced from 1961 rpm to 1200 rpm in 0.4 seconds which is assumed the time required during gear shift for the engine speed to achieve the calculated speed at higher gear.

Conclusion:

The response time for this transition is: 0.382 sec

Required vacuum level: 494 mbar

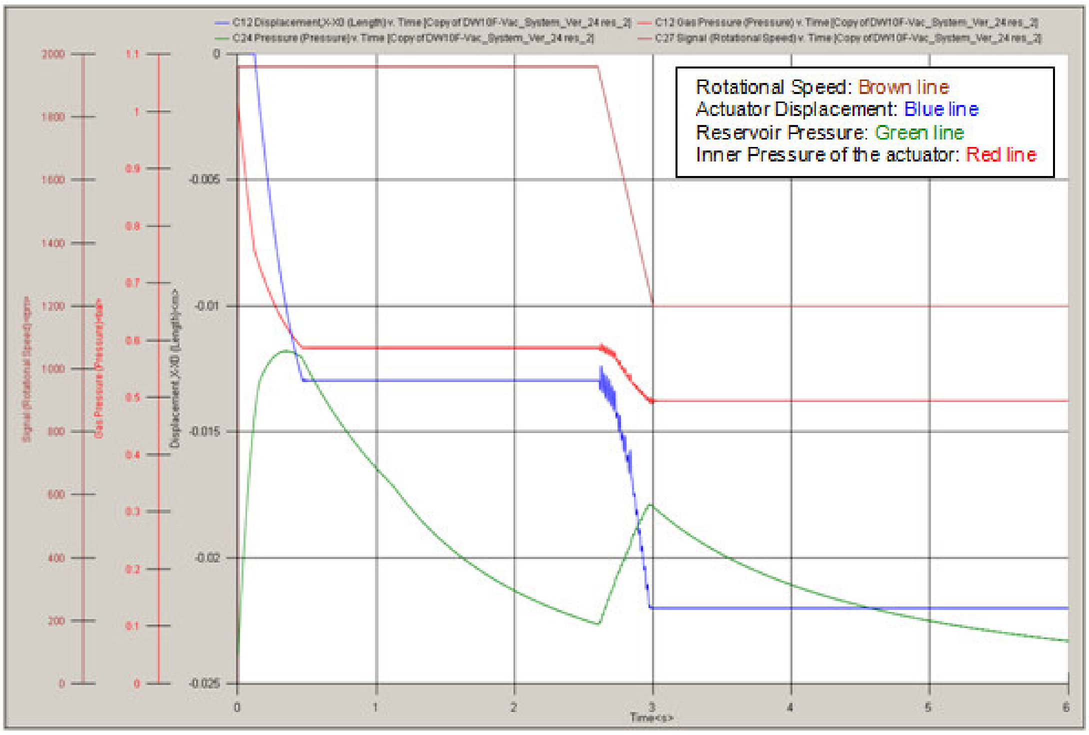


Figure 4.23 : Flowmaster results for 2-3 Gear-Shift

4.9.3 "1-3" gear-shift

Initial engine Speed: 3000rpm

Final engine Speed: 1000 rpm

Atmospheric pressure: 1 bar

Altitude: Sea level

Initially the analysis run with the 3000 rpm until the system reaches to steady state condition which is 2.6 seconds after the start. And then the engine speed reduced from 3000 rpm to 1000 rpm in 0.4 seconds which is assumed the time required during gear shift for the engine speed to achieve the calculated speed at higher gear.

<u>Conclusion:</u>

The response time for this transition is: 0.578 sec

Required vacuum level: 363 mbar

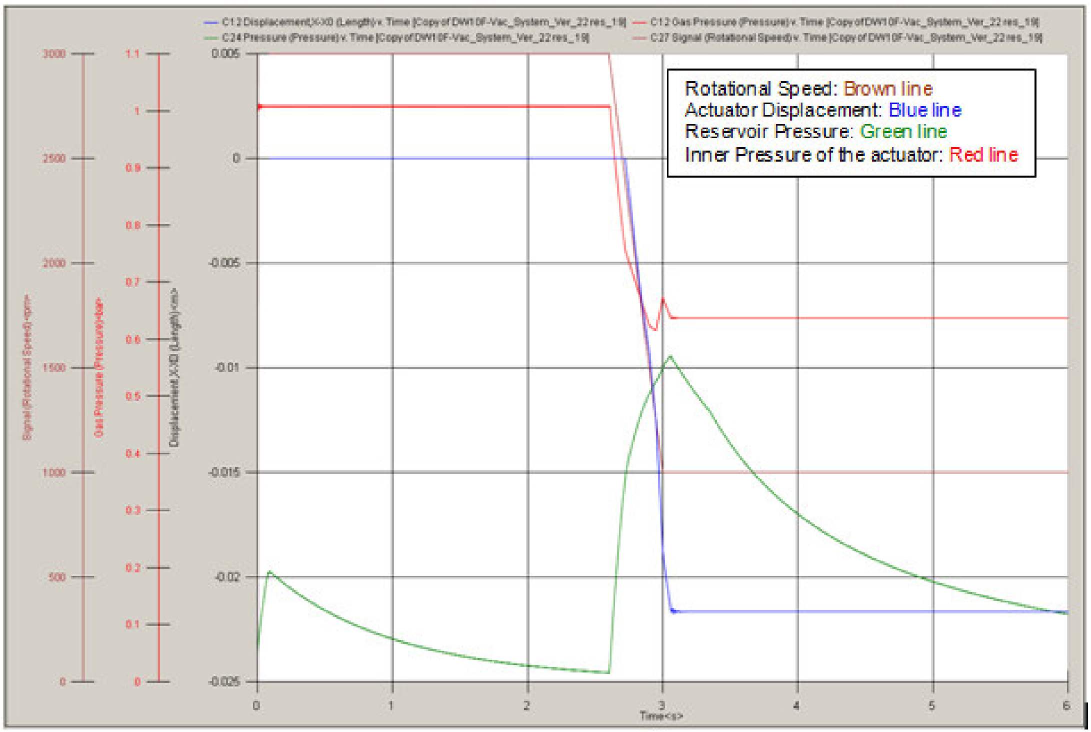


Figure 4.24 : Flowmaster results for 1-3 Gear-Shift

4.9.4 Engine start-up

Due to fail-safe open strategy of the turbin bypass and waste gate valves, it is required to close these valves after engine start-up. A simulation test was run in FLOWMASTER to determine the time taken for the TBV, Waste Gate and EGR valve to close on engine start-up.

Engine Speed = 750 rpm (idle)

Atmospheric pressure: 1 bar

Altitude: Sea level

The volume of the EGR actuator added to the reservoir volume because response time of the EGR is not critical to smooth transition.

<u>Conclusion</u>:

The response time for TBV: 1.328 sec

Required vacuum level: 313 mbar

The response time for Waste Gate: 1.878 sec

Required vacuum level: 526 mbar

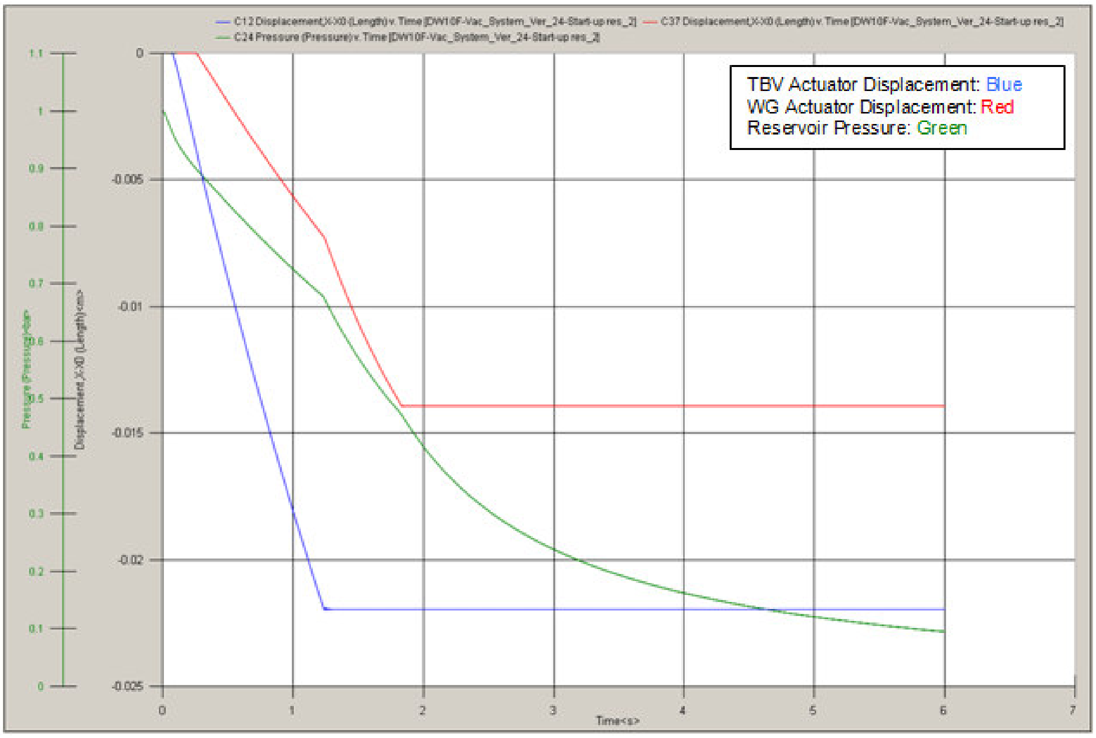


Figure 4.25 : Flowmaster results for engine start-up

4.9.5 Effect of altitude on response time

With the increasing altitude the ambient pressure drops. This change in the ambient pressure has a direct effect on the vacuum harness system, because the performance of the vacuum pump drops dramatically as the altitude increases.

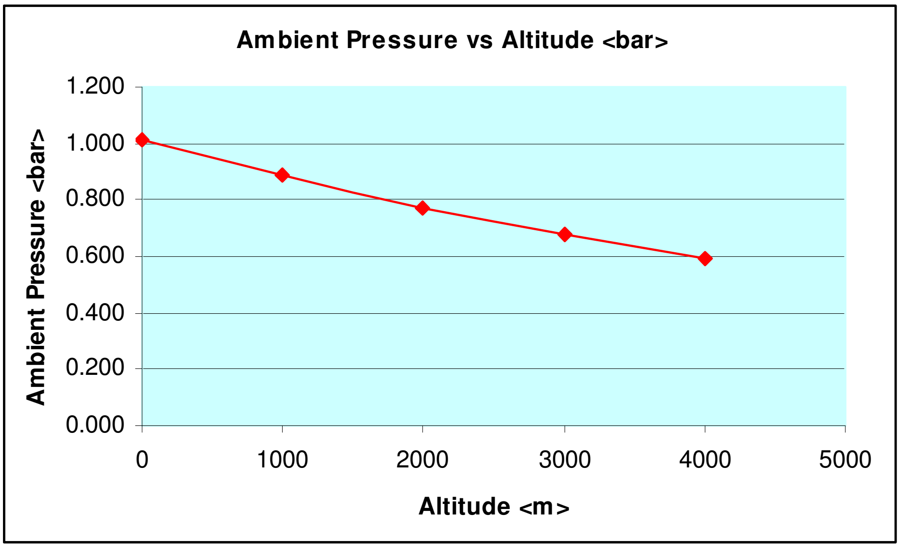


Figure 4.26 : Altitude vs. Ambient Pressure

4.9.5.1 "1-3" gear-shift

Initial engine Speed: 3000rpm

Final engine Speed: 1000 rpm

Altitude and ambient pressure: see figure given above

Initially the analysis run with the 3000 rpm until the system reaches to steady state condition which is 2.6 seconds after the start. And then the engine speed reduced from 3000 rpm to 1000 rpm in 0.4 seconds which is assumed the time required during gear shift for the engine speed to achieve the calculated speed at higher gear.

