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TOPIC

Evaluation of the energy performance of the centrifugal compressor of a pumping station

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Dedication

We dedicate this thesis to the countless individuals who have played a significant role in my academic journey, inspiring and supporting me every step of the way.

To our family OUZAID and REGUIGUESSAG, whose unwavering love, encouragement, and sacrifices have made it possible for me to pursue my education. Your belief in me has been my driving force, and I am forever grateful for your constant support.

To our advisor: Mr.BOUKSANI، professors, and mentors, thank you for your guidance, expertise, and patience. Your insightful feedback and constructive criticism have shaped me into a better scholar. Your commitment to my growth and success has been invaluable.

To the one who help us on providing the data we needed on this case of study KARBOUCH HASAN with a wormed heart we are thankful for all of his efforts that given on this case of study.

To the participants and respondents who generously shared their time and insights for this research, thank you for being the foundation of this study. Your contributions have been instrumental in shaping the findings and conclusions.

To the academic community, whose extensive research and scholarship have paved the way for our inquiry. Your work has guided me, propelling my curiosity and deepening my understanding.

Lastly, I dedicate this thesis to us as a testament to our perseverance, growth, and commitment to lifelong learning. This achievement represents years of hard work, dedication, and sacrifice. It serves as a reminder that dreams can become a reality with determination and passion.

May this thesis contribute to the existing body of knowledge and inspire future researchers and scholars to delve deeper into the subject matter.

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Pr : discharge pressure

Pa : suction pressure

Z: Compressibility coefficient

Qm: Mass flow

Ta: Suction temperature

Tr: Discharge temperature

Cp: Specific heat

n: Rotation speed

R: gas constant

Tcr: Critical mixture temperature

Pcr: Critical mixture pressure

K: adiabatic coefficients

Ɛ: Compression ratio

N: Polytropic coefficient

Wad: Adiabatic work

Wp: polytropic work

Wr: Actual work

η_{ad}: Adiabatic efficiency

η_{p}:Polytropic efficiency

ηg: Overall performance

Ti: temperature in stage i

Pu: the power transmitted to the gas

Na: the power absorbed

Abstract

In this study, we performed a thermodynamic verification to verify the performance of the k101 centrifugal compressor. As a result we obtained that the gap with the control room is acceptable . We verify the evolution of the effect of the degree of cooling between the two stages (Td1 and Ts2) on the performance K101A centrifugal compressor .so that the increase in cooling produces the increase of the polytropic efficiency (%) and the decrease of actual work(kJ/kg), and the absorbed power (w).

Résumé

Dans cette étude, nous avons effectué une étude thermodynamique pour vérifier les performances du compresseur centrifuge k101. En conséquence, nous avons obtenu que l'écart avec la salle de contrôle est acceptable, nous avons étudié l'évolution de l'effet du degré de refroidissement entre les deux étages sur les performances du compresseur centrifuge K101A.de sorte que plus le refroidissement est important, plus l'efficacité polytropique est grande et l'absence de travail réel et de puissance absorbée.

ملخص

في هذه الدراسة، أجرينا دراسة ديناميكية حرارية للتحقق من أداء ضاغط الطرد المركزي K101A وتوصلنا إلى أن نسبة الأخطاء صغيرة جدا مقارنة بالمصنع، ثم درسنا تطور تأثير درجة التبريد بين المرحلتين على أداء ضاغط الطرد المركزي K101A.حيث كلما زاد التبريد، زادت كفاءة الضاغط ونقص العمل الحقيقي والطاقة الممتصة.

General introduction

Faced with oil; gas, its competitor in the fuel markets appeared as a form of energy difficult to implement, especially due to the weight of the investments and the transport costs to the consumer.

The compression section is of paramount importance in the petroleum industry.

A turbocharger is necessary for these installations and provides energy to gases. This energy allows fluid to flow in a pipe to rise to the highest-level pupil.

Compressors increase the energies of gases.

We should also add that each mechanic must be able to choose compressors according to the technological requirements, the main parameters, the maintenance rules, and the theoretical basis for predicting the machine's condition in different operating conditions.

The work brought in this dissertation is structured into four chapters:

- The first chapter is devoted to general information about compressors and centrifugal compressors.
- The second chapter is devoted to describing the centrifugal compressor type **K101**.
- The third chapter is devoted to Performance calculation obtained from the **K101 compressor**.
- The fourth chapter is Interpretation and Results.

In addition, the dissertation also includes an introduction, a general conclusion, and bibliographic references.

Chapter I

General information about compressors

I.1 Compressor:

I.1.1 The Definition:

A gas compressor is a mechanical device that increases the pressure of a gas by reducing its volume. Compressors are similar to pumps: both increase the pressure on a fluid, and both can transport the fluid through a pipe. As gases are compressible, the compressor also reduces the volume of a gas. Liquids are relatively incompressible some can be compressed, the main action of a pump is to pressurize and transport liquids**.**

I.1.2 The goal of using a compressor:

This operation for the goal of:

- ✓ Circulating a gas in a closed circuit.
- ✓ Increasing (or reducing) gas flow pressure levels required for processing,
- ✓ Providing pressure differences to overcome system resistances, thus enabling gas flows through reactors, heat exchangers, pipes, and
- ✓ Refrigerating gas flows for cooling and liquefaction.

Other functions of a compressor include

- ✓ Providing compressed gas or air for combustion.
- ✓ Transporting process gases through pipelines.
- ✓ Provide compressed air for driving pneumatic tools.
- ✓ Circulating process fluid through a certain process**.**

I.1.3 Basic types of compressors:

Compressors are available in various types, as listed below:

- ✓ Positive Displacement Compressors
- ✓ Reciprocating compressor

- ✓ Centrifugal compressors
- ✓ Pipeline compressors

The following diagram shows a chart for basic compressor types.

These types are further characterized by the following:

Compression stages:

Air compressors are available in single or multiple stages of compression.

Multiple staging is usually adopted when the discharge pressure exceeds 80 psi.

Multiple-stage compressors offer energy efficiency as air is cooled between Compression stages and moisture is removed through cooling. The advantages of Multistage compression are:

• Better mechanical balance and uniform torque of multi-crank machines and a

Smaller flywheel;

• Increased volumetric efficiency as a result of lower pressure in the IP cylinder

clearances;

- ❖ Reduction of power to the drive;
- ❖ Possibility of operating at high speeds;
- ❖ Provisions for better lubrication due to the lower working temperature;
- ❖ Smaller leakage loss and lighter cylinders.

Type of Cooling:

Air compressors can be air, water, cooled-or oil.

Air-cooled compressors have either integrally mounted or separate oil or air coolers.

These require adequate ventilation to perform reliably. Water-cooled compressors require an adequate pressure of quality water. Remember that air compressors typically.

Reject about 2,000 - 2,500 Btu/Hr. for all hp. Water-cooled units are more energy Efficient**.**

Type of Drive:

The prime mover can be an electric motor, engine, steam, or turbine driven.

Electrically motor driven is the most common type found in industrial operations.

Engine or turbine-driven machines are sometimes used for heavy-duty applications and

In mobile units in remote locations where an electrical supply is not available. Such an

An example would be in a mining operation**.**

Lubrication (oil, oil-free):

Air compressors are available as dry/oil-free and lubricated. The types of compressors

Are as follows:

Table I.1: Types of compressors according to lubrication (oil, oil-free)

Non-Lubricated (Oil Free)	Lubricated
Rotary Screw & lobe	Rotary screw & lobe
Reciprocating	Reciprocating
Centrifugal	Non

free compressors are generally-Oil preferable in clean air applications such as food.

Industry, electronic manufacturing, controls and instrumentation, and hospital services.

Lubricated compressors for utility air services are acceptable if proper coalescingis done .

The filtration system is included**.**

Packaged or custom-built:

• Usually compressors are packaged items; units are custom built when heavy-duty

Machines for the mining industry or large energy projects are required.

• Most reciprocating and dry screw air compressors are shipped as self-contained

Packaged items requiring minimum tie-in connections.

• The centrifugal air compressor package system is typically not shipped as -selfcontained.

Operating pressures:

Pressure is the main parameter of compressed air which is usually expressed in the units

Psi or bar (psi = pounds/sq-inch); (1 bar = 10^5 Pa = 10^5 N/m^2= 14.504 psi).

• Low Pressure (0 to 150 psi)

• Medium Pressure (151 to 1000 psi)

• High Pressure (over 1000 psi).

I.1.4 The types of compressors :

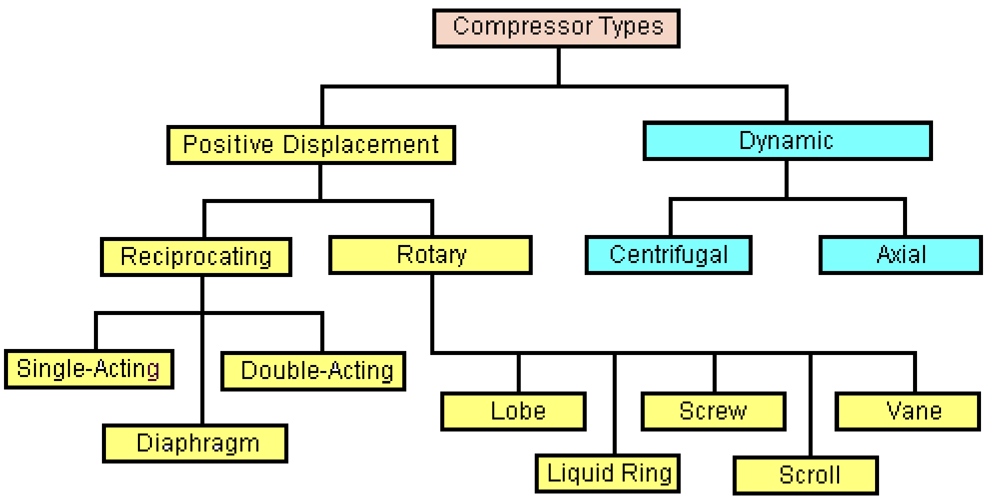

Figure I.1: The types of compressors [3]

➢ Diagonal or mixed-flow compressors:

Diagonal or mixed-flow compressors are similar to centrifugal compressors but have a radial and axial velocity component at the exit from the rotor. The diffuser is often used to turn diagonal flow to an axial rather than radial direction.

➢ Axial-flow compressors:

Axial-flow compressors are dynamic rotating compressors that use arrays of fan-like airfoils to compress the working fluid progressively. They are used where there is a requirement for a high flow rate or a compact design. The arrays of airfoils are set in rows, usually as pairs: one rotating and one stationary. The rotating airfoils, or blades or rotors, accelerate the fluid. The stationary airfoils, also known as stators or vanes, decelerate and redirect the flow direction of the fluid, preparing it for the rotor blades of the next stage. Axial compressors are almost always multi-staged, with the cross-sectional area of the gas passage diminishing along the compressor to maintain an optimum axial Mach number. Beyond about five stages or a 4:1 design pressure ratio, variable geometry is normally used to improve operation. Axial compressors can have high efficiencies, around 90% polytrophic at their design conditions. However, they are relatively expensive, requiring many components, tight tolerances, and quality-high materials. Axial-flow compressors can be found in medium to large gas turbine engines, in natural gas pumping stations, and within certain chemical plants.

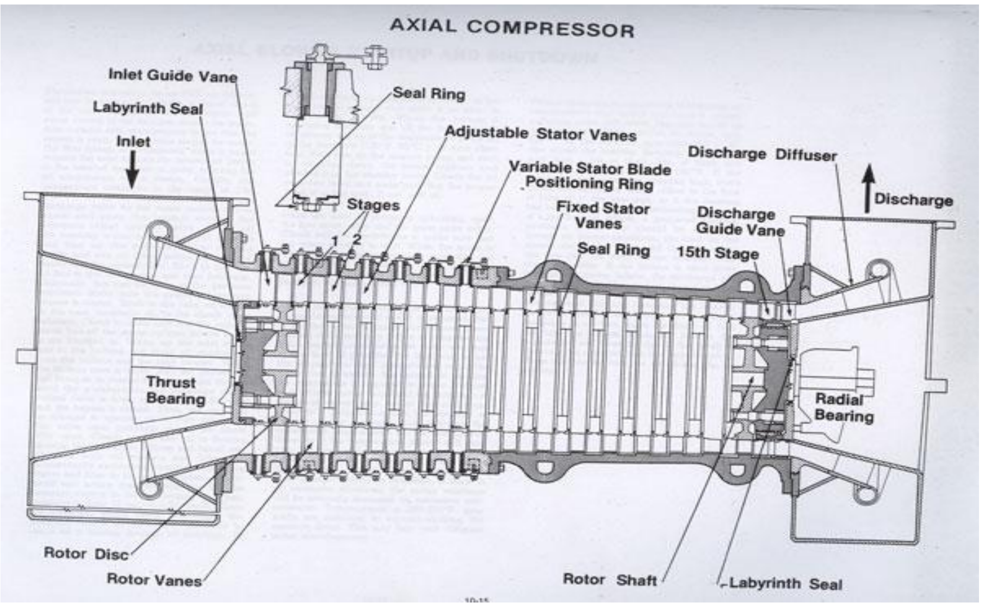

Figure I.2: Axial-flow compressor[3]

➢ Centrifugal compressor:

• The centrifugal air compressor is a dynamic compressor that uses high-speed rotating impellers to accelerate air. They are suitable for high gas volume applications in the chemical process industry, steel plants, oil refineries, and gas transmission systems. Important characteristics are listed below:

• Centrifugal compressors produce a high-pressure discharge by converting angular momentum imparted by a rotating impeller (dynamic displacement). Centrifugal compressors rotate at higher speeds than the other compressors to do this efficiently.

• These compressors are designed for higher capacity because airflow through the compressor is continuous.

• Capacity control is achieved by variable speed drives or by adjusting the inlet guide vanes. By closing the guide vanes, volumetric flows, and capacity are reduced. Centrifugal air compressors have a limited capacity control range, and below 70% percent capacity, the excess air can be blown off into the atmosphere.

• Centrifugal compressors run at relatively high speeds, and noise control is usually required.

• The centrifugal air compressor is an oil-free compressor by design. Shaft seals and atmospheric vents separate the lubricated-oil running gear from the air.

• To reach operating pressures, several impeller stages are required. They have low installation costs but are expensive because they are precision machines. They are pretty efficient، down to approximately 60% of their design output.

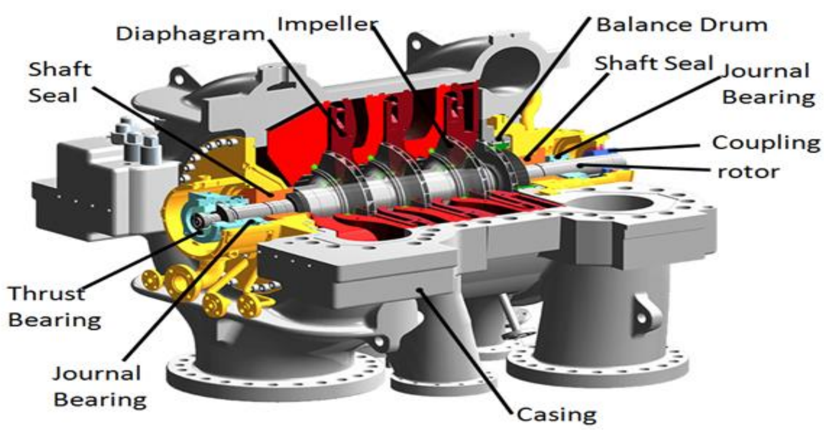

Figure I.3: Centrifugal compressor

➢ Reciprocating compressors:

Reciprocating compressors use pistons driven by a crankshaft. They can be either stationary or portable, single or multi-staged, and driven by electric motors or internal combustion engines. Small reciprocating compressors from 5 to 30 horsepower (hp) are commonly seen in automotive applications and are typically for intermittent duty. Larger reciprocating compressors well over 1000 hp (750 kW) are commonly found in large industrial and petroleum applications. Discharge pressures range from low to high (>18000 psi or 180 MPa). In certain applications, such as air compression, multi-stage double-acting compressors are said to be the most efficient and are typically larger and more costly than comparable rotary units.[6] Another type of reciprocating compressor is the swash plate compressor, which uses pistons that are moved by a swash plate mounted on a shaft - see Axial Piston Pump. Household, home workshop، and smaller job site compressors typically reciprocate 1½ hp. fewer compressors with an attached receiver tank.