Table 4.4: TBV response time vs. altitude

Altitude <m>	Ambiant Presuure <bar>	TBV Response Time <sec>	Line Colour on Curve
0	1.013	0.578	Light Blue
1000	0.885	0.64	Brown
2000	0.773	0.855	Green
3000	0.675	1.304	Red
4000	0.590	2.039	Blue

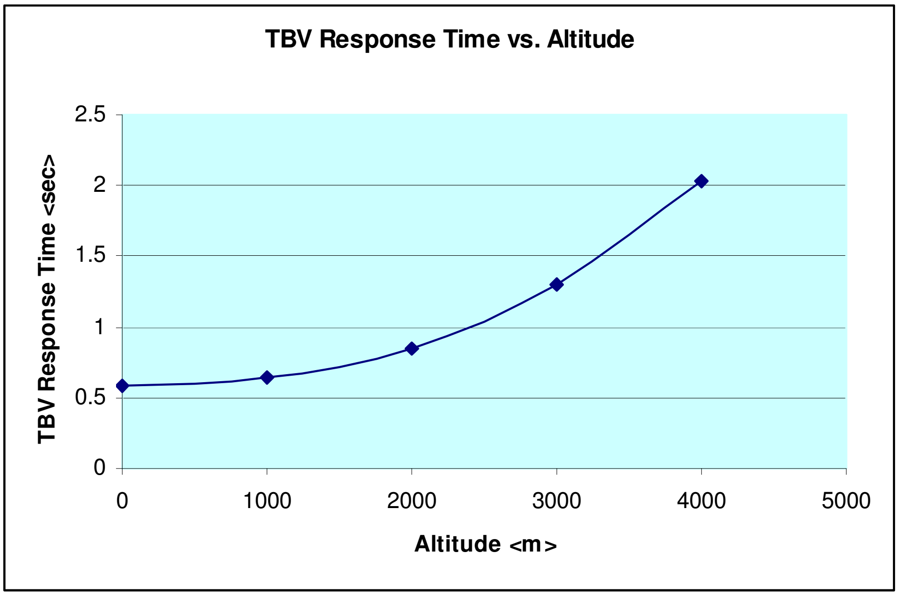


Figure 4.27 : TBV response time vs altitude

The performance of the system decreases with the increasing altitude. If the vehicle has such a gear-shift, the vacuum harness system will not be able to close the TBV valve on time.

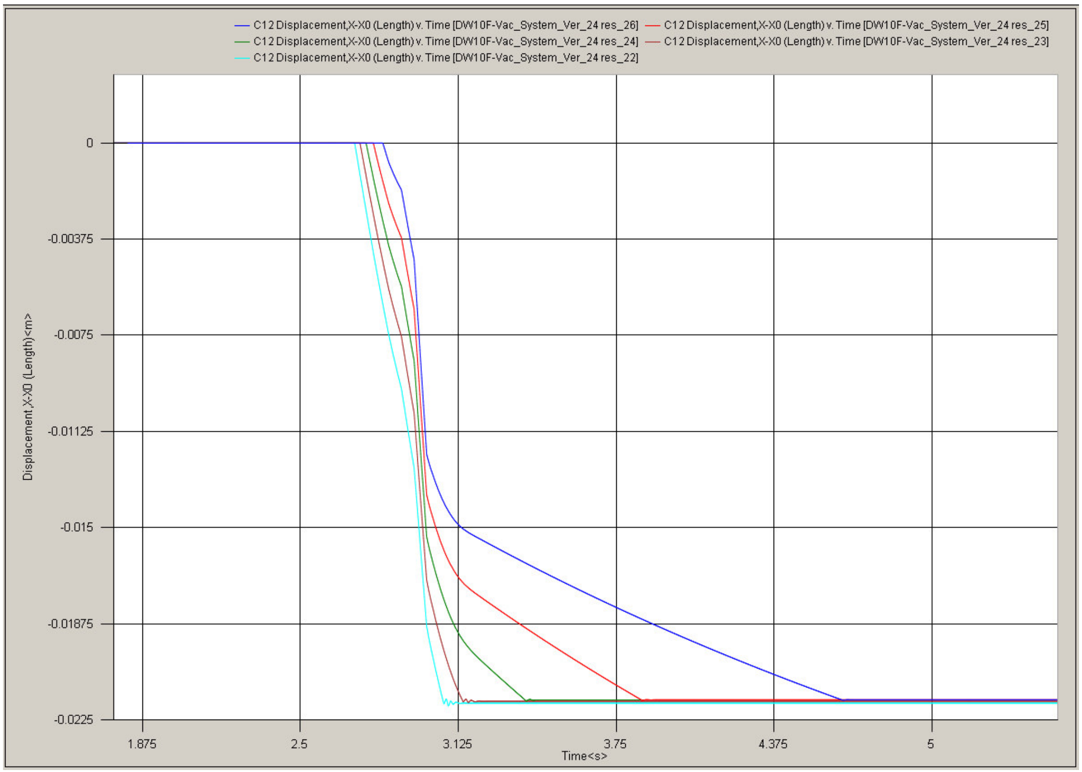


Figure 4.28 : The response time of the TBV @ sea level, 1000, 2000, 3000, 4000 m altitudes for 1-3 Gear shift

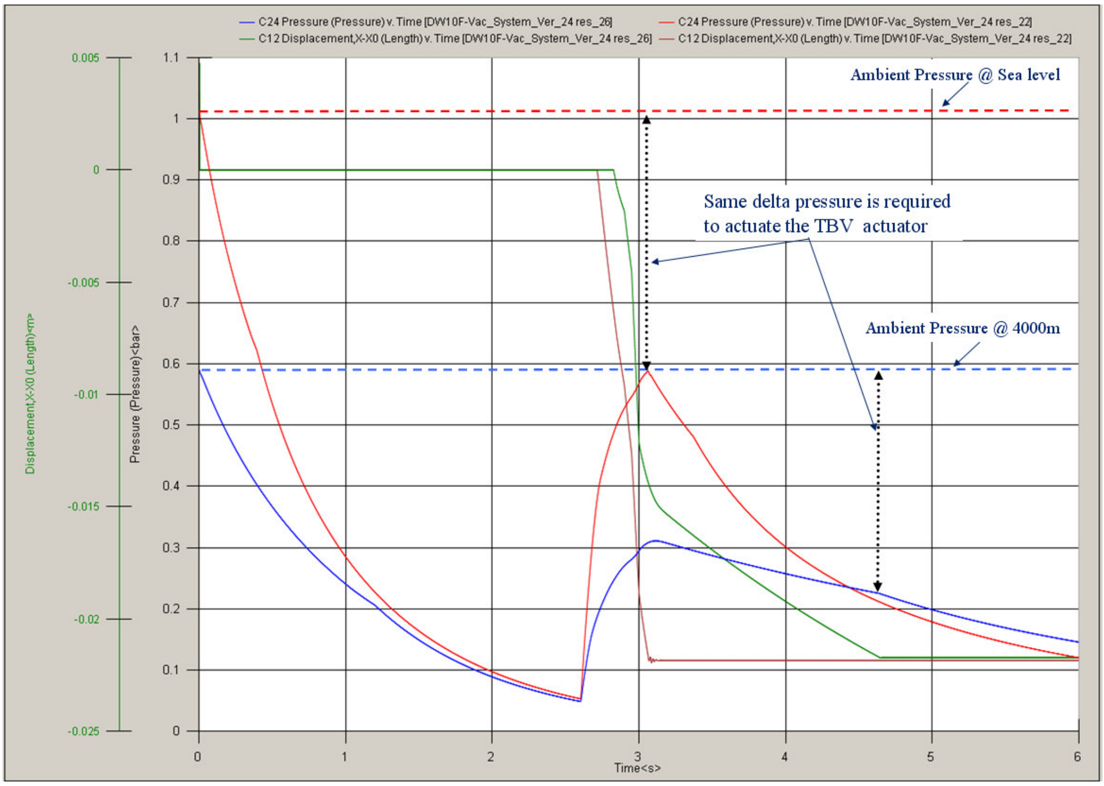


Figure 4.29 : Performance data @ sea level vs. performance data @ 4000m

4.9.5.2 "1-2" gear-shift

Initial engine Speed: 2200rpm

Final engine Speed: 1200 rpm

Altitude and ambient pressure: see figure given above

Initially the analysis run with the 2200 rpm until the system reaches to steady state condition which is 3.6 seconds after the start. And then the engine speed reduced from 2200 rpm to 1200 rpm in 0.4 seconds which is assumed the time required during gear shift for the engine speed to achieve the calculated speed at higher gear.

<u>Conclusion:</u>

Table 4.5: TBV response time vs. altitude

Altitude <m>	Ambiant Presuure <bar>	TBV Response Time <sec>	Line Colour on Curve
0	1.013	0.392	Light Blue
1000	0.885	0.385	Brown
2000	0.773	0.386	Green
3000	0.675	0.469	Red
4000	0.590	1.824	Blue

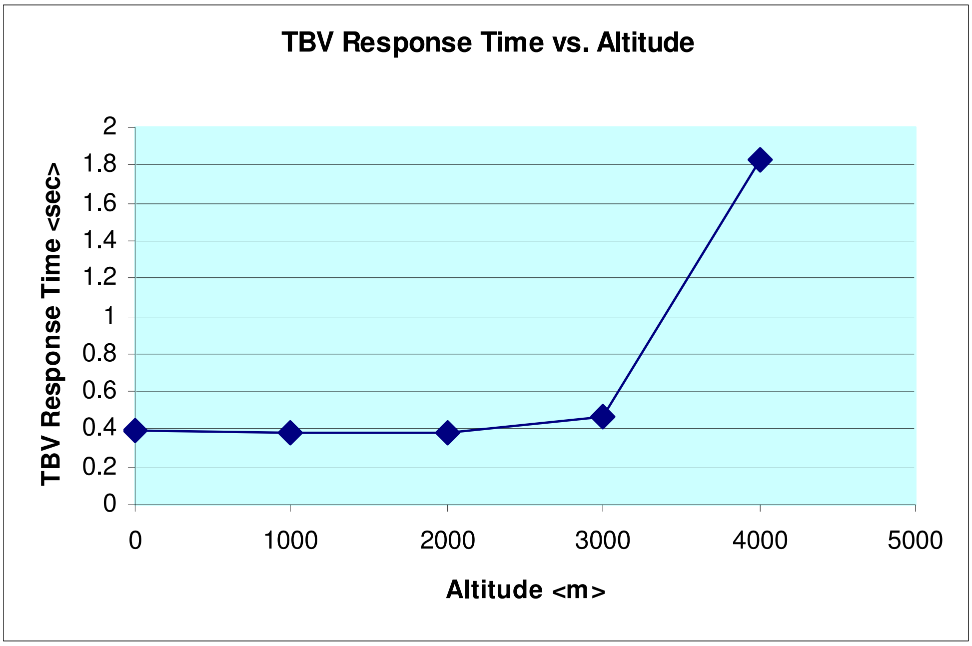


Figure 4.30 : TBV response time vs. Altitude

Although the performance of the system decreases with the increasing altitude, the response time for the TBV is sufficient up to 3000 m altitude. But After 3000 m altitude the response time of the TBV becomes critical.