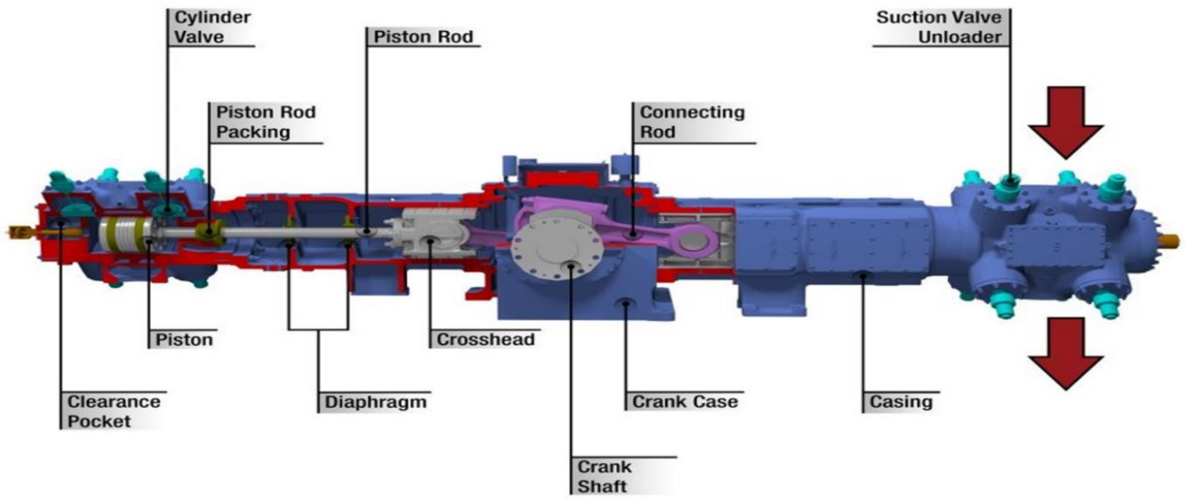

Figure I.4: Reciprocating compressors

➢ Rotary screw compressors:

• Rotary Screw air compressors are also positive displacement type units. However, instead of a piston, the rotary screw compressor increases pressure through the action of two intermeshing rotors within twin-bore housing. Air is trapped between the rotors and housing. As it moves along the rotor, air volume decreases, and pressure increases because the allowable space for the trapped air is progressively reduced. The single-stage helical or spiral lobe oil-flooded screw air compressor is the most common rotary air compressor. Important characteristics of rotary screw compressors are highlighted below:

• There are no valves. These units are oil-cooled; the oil seals the internal clearances. The oil coolers, in turn, are either air or water-cooled.

• Screw compressors are available in both oil-injected and oil-free types. The oil-free rotary screw air compressor utilizes specially designed air ends to compress air, without oil, in the compression chamber. This yields true oil-free air.

• Since the cooling occurs immediately within the compressor, the moving parts never experience extreme operating temperatures.

• Capacity control for these compressors is accomplished by variable speed or variable displacement. A slide valve is positioned in the casing for the latter control technique. The slide valve opens as the compressor capacity is reduced, directing some compressed air back to the suction. Advantages of the rotary screw compressor include smooth, pulse-free air output, in a compact size, with high output volume, over a long life.

• free-Oil rotary screw air compressors are available in two configurations: dry and water injected. They have the same benefits as the injected-oil screw compressors but compress in two stages and have no lubricant in contact with the air while passing through the compressor.

Figure I.5: Rotary screw compressor

Rotary vane compressors:

Rotary vane compressors consist of a rotor with several blades inserted in radial slots in the rotor. The rotor is mounted offset in a larger housing lar or more complexthat can be circu. As the rotor turns, blades slide in and out of the slots keeping contact with the outer wall of the housing. Thus, a series of decreasing volumes is created by the rotating blades. Rotary Vane compressors are, with piston compressors، one of the oldest compressor technologies. With suitable port connections, the devices may be either a compressor or a vacuum pump. They can be either stationary or portable, staged, and-single or multi driven by electric motors or internal combustion engines. Dry vane machines are used at relatively low pressures (e.g., 2 bar or 200 kPa; 29 psi) for bulk material movement، while oil-injected machines have the necessary volumetric efficiency to achieve pressures up to about 13 bar (1300 kPa; 190 psi) in a single stage. A rotary vane compressor is well suited to electric motor drive and is significantly quieter than the equivalent piston compressor. Rotary vane compressors can have mechanical efficiencies of about 90%.

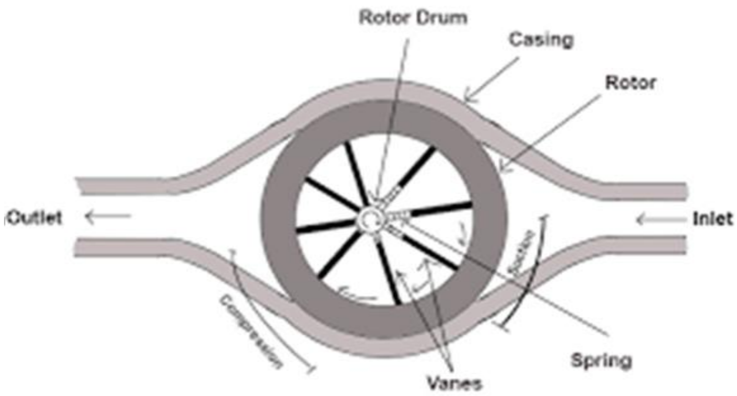

Figure I.6: Rotary vane compressor

➢ Scroll compressors:

A scroll compressor, scroll pump, and scroll vacum pump use two interleaved spiral-like vanes to pump or compress fluids such as liquids and gases. The vane geometry may be involute, archimedean spiral, or hybrid curves. They operate more smoothly, quietly, and reliably than other compressors in the lower volume range .One of the scrolls is often fixed, while the other orbits eccentrically without rotating, trapping and pumping، or compressing pockets of fluid or gas between the scrolls. This type of compressor was used as the supercharger on Volkswagen G60 and G40 engines in the early 1990s.

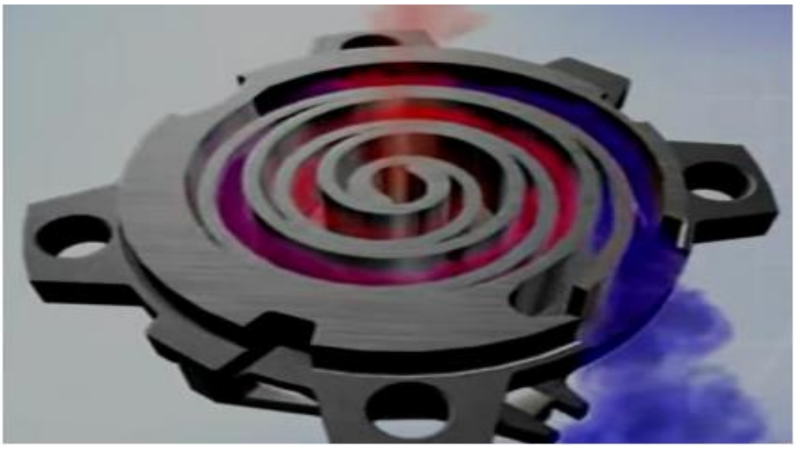

Figure I.7: Scroll compressors

- ## Diaphragm compressors :

A diaphragm compressor (also known as a membrane compressor) is a variant of the conventional reciprocating compressor. Gas compression occurs by moving a flexible membrane instead of an intake element. The forth-and-back movement of the membrane is driven by a rod and a crankshaft mechanism. Only the membrane and the compressor box come in contact with the compressed gas. Diaphragm compressors are used for hydrogen and compressed natural gas (CNG) and in several other applications.

The photograph included in this section depicts a three-stage diaphragm compressor used to compress hydrogen gas to 6000 psi (41 MPa) for use in a prototype compressed hydrogen and compressed natural gas (CNG) fueling station built in downtown Phoenix, Arizona، by the Arizona Public Service Company (an electric utilities company). Reciprocating compressors were used to compress the natural gas. The prototype alternative fueling station was built in compliance with all of the prevailing safety, environmental and building codes in Phoenix to demonstrate that such fueling stations could be built in urban area.

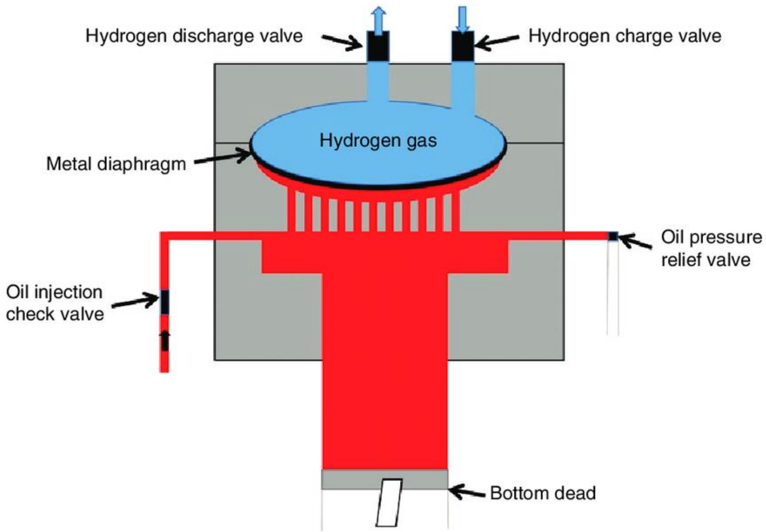

Figure I.8: Diaphragm compressors

The air bubble compressor، a mixture of air and water generated through turbulence، can fall into a subterranean chamber where the air separates from the water. The weight of falling water compresses the air at the top of the chamber. A submerged outlet from the chamber allows water to flow to the surface at a lower height than the intake. An outlet in the chamber's roof supplies the compressed air to the surface. A facility on this principle was built on the Montreal River at Ragged Shutes near Cobalt, Ontario، in 1910 and supplied 5,000 horsepower to nearby mines[1].

I.2 Centrifugal Compressor:

I.2.1 The Definition of Centrifugal Compressor:

Centrifugal compressors, sometimes called radial compressors, are a sub-class of dynamic axisymmetric work-absorbing turbomachinery.

They achieve pressure rise by adding energy to the continuous flow of fluid through the rotor/impeller. A substantial portion of this energy is kinetic, converted to increased potential energy/static pressure by slowing the flow through a diffuser. The static pressure rise in the impeller may roughly equal the rise in the diffuse[2].

I.2.2 Why Centrifugal?

- There are various benefits of being centrifugal like
- It is a mature technology
- Suitable for large capacities
- Power Range from 0.4 to 40 MW
- Small footprint
- High Availability (99%)
- Less Maintenance.

I.2.3 The components centrifugal compressor:

A centrifugal compressor comprises four main components – impeller, inlet, collector diffuser.

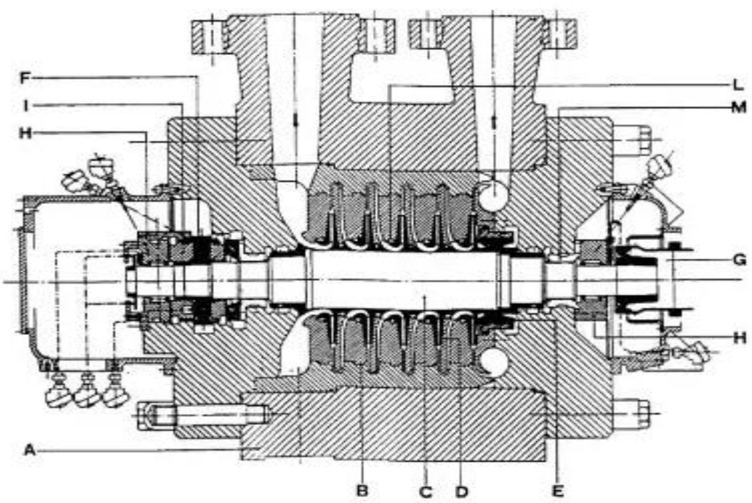

Figure I.9: The components centrifugal compressor

A: Outer casing

B: Stator parts called 'Diaphragm bundle.'

C: Rotor

D: Impellers

E: Balance drum

F: Thrust collar

G: Hub

H: Journal Bearing

I: Thrust Bearing

J: Labyrinth Seals

K: Oil film end seal

I.2.3.1 Impellers:

The impellers are mounted on a steel shaft inside the compressor، and this assembly is known as the compressor rotor. The function of the rotor is to provide velocity to the gas through blades attached to a rotating disc. Depending upon the desired output, these blades can be forward, radial, or backward-leaning. Since backward-leaning blades provide the widest range of efficiency, most multistage compressors use them.

Figure I.10: Impellers

I.2.3.2 Casing and Inlet:

A casing or housing generally protects or guards all the compressor components. A case consists of multiple numbers of bearings that provide radial and axial support to the rotor. It also contains multiple nozzles and all the inlets and discharge flow connections to introduce and extract flow from the compressor. This protective case is generally manufactured out of cast iron or steel. The casing is of two types:

- Horizontally split
- Vertically split

I.2.3.3 Collector:

The gas is collected and delivered to the discharge flange at the last impeller stage. A collector is employed to gather the gas discharged through the diffuser. The collector also contains certain valves and other instrumentation which serve the purpose of controlling the compressor.

Figure I.11: Collector

I.2.3.4 Diffuser:

The impeller removes the gas at high velocity into a diffuser passage, asmall space between adjacent diaphragms that normally change the gas flow 180° to guide it toward the next impeller. The diffuser usually comprises two walls that create a radial channel. Due to this arrangement, the velocity of the gas decreases، and its dynamic pressure is converted into static pressurethe— diffuser passages.

Figure I.12: Diffuser

I.2.4 Working of Centrifugal Compressor:

- After the start-up, gas or air is introduced from the air tank or any other source into the centrifugal compressor.
- After entering the compressor, the air strikes the impeller, which contains multiple radial blades rotating with the rotation of the impeller.
- As the air strikes the impeller blades, the air pushes to the center of the impeller by centrifugal force.
- The impeller blades, after striking air, provide kinetic energy to the air, increasing the air's velocity.
- The air enters the diffuser area after passing through the impeller. This diffuser contains multiple stationary vanes. As soon as the air enters the diffuser area, its speed or velocity of flow of the air starts to decrease.
- Now, according to Bernoulli’s principle, the velocity square is inversely proportional to pressure. The diffuser converts this increased air velocity into pressure energy before the air is drawn into the center of theimpeller. This increase in the impeller’s pressure under most conditions will roughly equal the increase in diffuser pressure.

I.2.5 The different types of centrifugal compressors :

We can distinguish 3 main groups following[5]:

I.2.5.1 Compressors with horizontally open bodies:

Horizontally open bodies, obviously consisting of half-bodies united on the plane

Horizontal joints are used for remaining operating pressures below 60 bars. If there are any, lubricating oil piping and all other Connections of the compressor with the rest of the installation are done normally with the lower half body. Thanks to this system، it is enough to remove the bolts of connection along the horizontal parting plane to lift the upper half of the body and easily access all internal components of the compressor، such as the rotor, diaphragms، and labyrinth joints.

Compressors with horizontally open bodies are indicated by the acronym MCL and Can be subdivided in turn according to the number of compression stages.

Figure I.13: Compressors with horizontally open bodies

➢ MCL compressors:

These are multi-stage compressors comprising a stage of Compression.