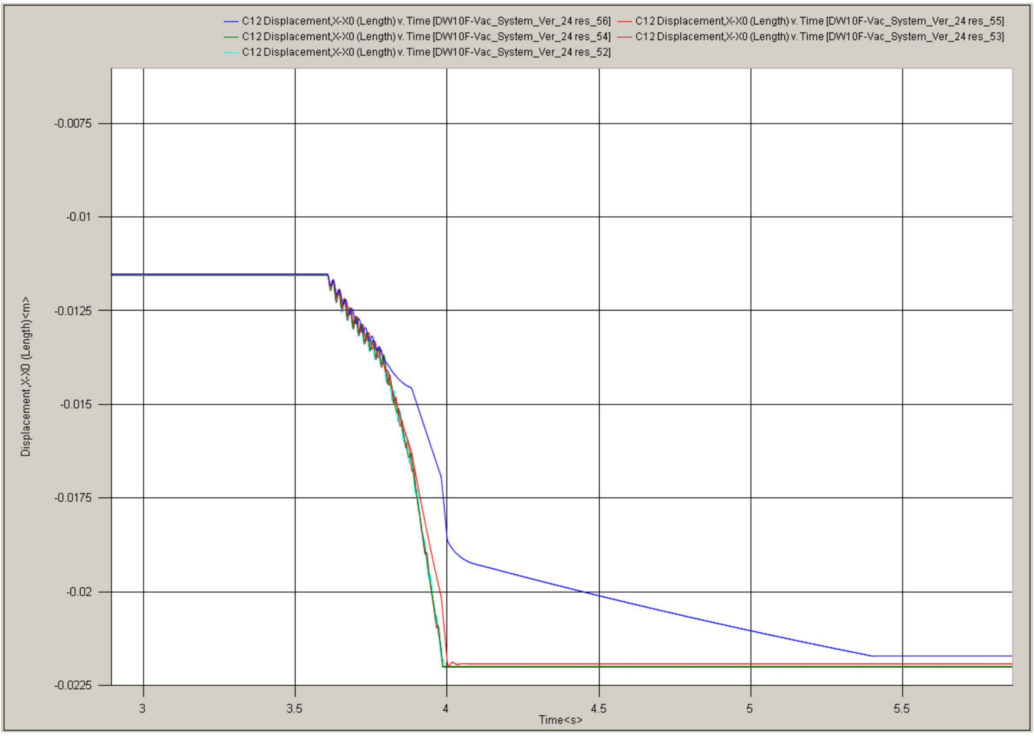


Figure 4.31 : The response time of the TBV @ sea level, 1000, 2000, 3000, 4000 m altitudes for 1-2 Gear shift

4.9.6 Effect of the reservoir volume

4.9.6.1 “1-3” gear-shift

Initial engine Speed: 3000rpm

Final engine Speed: 1000 rpm

Atmospheric pressure: 1 bar

Altitude: Sea level

Reservoir volume: See table given below

Initially the analysis run with the 3000 rpm until the system reaches to steady state condition which is 2.6 seconds after the start. And then the engine speed reduced from 3000 rpm to 1000 rpm in 0.4 seconds which is assumed the time required during gear shift for the engine speed to achieve the calculated speed at higher gear.

Table 4.6: The effect of the reservoir volume on TBV response times

Reservoir Volume <cc>	TBV Response Time <sec>	Line Colour on Curve
500	0.521	Pink
300	0.568	Light Blue
200	0.612	Brown
150	0.698	Green
100	0.778	Red
50	0.861	Blue

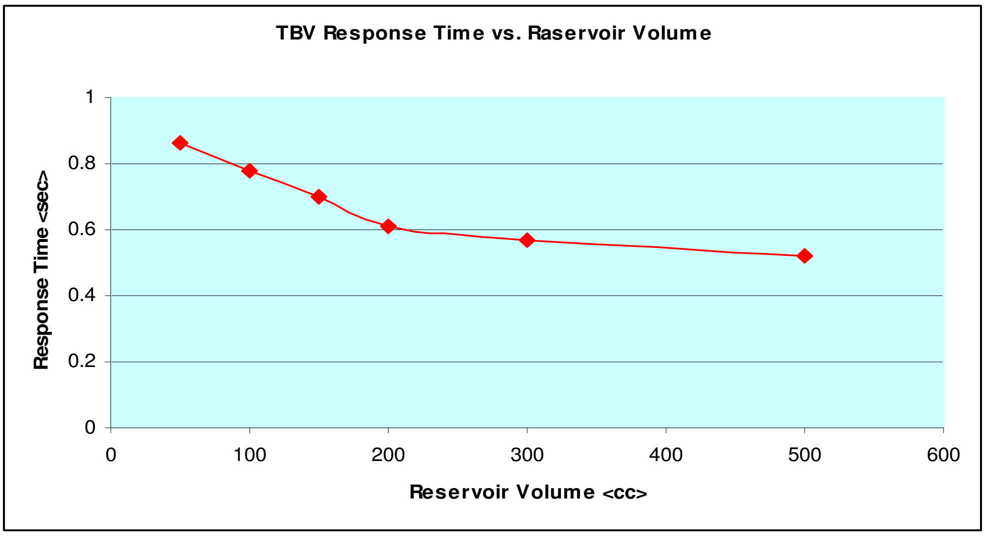


Figure 4.32 : The effect of Reservoir Volume on TBV opening

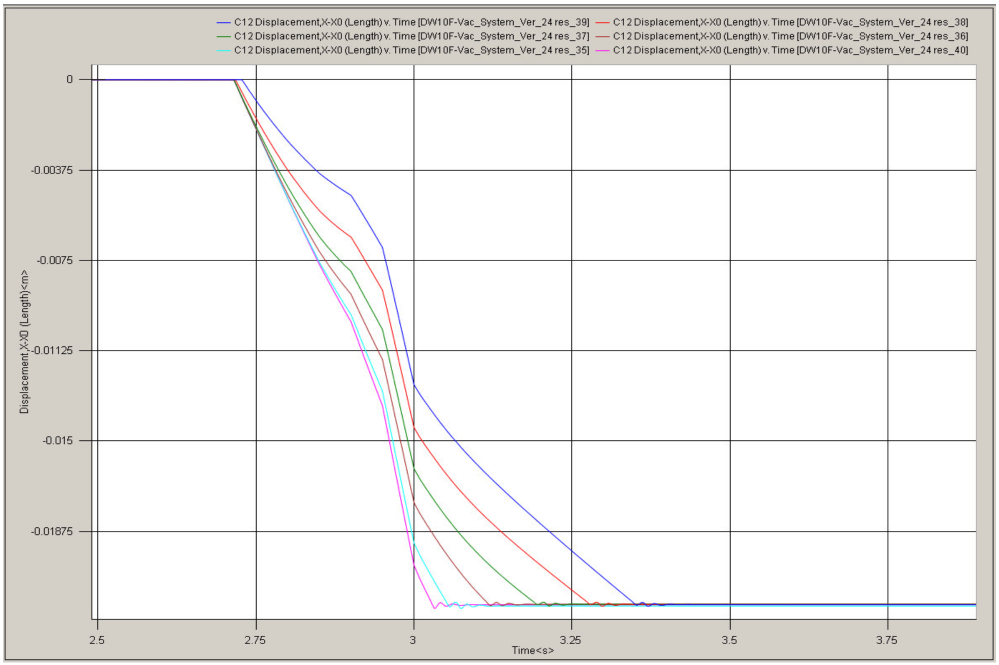


Figure 4.33 : Response curves of the TBV valve at different reservoir volumes

As it seen from the curves the response time of the TBV increases with increasing Reservoir volume. But the effect of the reservoir volume increase is very low for the volumes higher than 200 cc.

4.9.6.2 "1-2" gear shift

Initial engine Speed: 2200rpm

Final engine Speed: 1200 rpm

Atmospheric pressure: 1 bar

Altitude: Sea level

Reservoir volume: See table given below

Initially the analysis run with the 2200 rpm until the system reaches to steady state condition which is 3.6 seconds after the start. And then the engine speed reduced from 2200 rpm to 1200 rpm in 0.4 seconds which is assumed the time required during gear shift for the engine speed to achieve the calculated speed at higher gear.

Table 4.7: The effect of the reservoir volume on TBV response times

Reservoir Volume <cc>	TBV Response Time <sec>	Line Colour on Curve
500	0.381	Pink
300	0.389	Light Blue
200	0.388	Brown
150	0.388	Green
100	0.386	Red
50	0.511	Blue