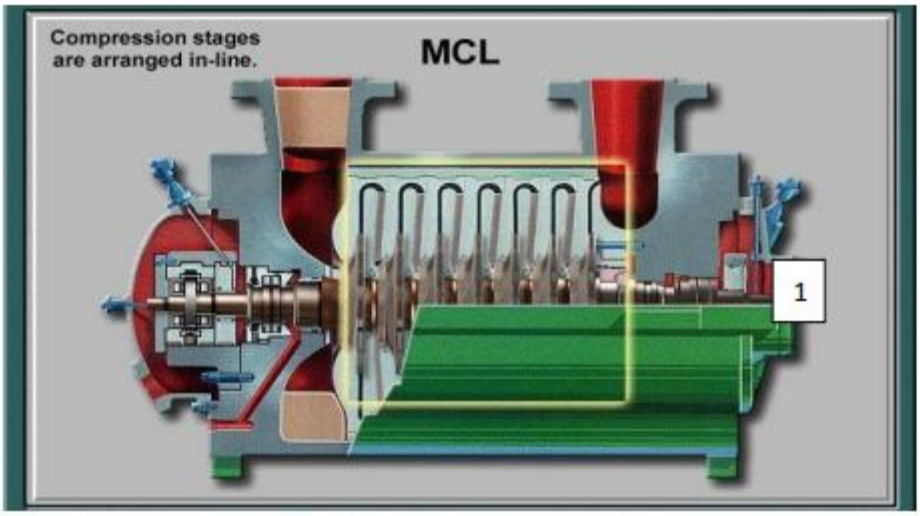

Figure I.14: MCL compressors

➢ 2MCL compressors:

These stage compressors group-multi two stages of Series compression with intermediate refrigeration in the same machine.

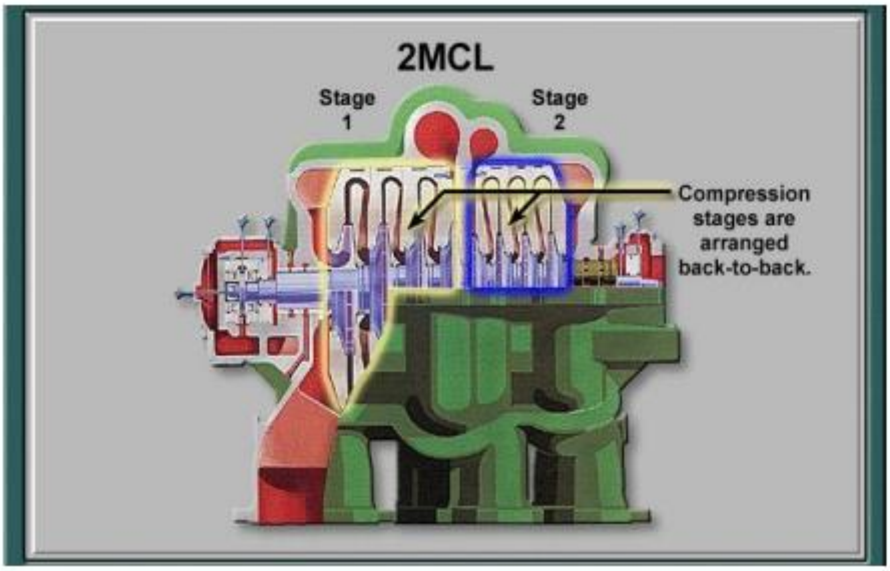

Figure I.15: 2MCL compressors

➢ 3MCL compressors:

These are multi-stage compressors، generally with more than two compression stages made

In a single body. In general, they are used for services where there is a need for

Compress different gas flow rates to different pressure levels, either with aninjection or with

Gas extraction during compression.

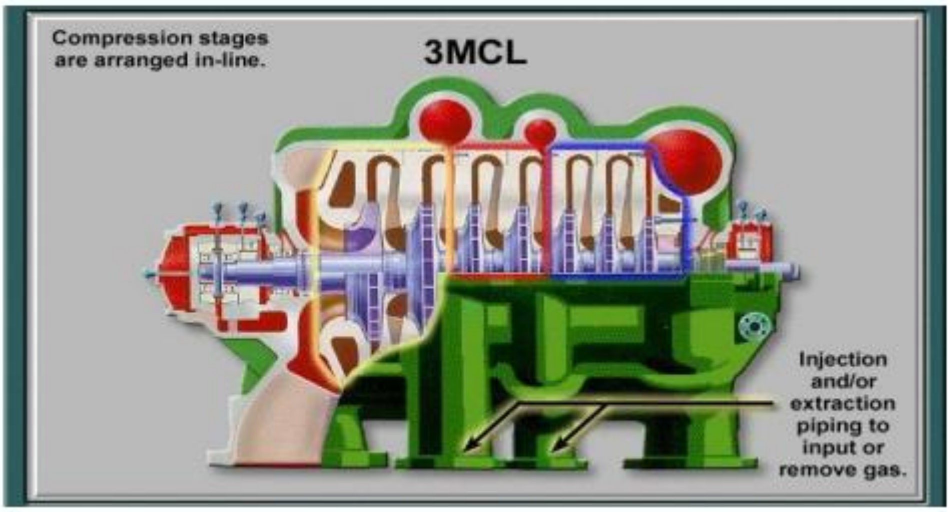

Figure I.16: 3MCL compressors

I.2.5.2 Compressor with vertically open bodies

The vertically open bodies consist of a cylinder closed at the ends by two flanges. It is for this reason that this type of compressor is called abarrel there. Compressors, usually multi-stage, can operate at high pressures (up to 700K kg/cm2) .Hp rotor and diaphragms inside the body do not differ from those of MCL compressors.

- **BCL Compressors:**

These are type-barrel compressors with a single compression stage.

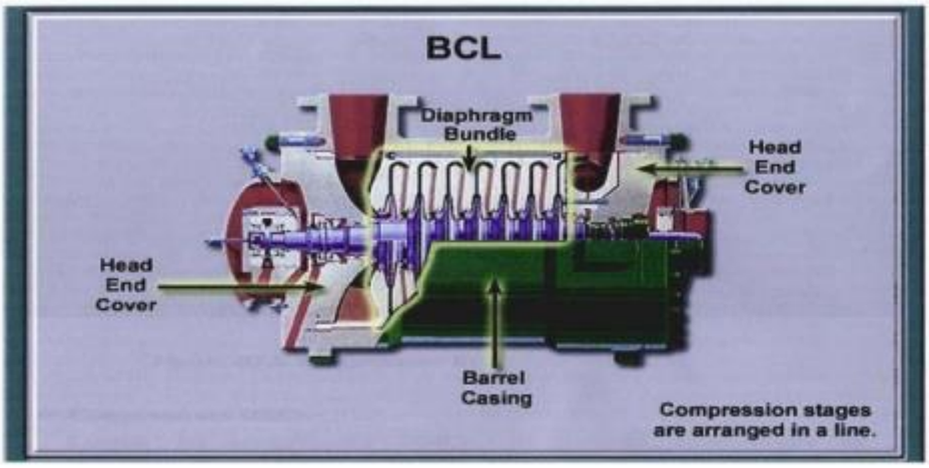

Figure I.17: BCL Compressors

- **2BCL compressors:**

These are barrel-type compressors with two compression stages in series in one body.

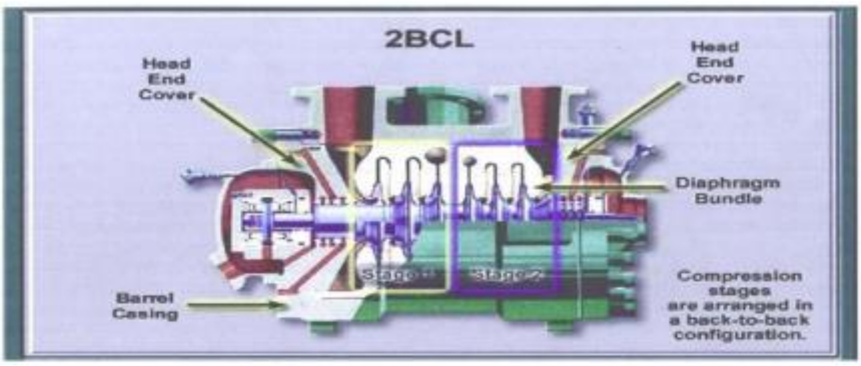

Figure I.18: 2BCL compressors

- **DBCL Compressors**

Like DMCL compressors, they perform two compression stages in parallel.

In a single body.

I.2.5.3 Compressors with bell-shaped bodies

- **DBCL with bell-shaped body:**

pressure-High barrel compressors have bell-shaped bodies and are closed by segments instead of bolts.

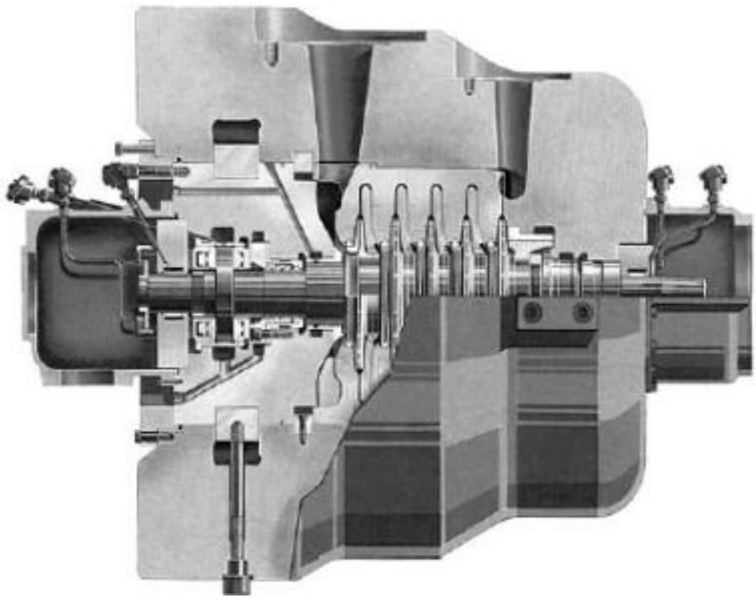

Figure I.19: DBCL with bell-shaped body

- **PCL-VHP compressors:**

The bodies of these compressors are bell-shaped with a single closing flange on one vertical plane instead of two; as for BCLs, generally، they are used for transporting natural gas suction، and discharge flanges are lateral and opposite to be able to connect them more easily to the pipelines of the pipeline.

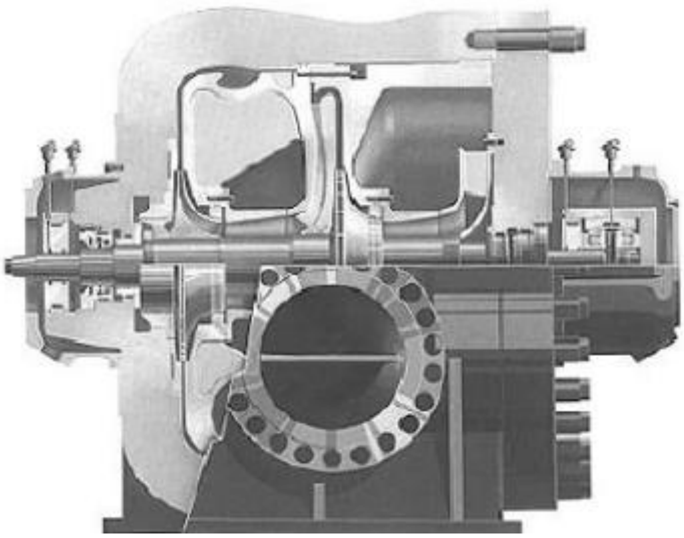

Figure I.20: PCL-VHP compressors

- **SR-type compressors:**

These are compressors for services at relatively low pressures.

Their characteristics are to have several shafts and relative wheels, cantilevered. The wheel is of the open type، that is to say، without acounter disc, to allow speeds ofHigh compression devices for each stage. The suction of each wheel is axial، while the discharge is radial. These compressors are typically used for ،air, steam, geothermal applications etc.

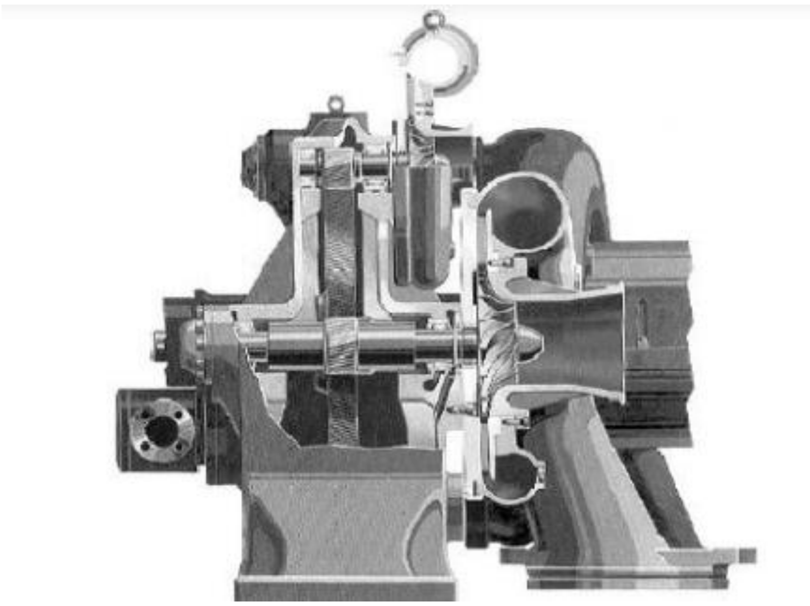

Figure I.21: SR-type compressors

I.2.6 Advantages of Centrifugal Compressor:

- ✓ They are relatively agile and easy to manufacture compared to other compressors.
- ✓ Since they do not require any special type of foundation, centrifugal compressors are highly energy efficient and reliable.
- ✓ They consist of only a few rubbing parts and are completely free-oil.
- ✓ They generate a higher-pressure ratio per stage than the axial flow compressor.
- ✓ Dynamic Compressor: Achieves a pressure rise by adding Kinetic Energy /Velocity to fluid
- ✓ Narrow operating range: Operates close to the design point due to its characteristics.
- ✓ Capacity control is simple، using either a Suction Throttle or Speed Control.
- ✓ Can be used for pushing large volumes of gas (large volumetric capacity

I.2.7 Disadvantages of Centrifugal Compressor:

- ✓ They can only produce a limited amount of pressure and cannot be relied upon for very high compression.
- ✓ Due to thier operation in a higher speeds requires a steady and sturdy mounting.

I.2.7 Compressor sealing:

I.2.7.1 Labyrinth seals:

A labyrinth seal is a type of mechanical seal that minimize gas leakages. Labyrinth seals on rotating shafts provide non-contact sealing action by controlling fluid passage through teeth.

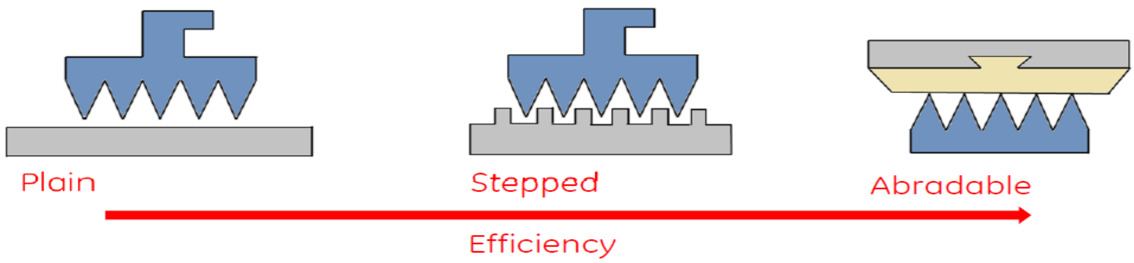

Figure I.22: Labyrinth seals

The main purpose is to minimize internal leakages close to the impellers due to different pressure values at the inlet and discharge.

I.2.7.2 Sealing gas system :

This system supplies filtered gas (buffer gas) to the seals mounted on the ends of the compressor shaft to prevent the passage of Process gas to fittings.
To ensure tightness, this gas is injected into the chambers which contain the seals at a pressure slightly higher than the pressure suction.