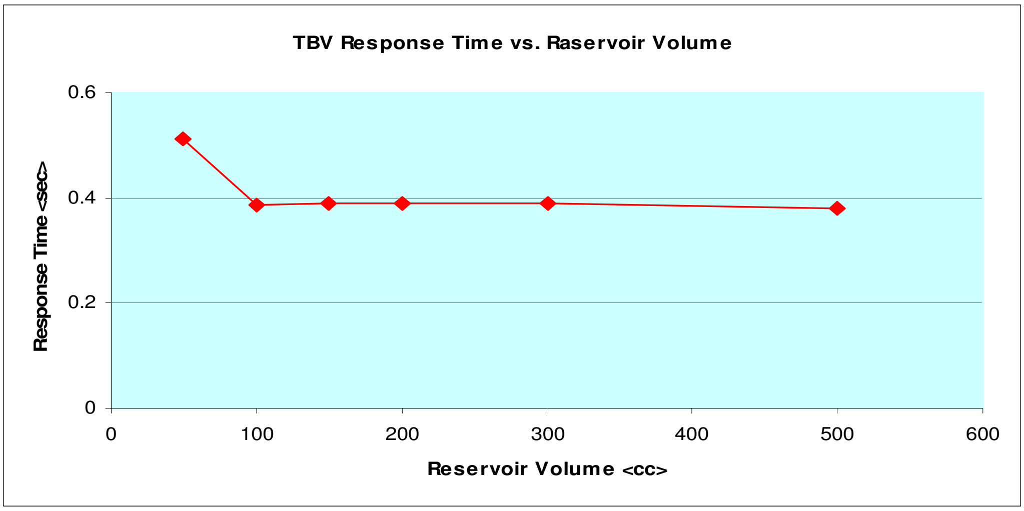


Figure 4.34 : The effect of Reservoir Volume on TBV opening

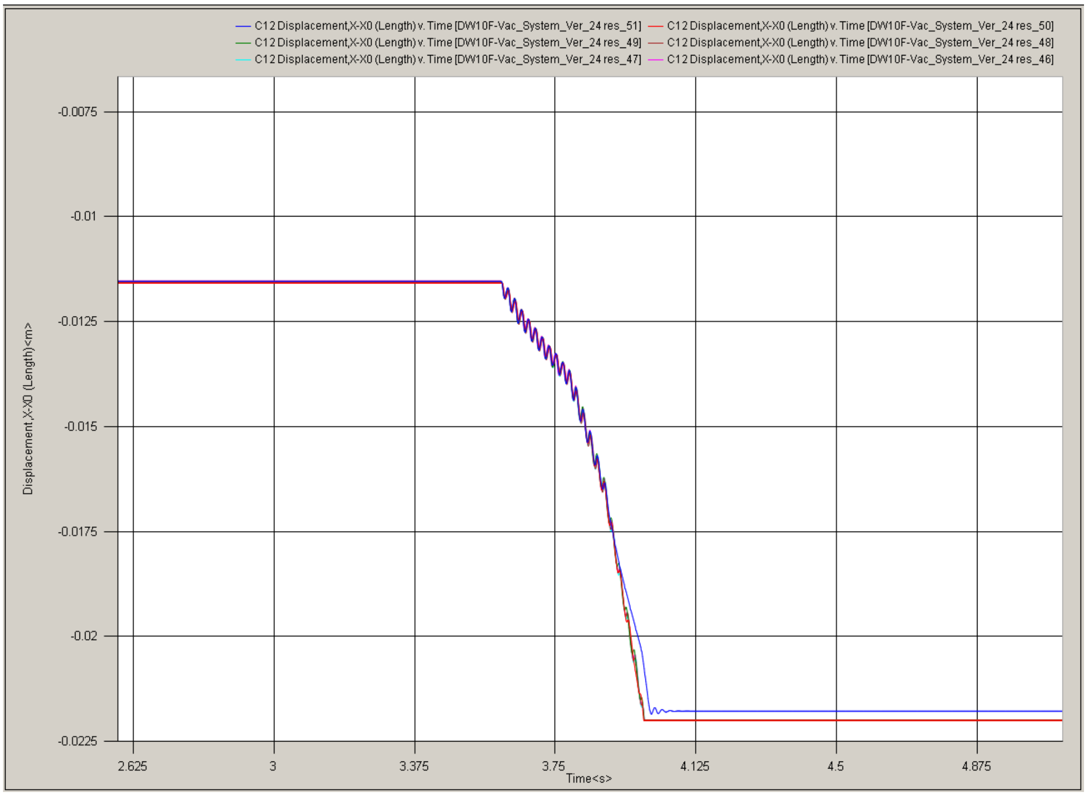


Figure 4.35 : Response curves of the TBV valve at different reservoir volumes

4.9.7 Conclusion:

1. Under all conditions the vacuum system closes the TBV valve. The critical parameter is the response time of the TBV.
2. If the gear shifts done successively (1-2, 2-3, 3-4, 4-5 and 5-6), there will be no problem with the response times of the TBV, i.e. the vacuum harness system will satisfy the required 500ms response time requirement.
3. TBV will be closed in 1.328 seconds at start-up.
4. If the vehicle has a 1-3 gear shift the response time becomes critical after 1000m altitude, else the response time becomes critical after 3000m altitude.
5. There 2 type of the EVRV's which provided by Pierburg which are standard EVRV and high flow EVRV. Due to high reservoir volumes and spring rate of the actuators high flow EVRV's must be selected.
6. 250-300 cc reservoir volume is acceptable for the application.
7. Actuator effective area, spring rate and force on flap valve have a strong effect on the vacuum harness system performance.
8. The effect of the brake booster shall be investigated for the worst case scenarios.

5. EXPERIMENTAL ANALYSIS

5.1 Engine Test Results

Engine tests were performed at Ford Otosan Gölcük plant dyno cells. During the tests, APA204/8 was used which is cradled AC machine with pendulum bearings with squirrel cage rotor, equipped with load cell for torque reading.

Torque capacity : 900 Nm constant up to 3000 rpm.

Power capacity : 270kW between 4000-8000 rpm.

Type of operation : Active and Passive mode (for driving & braking)

Sense of rotation : CW & CCW

Overall accuracy : 0.3% full scale torque +/-1 rpm

Control accuracy : 0.3% torque +/-5 rpm

Torque response : 10-20 ms for 0- maximum torque

Speed response : 5000 rpm/sec for nominal torque

Inertia : 0.94 kg.m^2

Engine which was equipped with the new serial sequential turbo and vacuum system had been run to take measurements from the turbin bypass valve (TBV) in terms of response time and vacuum inside the actuators. The results from the engine is compared with the theoretical results which are attained from 1-d flow analysist.

Engine is not equipped with all real components since the prototype parts are not available yet. To perform the tests some components are replaced with similar ones that have almost same performance., although this has an affect to real performance of the engine.

One of the components that is not availabe on the engine is the solenoid valve which affects the response time of the actuators. In replace of the selected valves, similar solenoid valves are used to simulate the test.

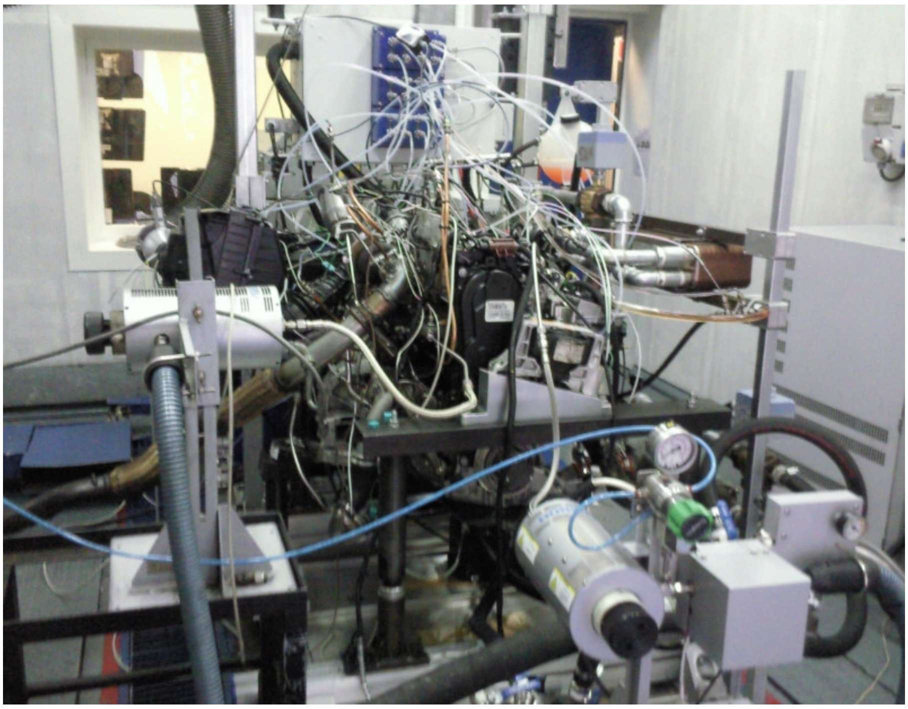


Figure 5.1 : Engine dyno test at Ford Otosan

During the tests, only the response time of TBV valve was read since it is the most critical one that affects the emissions. TBV valve has a capacity of 235 cc while the reservoir in the system has 250 cc volume. Response time values were obtained at idle speed (750 rpm), 1250 rpm and 2500 rpm at ambient pressure 1 atm.

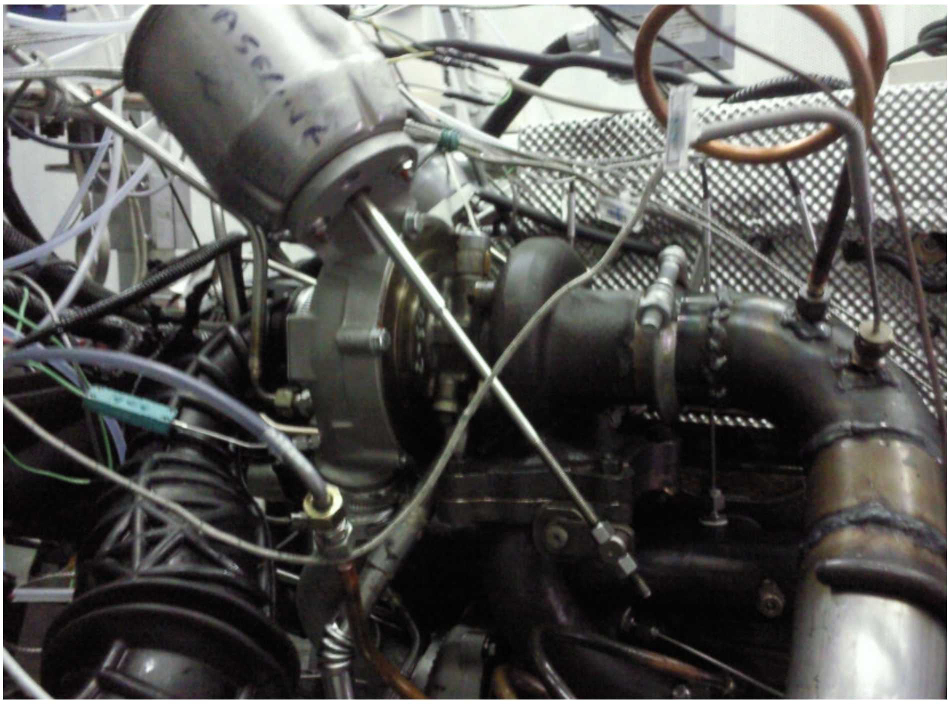


Figure 5.2 : New serial sequential turbo

The target value for turbine bypass valve response time for a full stroke opening is 500 ms. There wasn't any data read during the tests regarding the other valves westgate and egr by pass valve due to less effectiveness to the emission system. The performance of the turbin bypass valve is affected the engine power characteristics and emission values.

Response times were measured at idle speed 750 rpm with ambient pressure 1 atm with different PWM signals. Response times (blue lines) at turbine bypass valve (TBV) were read from the engine as 700 ms - 800 ms.

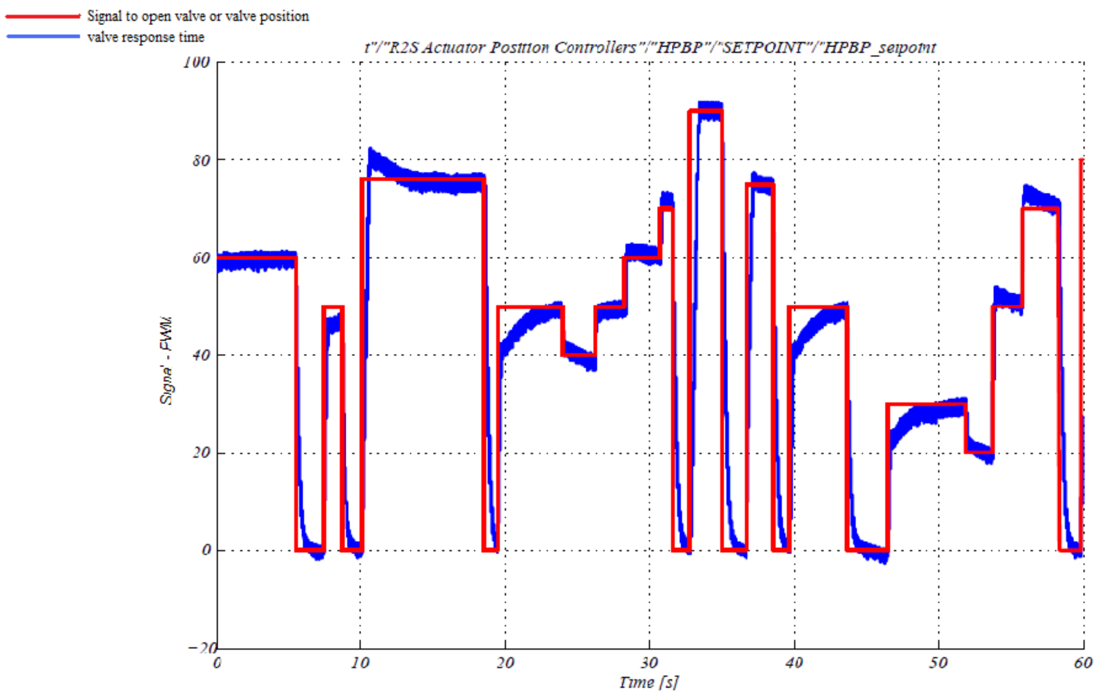


Figure 5.3 : TBV response times @ 750 rpm

After the idle speed, response times were measured at 1250 rpm engine speed at ambient pressure 1 atm. Due to increase in the flowrate of the vacuum pump, it is expected to see shorter response times at TBV valve. Response time on TBV valve was measured around 550 ms which is an acceptable value for the response time of turbin bypass valve .

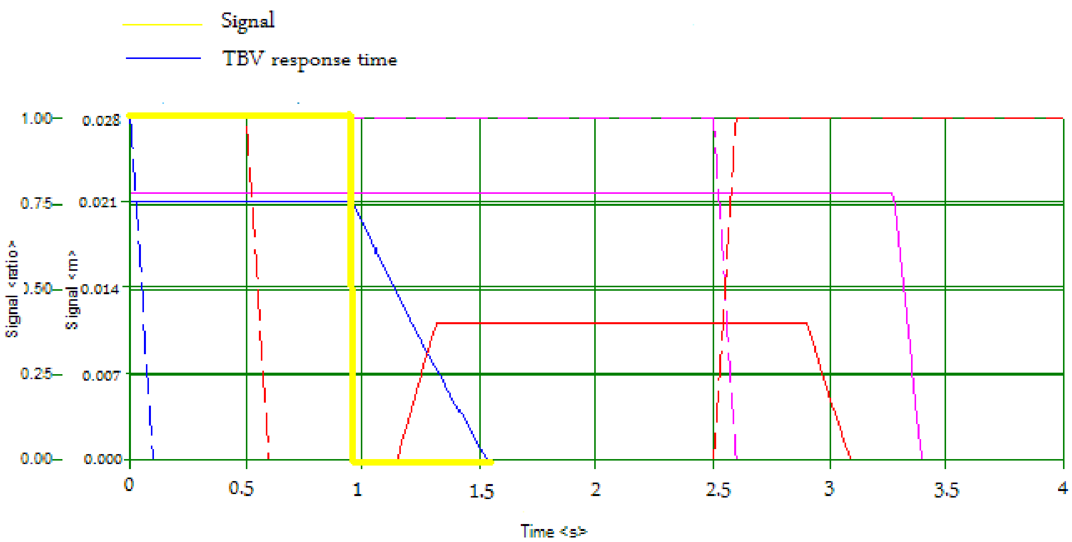


Figure 5.4 : TBV response times @ 1250 rpm

Final measurements were recorded at 2500 rpm with a 250cc reservoir at ambient pressure 1 atm and at higher altitudes. Response time on TBV in these conditions was 500 ms. The vacuum inside the actuator was read as 400 mbar for a full opening. When the ambient pressure drops, such as 0.59 bar, then the response time will increase to 1.5 sec.

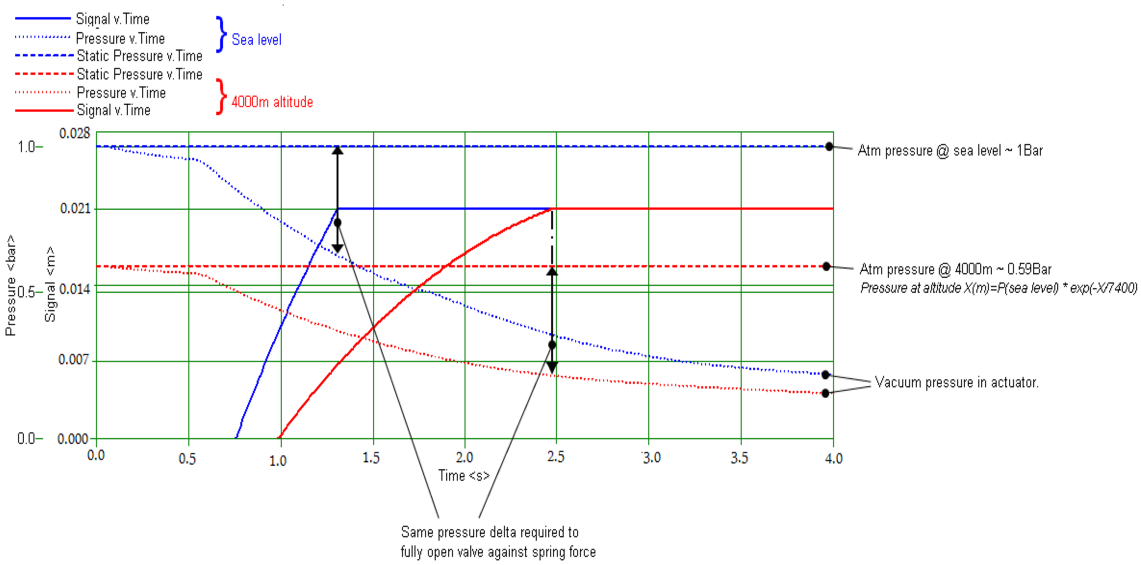


Figure 5.5 : TBV response times @ 2500 rpm

5.2 Conclusion

Engine tests show that 1d simulation model is consistent with the real results and can be useable to predict the performance of the system which also contributes an easy improvement in the current design.

Target value for the TBV response time which is 500 ms has almost met. At idle speed, it is higher than the projected value and when the engine rpm is increased, the response time is getting closer to the targeted value.

When the ambient pressure drops, response times were dramatically decreased. At 0.59 mbar, at 2500 rpm engine speed, the response time is 3 times the value that is targetted. Moreover, the vacuum inside the actuator is also compatible with the results from 1d simulation. Finally, the reservoir volume which is 250 cc does practically work in this system to meet the defined specifications.

Although there is a slice difference between the experimental and simulation results, it can be considered consistent.

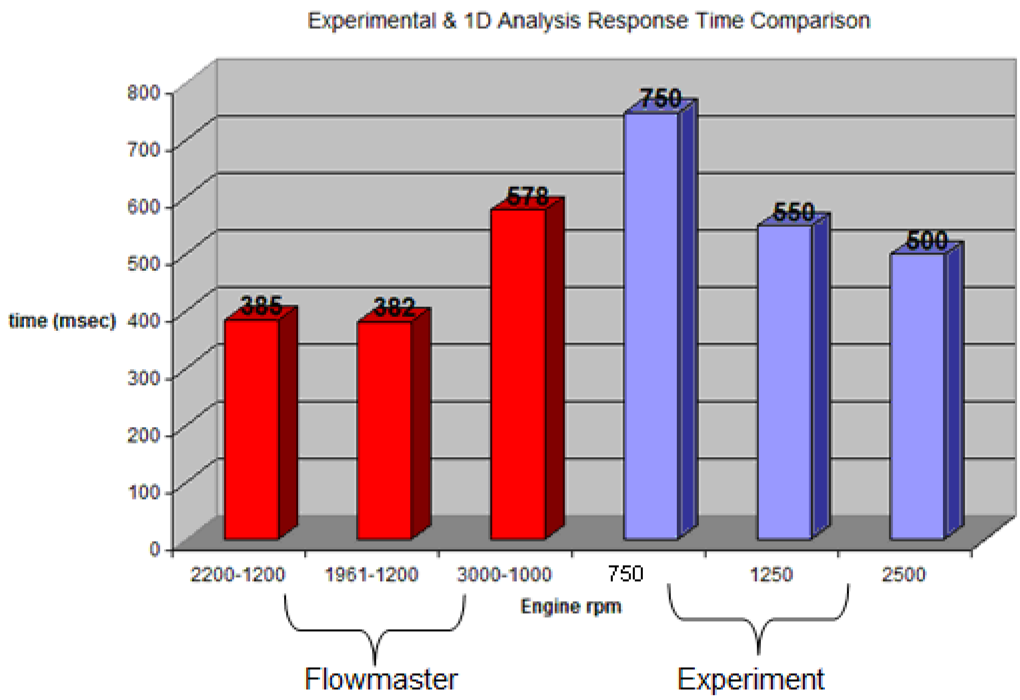


Figure 5.6 : Experimental & 1D analysis results comparison

The reason why the simulation results are a bit different from the test results is the data used in flowmaster. To simulate the vacuum system, it is required experimental inputs that were extracted before. Moreover, some assumptions during the model preperation do some effects to the results. If it is considered that for every component in the model has an experimental input, the results from the engine tests are really similar to each other. The engine used for measurements is the prototype engine and includes lack of components. The valves in the test are not the same but similar. Finally, there were some problems during the calibration of the engine and still some adjustments are required to run the engine at specified conditions. Therefore it can be said that the model can usable for further steps and improvements.

6. CONCLUSION AND RECOMMENDATIONS

The major purpose of this study was to correlate the 1d simulation and experimental work related to the vacuum system used for controlling the actuation of pneumatic valves in internal combustion engines in order to develop and improve system performance in terms of response times without testing or production of the components. Measurements have been done to verify the implementation. Due to unfortunate circumstances only a limited number of measurements could be done.

After the correlation of the 1D model with engine tests, by putting different variations to the system, it would be easy to see what affects the performance and what parameters should be changed. This will also contribute to the engineer to make a cost reduction during the optimization besides time reduction.

This study shows that the experimantal results are nearly same with results from 1D simulation analysis. As a next step of the study, after the correlation process, the parameters that affect the performance of the system can be evaluated in the simulation which are proposed in the 1D model section and it is seen or understood the impact to the system. So that, it would be much cheaper and easy to find the optimized design for the system. To improve the system performance and capability, high flow EVRVs can be selected. Using a bigger reservoir than 250 cc does not affect the system performance excessively. In case of selecting a bigger reservoir causes only higher costs, extra mass and package issue on the engine. Another recommendation for the system improvement is the spring stiffness of the vacuum actuator. Lower spring stiffness values mean lower response times. However, lower values can cause fluctuation of the valve which is not a desired condition during turbo is running. So this should be considered during the improvement phase. Another parameter for the system improvement is the vacuum pump which is the flow source of the system. To design and produce a vacuum pump is the most difficult and expensive component in this design. If the targets are hardly met in the system, there is no need to change the vacuum pump. Otherwise, it is necessary to use a high flow performance vacuum pump which has higher flow rates for the auxiliary circuit.

In addition to the physical components on the engine, ambient conditions are also effective to the performance of the vacuum system. At higher altitudes, since the desired vacuum is less than the value at sea level, the response times are analyzed and measured higher than the normal conditions.

The current implementation of the models should be investigated with more measurements to discover the cause(s) of the slice current difference between the simulations and the measurement. The pumping rate or gas flow should be measured, in order to exclude an incorrect implemented pump characteristic. Next to the pressure in the reservoir, the pressure and temperature should be measured on more locations of the vacuum system. Based on the current pipe component, different shaped and dimension pipes should be made. Also the effects of the apertures of small pipes should be taken into account.

REFERENCES

[1] **Galindo, J., Climent, H., Guardiola, C., and Domenech, J**., (2009). Modeling the Vacuum Circuit of a Pneumatic Valve System. *Journal of Dynamic Systems. Vol.131***.**

[2] **Pfeiffer Vacuum GmbH**, (2009). Vacuum Technology Know How, pp. 8-28.

[3] ***C*hew, A. D.,** (2005). Mechanical Vacuum pumps. *BOC Edwards*. Crawley, United Kingdom.

[4] **Mans, J.,** (2006). Vacuum Physics and Technology.*Phyisics 4052.*

[5] **Marquardt, N.**. Introduction to the Principles of Vacuum Physics. *Institute for Accelerator Physics and Synchrotron Radiation.* University of Dortmund. 44221 Dortmund, Germany.

[6] **Rack, P.,** Vacuum Technology. *Introduction to Materials Science and Engineering Lecture Notes.* University of Tennessee, Knoxville.

[7] **O'Keefe Controls Co**., (2000). Choked Flow of Gases.

[8] **Umrath, W.,** (1998). Fundementals of Vacuum Technology.

[9] **Nagendiran, S. R., Sivanantham, R., and Suresh Kumar, J.,** (2009). Improvement of the Performance of Cam-Operated Vacuum Pump for Multi Jet Diesel Engine, 2009-01-1462.

[10] **Url-1** < *www.autoshop101.com/forms/brake05.pdf*>, accessed at 15.10.2010.

[11] **Miyake, T. and Okada, H..** An Electrically Operated EGR Valve that Reduces Automotive Emissions. *Technical Reports*.

[12] **Url-2** < *http://en.wikipedia.org/wiki/Turbocharger*>, accessed at 30.09.2010.