I.2.8 Surge phenomenon :

The Surge phenomenon is an axial one-dimensional instability affecting the global compression system. If the flow decreases, losses tend to increase in the compressor. When the machine operates in such a regime, fluctuations in the flow average can be observed, which can even go up to a total reversal of flow direction. Extreme fatigue endured by the blades makes this phenomenon very dangerous for the compressor[8].

I.2.8.1 Possible causes of Surge:

Causes that can cause compressor surge Centrifugal are:

- ❖ Reduced gas density due to weight change in the molecular weight of the gas or due to a rise in temperature at the inlet.
- ❖ Reduction of the compressor's rotation speed at the new rotation speed cannot produce the pressure the load requires.
- ❖ Increase pressure required by the load or by the industrial process served by the compressor.
- ❖ Blockage of any flow or suction chain or repression.
- ❖ Deformation of the geometry of the wheel by erosion and fouling.

I.2.8.2 Consequences of the surge:

- ➢ Unstable flow and pressure
- ➢ Damage increasing with severity to seals, bearings, impellers, and shaft
- ➢ Increased seal clearances and leakage
- ➢ Lower efficiency
- ➢ Reduced compressor life

It is possible to highlight in a diagram rate of compression – the rate of a limited operating zone by a curve called the Surge Limit Line SLL.

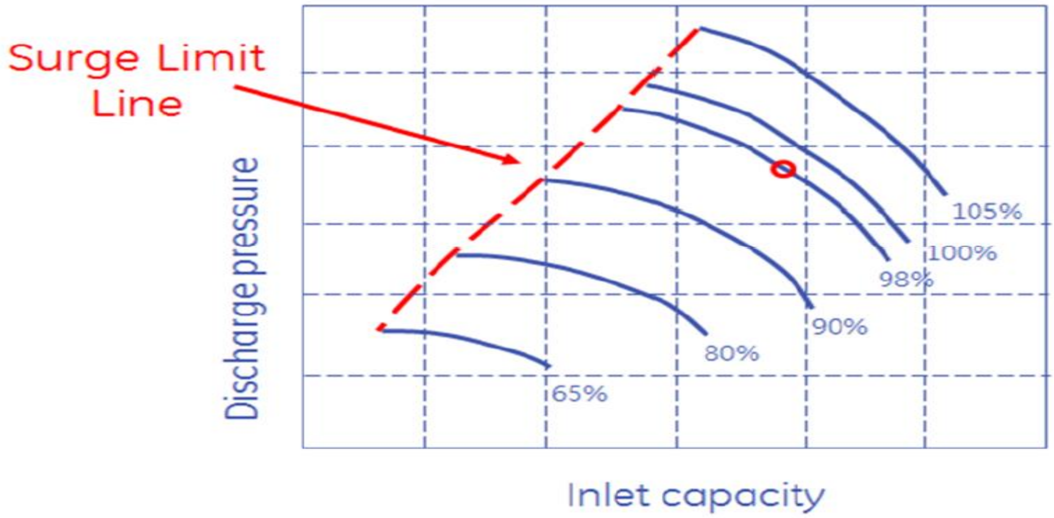

Figure I.23: Consequences of the surge

To allow the machine to always work above the limit of the surge, it is essential to permanently ensure the compressor has sufficient gas flow by an anti-surge protection system.

I.2.8.3 Anti-surge protection system:

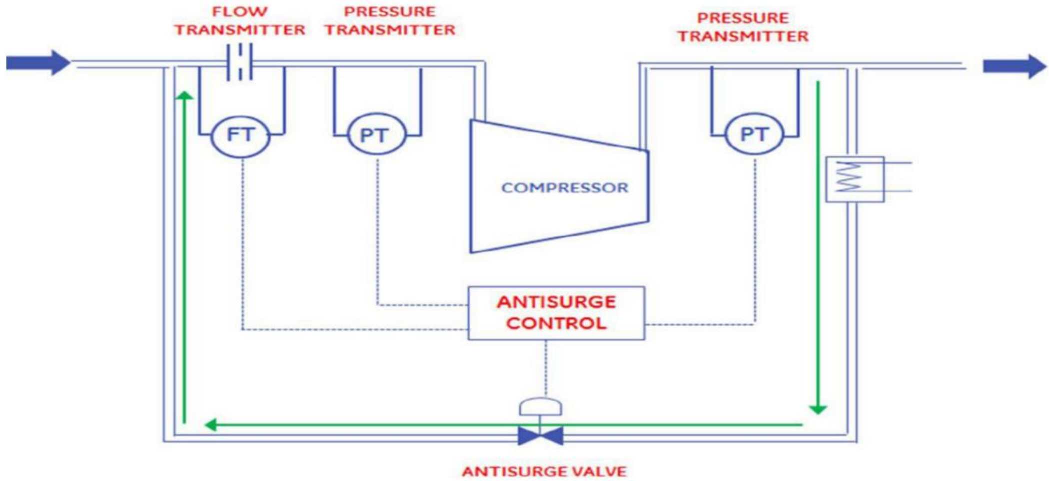

Figure I.24: Anti-surge protection system

This system contains valves, measuring, and control instruments. In case of a decrease in flow resulting from an increase in the ratio of compression, the purpose of this device is to:

- Either artificially increase the flow that crosses the compressor.
- Either to reduce the compression ratio.

I.3 pumping station:

Pump stations are large industrial facilities that maintain the flow and pressure of oil by receiving oil from the pipeline, re-pressurizing it, and sending it back into the pipeline system. (Compressor stations do the same thing for natural gas.)

Pump stations are generally constructed every 20-100 miles along a pipeline route, depending on the terrain, the capacity of both the pipeline and the station, and the type of product moving through.

Figure I.25: pumping station

Chapter II

Description of K101A compressor

II Description of K101A compressor:

II.1 The Definition:

The K101A compressor is a DRESSER-CLARK Model 441B 5/5 vertical seal "barrel" cylindrical centrifugal compressor.

The DRESSER-CLARK 441B 5/5 designation is:

- The first two digits of the model number refer to the capacity at the maximum pressure supported by the model in question.
- The third digit is the size of the frame determined by physical criteria (ex: diameter of the housing bore)
- the letter "B" designates the product line,
- 5/5 is the number of impellers in the first and second sections successively.
- Two types of rotor configurations are available depending on the specific operating conditions:
- ✓ Unit with a "direct" rotor.
- ✓ unit with a "back-to-back" rotor [9].

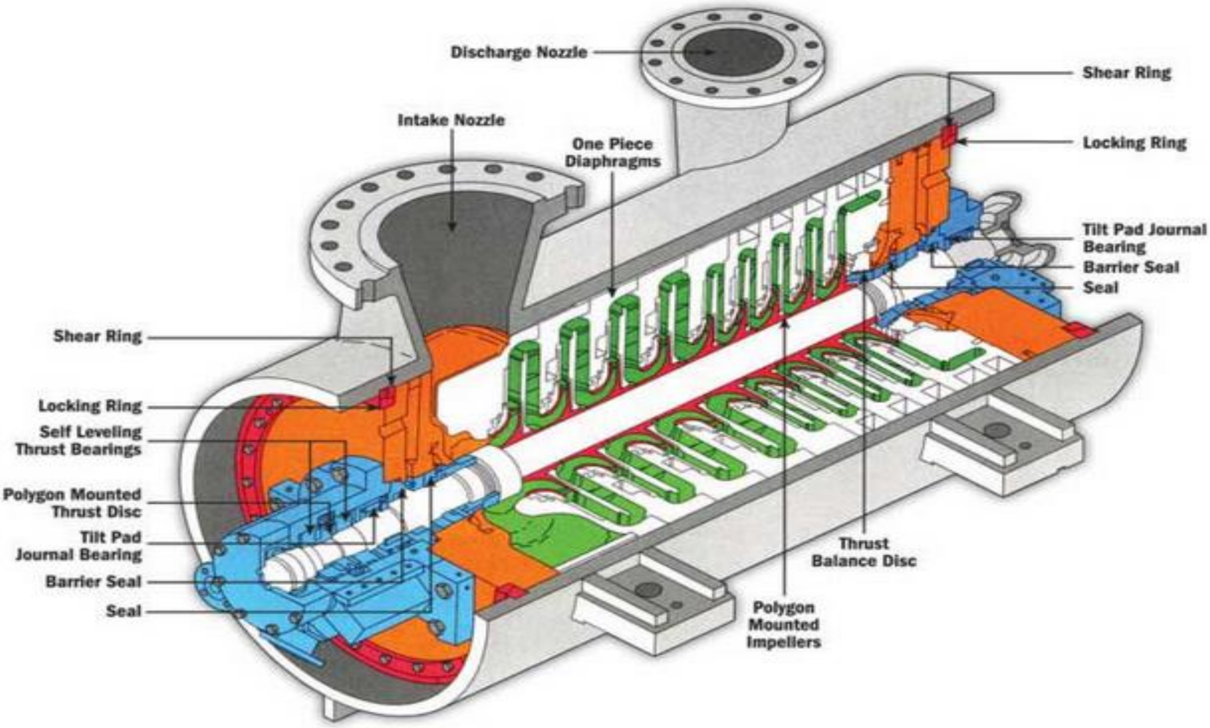

Figure II.1: The Definition of K101A compressor

II.2 The components :