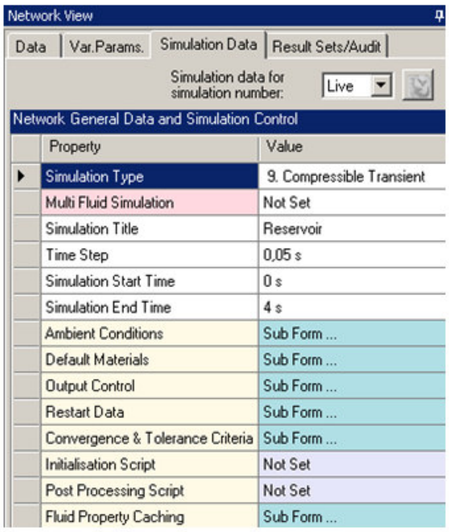
[13] **Benno, G. and Joachim, K**., (2003). The Basics of Vacuum Technology.

[14] **Berkien, F. and Goossens, M.,** (2010). Actuator and process evaluation of waste gate turbochargers. *Mitsubishi Turbochargers*.

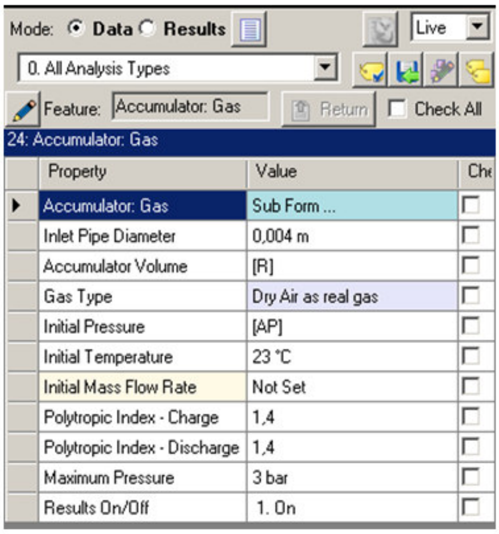
APPENDICES

APPENDIX A.1 : Flowmaster model, scripts and inputs

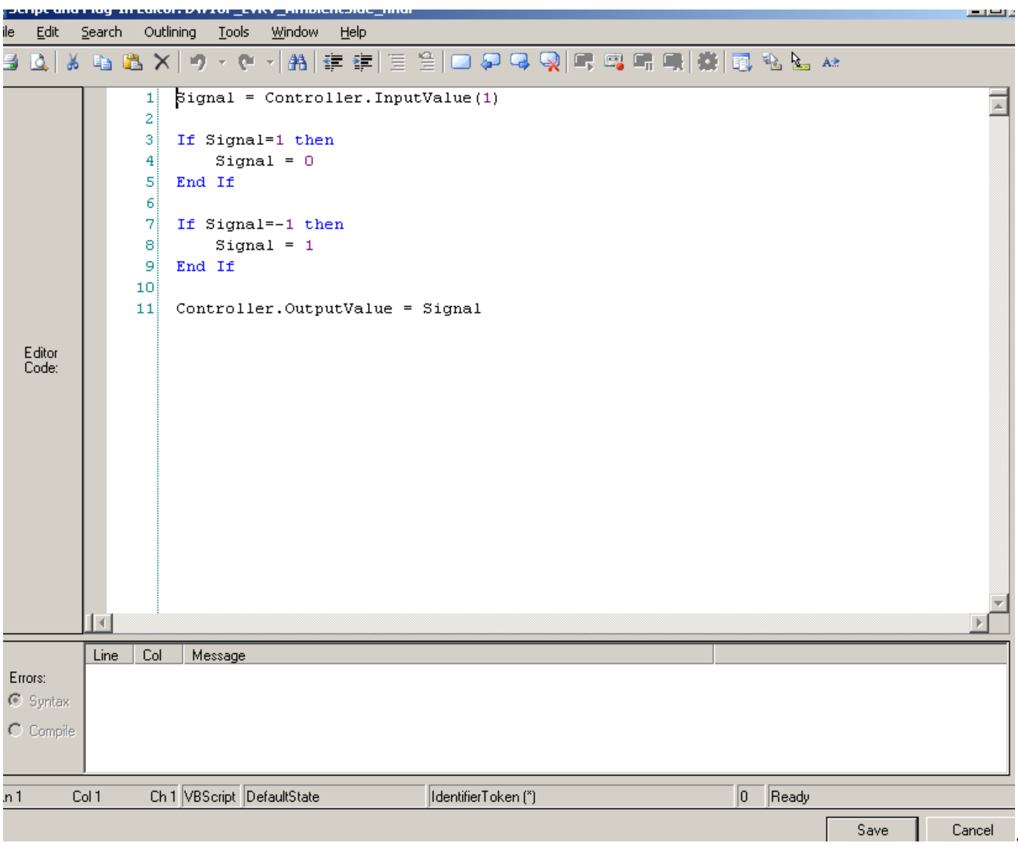
APPENDIX A.1



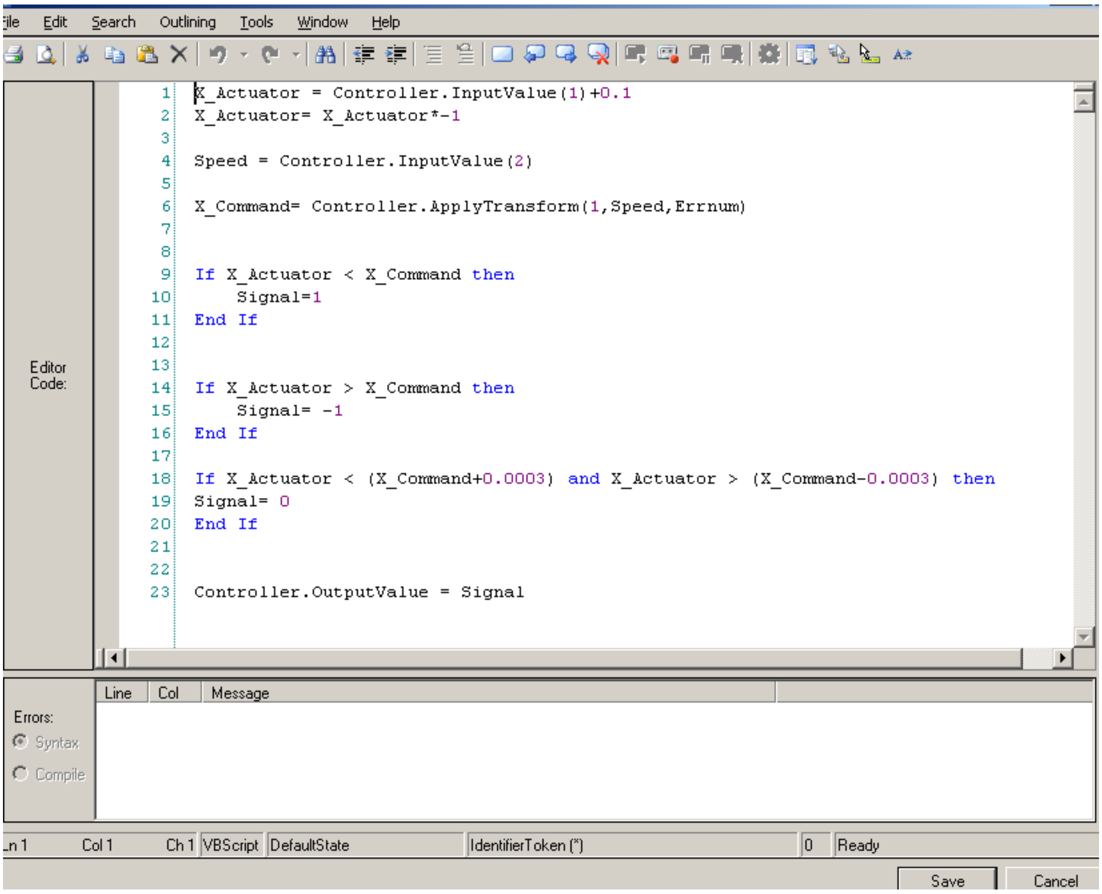
Network View
Data
Var.Params.
Simulation Data
Result Sets/Audit
Simulation data for simulation number:
Live
Network General Data and Simulation Control
Property
Value
Simulation Type
9. Compressible Transient
Multi Fluid Simulation
Not Set
Simulation Title
Reservoir
Time Step
0,05 s
Simulation Start Time
0 s
Simulation End Time
4 s
Ambient Conditions
Sub Form ...
Default Materials
Sub Form ...
Output Control
Sub Form ...
Restart Data
Sub Form ...
Convergence & Tolerance Criteria
Sub Form ...
Initialisation Script
Not Set
Post Processing Script
Not Set
Fluid Property Caching
Sub Form ...

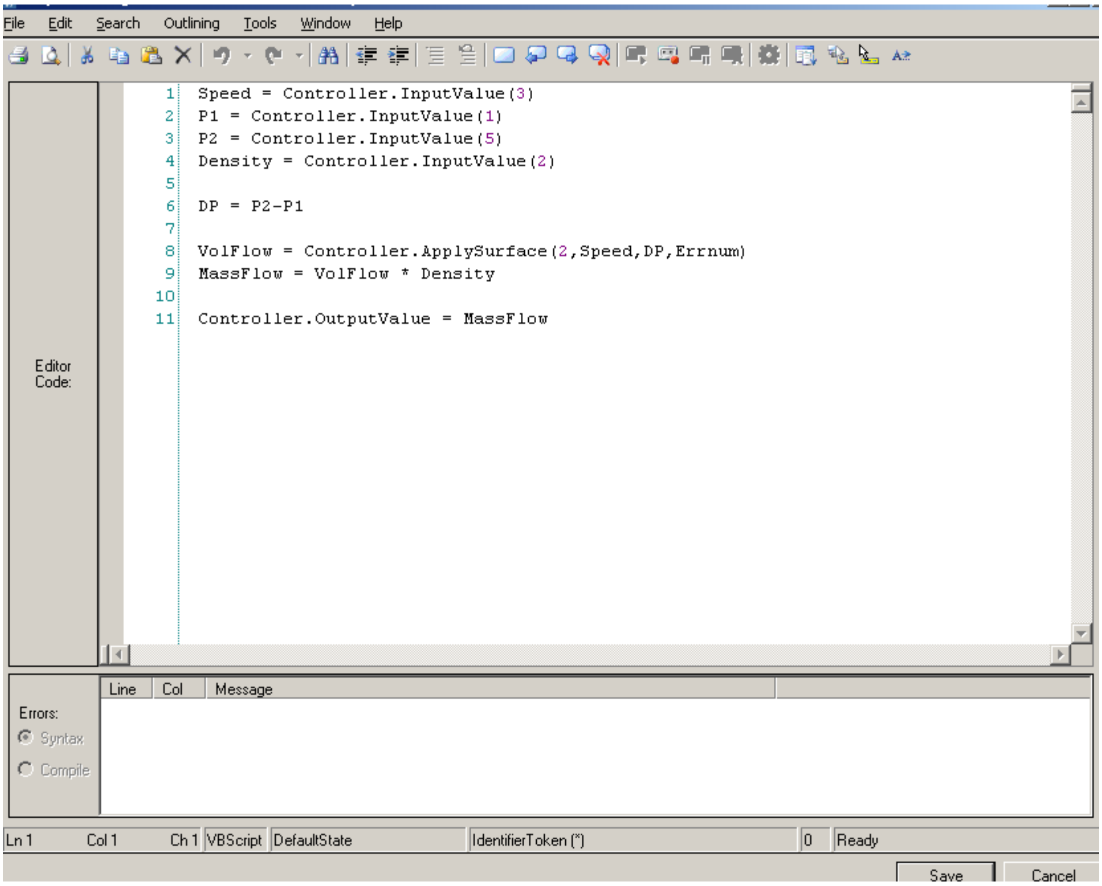


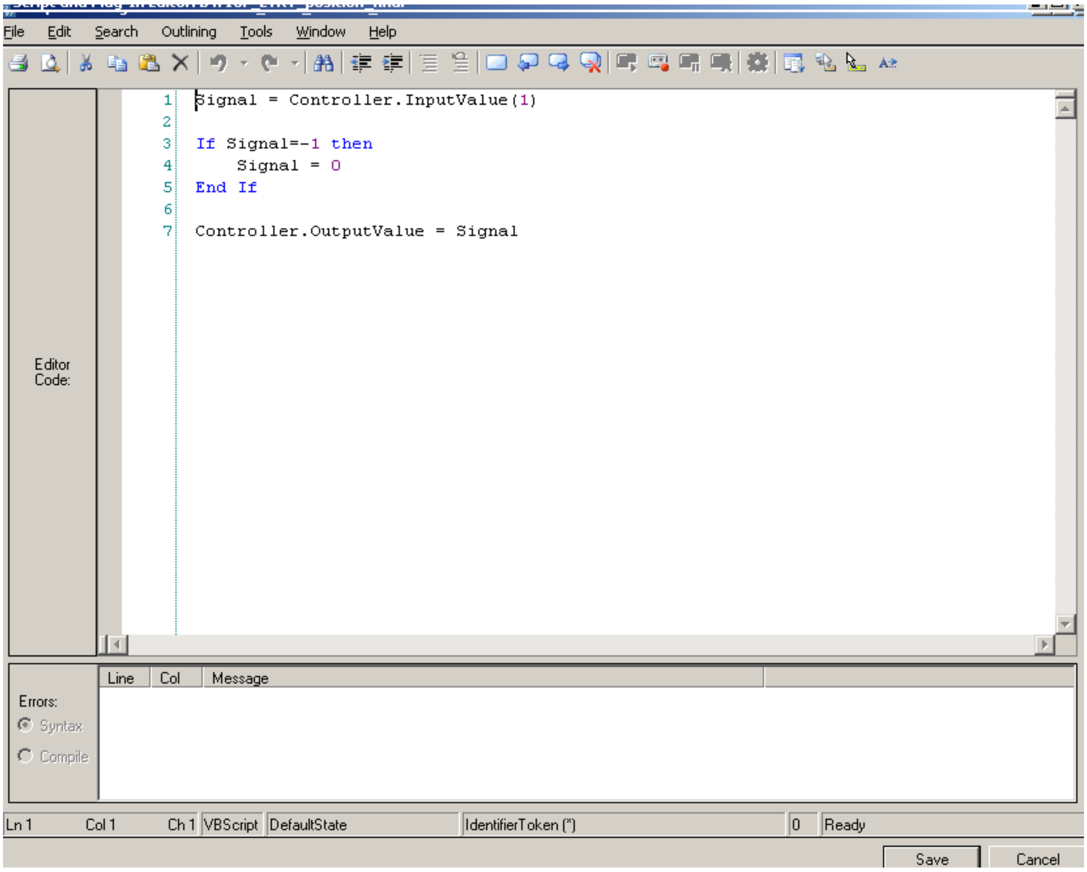
Mode:
Data
Results
Live
0. All Analysis Types
Feature:
Accumulator: Gas
Return
Check All
24: Accumulator: Gas
Property
Value
Accumulator: Gas
Sub Form ...
Inlet Pipe Diameter
0,004 m
Accumulator Volume
[R]
Gas Type
Dry Air as real gas
Initial Pressure
[AP]
Initial Temperature
23 °C
Initial Mass Flow Rate
Not Set
Polytropic Index - Charge
1,4
Polytropic Index - Discharge
1,4
Maximum Pressure
3 bar
Results On/Off
1. On

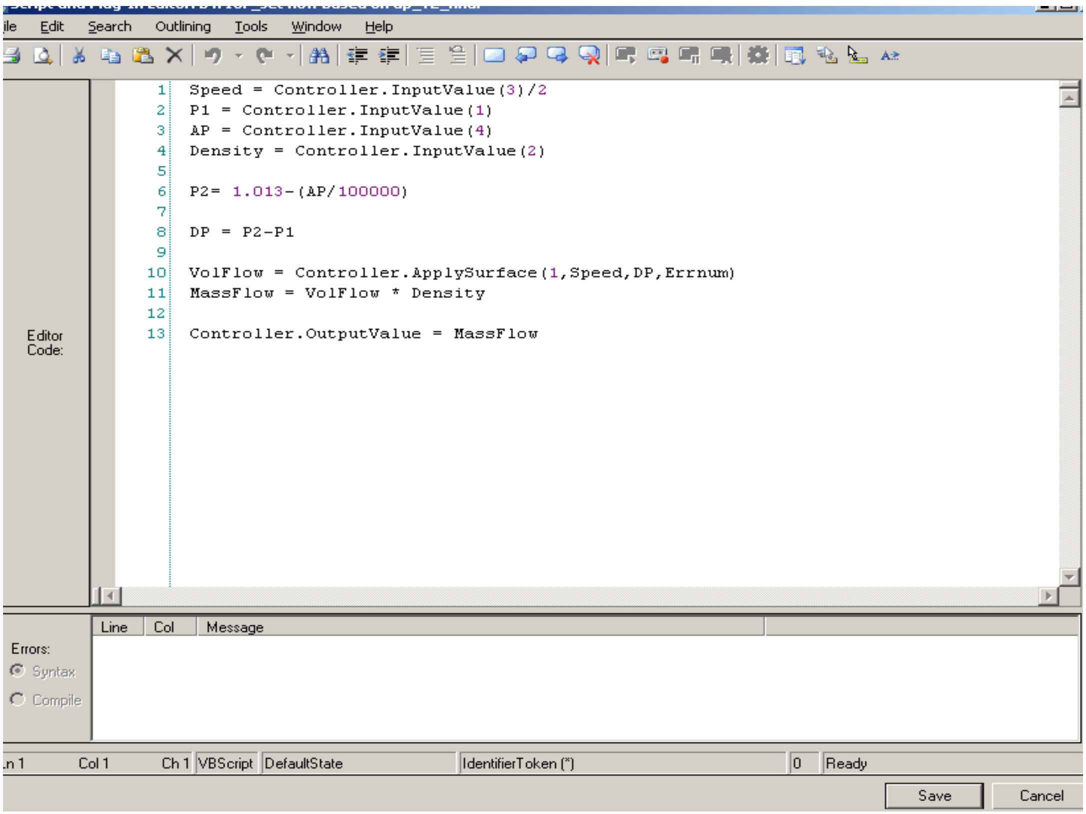


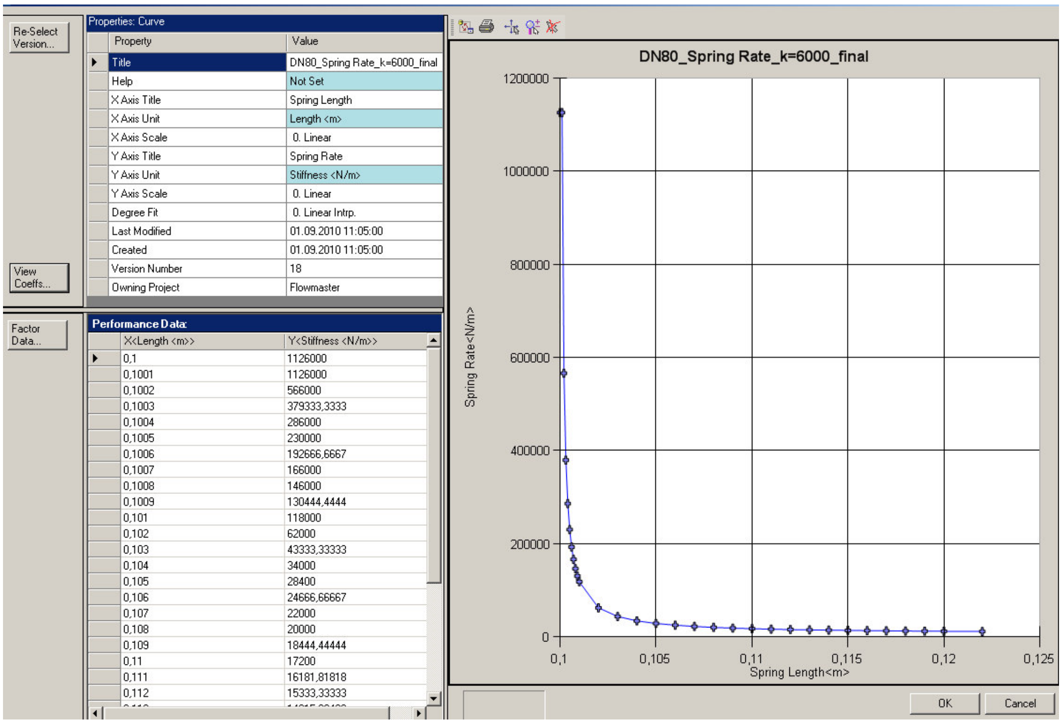
Edit
Search
Outlining
Tools
Window
Help
Signal = Controller.InputValue(1)
If Signal=1 then
Signal = 0
End If
If Signal=-1 then
Signal = 1
End If
Controller.OutputValue = Signal
Editor Code:
Line
Col
Message
Errors:
Syntax
Compile
Col 1
Ch 1
VBScript
DefaultState
IdentifierToken (")
0
Ready
Save
Cancel

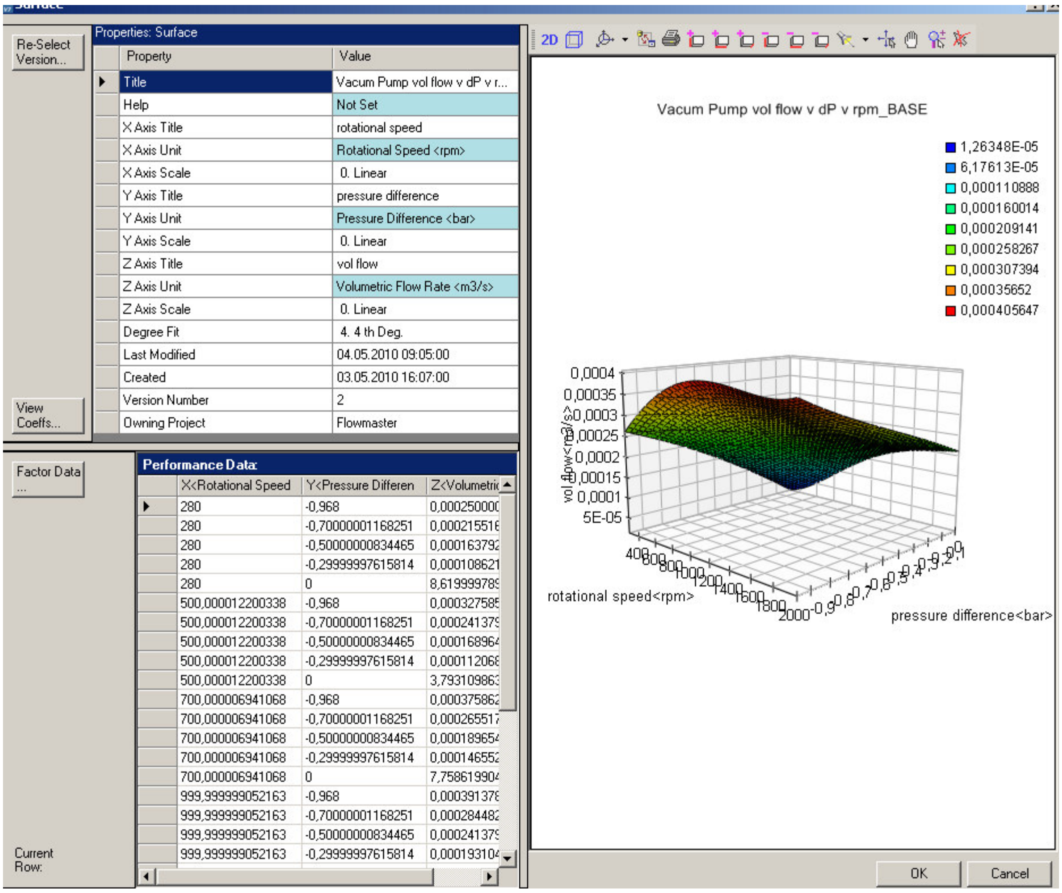

File Edit Search Outlining Tools Window Help
X_Actuator = Controller.InputValue(1)+0.1
X_Actuator= X_Actuator*-1
Speed = Controller.InputValue(2)
X_Command= Controller.ApplyTransform(1,Speed,Errnum)
If X_Actuator < X_Command then
Signal=1
End If
If X_Actuator > X_Command then
Signal= -1
End If
If X_Actuator < (X_Command+0.0003) and X_Actuator > (X_Command-0.0003) then
Signal= 0
End If
Controller.OutputValue = Signal
Editor Code:
Line Col Message
Errors:
Syntax
Compile
Ln 1 Col 1 Ch 1 VBScript DefaultState IdentifierToken (*) 0 Ready
Save
Cancel


File Edit Search Outlining Tools Window Help
Speed = Controller.InputValue(3)
P1 = Controller.InputValue(1)
P2 = Controller.InputValue(5)
Density = Controller.InputValue(2)
DP = P2-P1
VolFlow = Controller.ApplySurface(2,Speed,DP,Errnum)
MassFlow = VolFlow * Density
Controller.OutputValue = MassFlow
Editor Code:
Line Col Message
Errors:
Syntax
Compile
Ln 1 Col 1 Ch 1 VBScript DefaultState IdentifierToken (*) 0 Ready
Save
Cancel


File Edit Search Outlining Tools Window Help
Editor Code:
Signal = Controller.InputValue(1)
If Signal=-1 then
Signal = 0
End If
Controller.OutputValue = Signal
Line Col Message
Errors:
Syntax
Compile
Ln 1 Col 1 Ch 1 VBScript DefaultState IdentifierToken (*) 0 Ready
Save
Cancel


ile Edit Search Outlining Tools Window Help
Editor Code:
Speed = Controller.InputValue(3)/2
P1 = Controller.InputValue(1)
AP = Controller.InputValue(4)
Density = Controller.InputValue(2)
P2= 1.013-(AP/100000)
DP = P2-P1
VolFlow = Controller.ApplySurface(1,Speed,DP,Errnum)
MassFlow = VolFlow * Density
Controller.OutputValue = MassFlow
Line Col Message
Errors:
Syntax
Compile
.n 1 Col 1 Ch 1 VBScript DefaultState IdentifierToken (*) 0 Ready
Save
Cancel


Re-Select Version...
View Coeffs...
Factor Data...
Properties: Curve
Property
Value
Title
DN80_Spring Rate_k=6000_final
Help
Not Set
X Axis Title
Spring Length
X Axis Unit
Length <m>
X Axis Scale
0. Linear
Y Axis Title
Spring Rate
Y Axis Unit
Stiffness <N/m>
Y Axis Scale
0. Linear
Degree Fit
0. Linear Intrp.
Last Modified
01.09.2010 11:05:00
Created
01.09.2010 11:05:00
Version Number
18
Owning Project
Flowmaster
Performance Data
X<Length <m>>
Y<Stiffness <N/m>>
0,1 1126000
0,1001 1126000
0,1002 566000
0,1003 379333,3333
0,1004 286000
0,1005 230000
0,1006 192666,6667
0,1007 166000
0,1008 146000
0,1009 130444,4444
0,101 118000
0,102 62000
0,103 43333,33333
0,104 34000
0,105 28400
0,106 24666,66667
0,107 22000
0,108 20000
0,109 18444,44444
0,11 17200
0,111 16181,81818
0,112 15333,33333
DN80_Spring Rate_k=6000_final
Spring Rate<N/m>
1200000
1000000
800000
600000
400000
200000
0
0,1
0,105
0,11
0,115
0,12
0,125
Spring Length<m>
OK
Cancel


Re-Select Version...
View Coeffs...
Factor Data ...
Current Row:
Properties: Surface
Property
Value
Title
Vacum Pump vol flow v dP v r...
Help
Not Set
X Axis Title
rotational speed
X Axis Unit
Rotational Speed <rpm>
X Axis Scale
0. Linear
Y Axis Title
pressure difference
Y Axis Unit
Pressure Difference <bar>
Y Axis Scale
0. Linear
Z Axis Title
vol flow
Z Axis Unit
Volumetric Flow Rate <m3/s>
Z Axis Scale
0. Linear
Degree Fit
4. 4 th Deg.
Last Modified
04.05.2010 09:05:00
Created
03.05.2010 16:07:00
Version Number
2
Owning Project
Flowmaster
Performance Data
X<Rotational Speed
Y<Pressure Differen
Z<Volumetric
280 -0,968 0,000250000
280 -0,70000001168251 0,000215516
280 -0,50000000834465 0,000163792
280 -0,29999997615814 0,000108621
280 0 8,619999789
500,000012200338 -0,968 0,000327585
500,000012200338 -0,70000001168251 0,000241379
500,000012200338 -0,50000000834465 0,000168964
500,000012200338 -0,29999997615814 0,000112068
500,000012200338 0 3,793109863
700,000006941068 -0,968 0,000375862
700,000006941068 -0,70000001168251 0,000265517
700,000006941068 -0,50000000834465 0,000189654
700,000006941068 -0,29999997615814 0,000146552
700,000006941068 0 7,758619904
999,999999052163 -0,968 0,000391378
999,999999052163 -0,70000001168251 0,000284482
999,999999052163 -0,50000000834465 0,000241379
999,999999052163 -0,29999997615814 0,000193104
Vacum Pump vol flow v dP v rpm_BASE
1,26348E-05
6,17613E-05
0,000110888
0,000160014
0,000209141
0,000258267
0,000307394
0,00035652
0,000405647
0,0004
0,00035
0,0003
0,00025
0,0002
0,00015
0,0001
5E-05
vol flow<m3/s>
rotational speed<rpm>
pressure difference<bar>
OK
Cancel

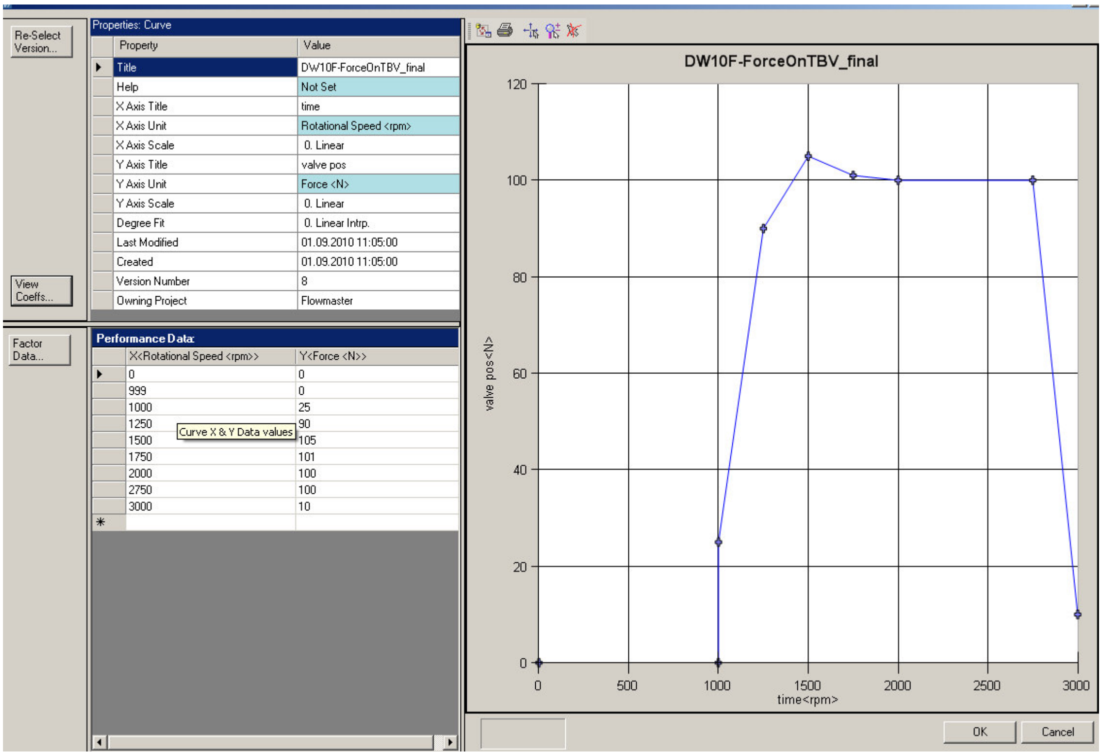


Figure A.1 : Scripts and inputs in flowmaster

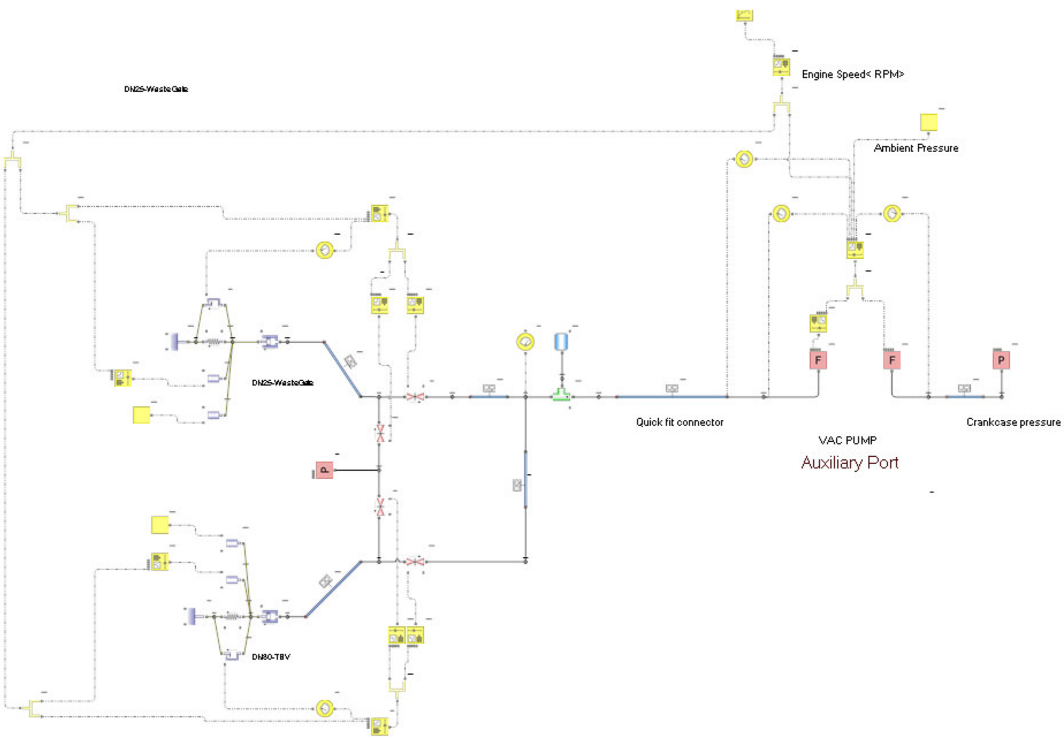


Figure A.2 : Vacuum system 1D model in flowmaster

CURRICULUM VITAE

Candidate's full name: Ömer Ceyhun

Place and date of birth: Ankara-20.04.1986

Permanent Address: Bağlarbaşı Mahallesi Sülün Sokak No:12/7

Maltepe-İstanbul

Universities and
Colleges attended: İstanbul Technical University, M.Sc. (2011)

İstanbul Technical University, B.Sc. (2008)

Ankara Polatlı Anatolian High School (2003)