Compressor components are divided into two categories:

• Fixed parts

• Rotating parts

Even if there are different types of centrifugal compressors, the components are mostly the same.

Because many machines can have the same TAG, the internal components make the difference

between machines used for different services and handling various types of gas.

II.2.1 Fixed parts :

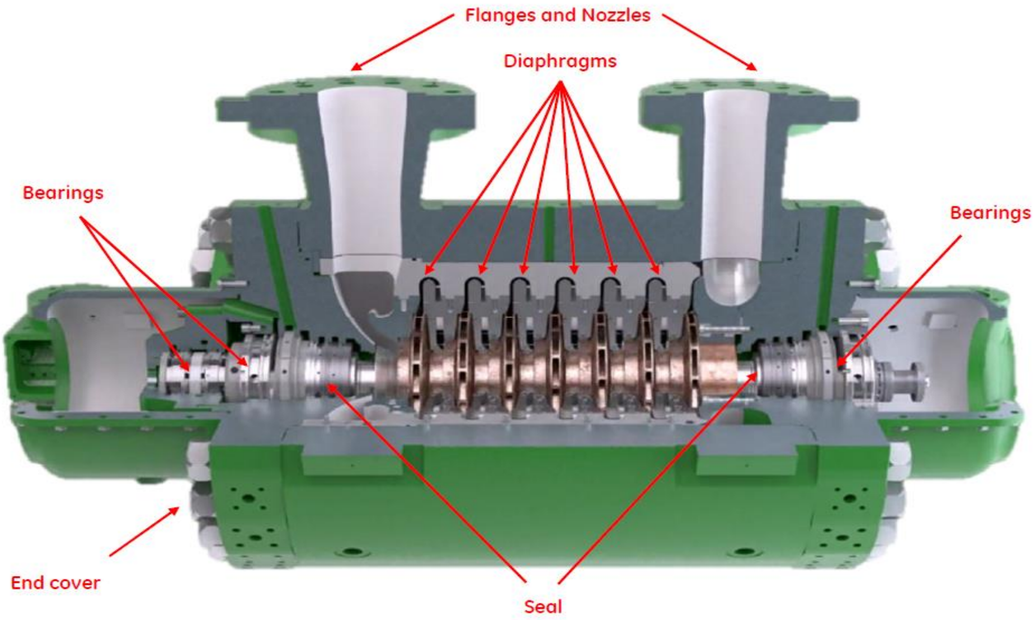

Figure II.2: Fixed parts

Casing:

Also called the body for a centrifugal compressor, envelopes and covers

are obtained by forging to make the material more homogeneous and, therefore, more resistant, considering the high pressures these compressors must work. Typically, carbon steel is used for the cylindrical body. The brackets and closing flanges are the adopted carbon content (0.2-0.25%).

Figure II.3: Casing

The diaphragms:

In K101A compressors, the diaphragms in two halves may be placed into split cylinder halves called counter casings.

The assembled counter casings, called diaphragm bundle assembly, are horizontally split to permit the installation of the rotor.

A compressor bundle is formed when the bundle halves are bolted with the rotor inside.

There are two designs used for the diaphragm bundles. The first design uses a thin, horizontally split counter casing with a cylindrical outer surface.

The bore is machined with diaphragm seals.

These bundle halves are bolted together using a series of bolts.

In the second design, the diaphragms in each bundle half are held together by a through bolt.

Figure II.4: The diaphragms

Nozzles:

Nozzles are usually welded to the casings only for high-pressure compressors (K101A type). The nozzles are machined inside the casing, and the piping is directly connected to the casing.
According to process requirements, the K101A compressor nozzles can be in the upper or lower part of the casing.

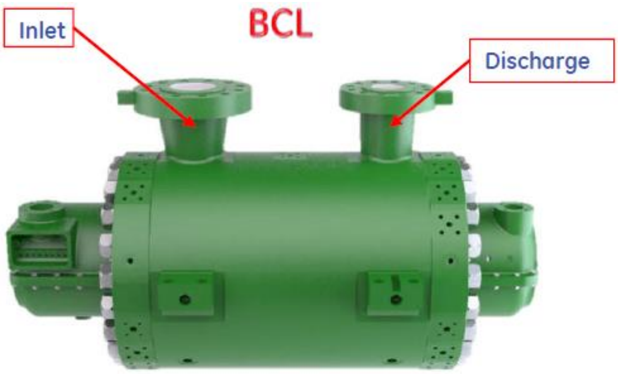

Figure II.5: Nozzles

Journal bearing:

The functions of the journal bearing are:

- Support the rotor weight
- Create a lifting effect with the lube oil
- Damping effect

Journal bearing with Tilting pads, tilting according to a sense of rotation, are adopted to have better stability, reducing cross-coupling effects and oil whips. The pad surface is made of a thin babbitt/nonfiction metal layer.

Figure II.6: Journal bearing

Two types of possible lubrication:

- Direct lubrication: Lube oil is provided for each pad through a groove manufactured on the pads.
- Flooded lubrication: The oil enters from an orifice in the upper part of the bearing and exits in the lower part after the lubrication of the pads.

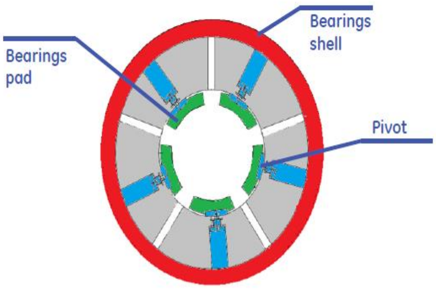

Figure II.7: Parts of Journal bearing

Thrust bearing:

The function of the thrust bearing is to compensate for part of the axial thrust. The bearing is divided into an Active and an Inactive part; if the compressor is operating correctly, the Inactive part is under the effect of the axial thrust during transient operation.

Both the Active and Inactive sides are the same.

The Thrust bearing type used is called 'Double acting self-equalizing' because the load is equally distributed among the pads.

The pad surface is made of a thin babbitt/non-friction metal layer.

Figure II.8: Thrust bearing

Two types of possible lubrication:

• Direct Lubricated: lube oil is provided for each pad through a groove manufactured on the pads.

• Flooded: Oil fills the bearing assembly and drains from the top through a flow control orifice.

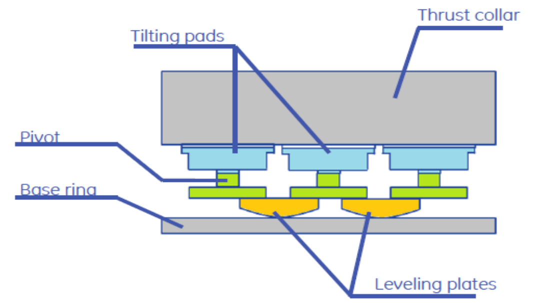

Figure II.9: parts of Thrust bearing

Instrumentation – axial displacement and radial vibration probes:

During the operation of the compressor, the change in the compression process due to various phenomena, like a surge or increased clearances, can create displacements or vibrations.
To monitor the effects on the compressor, axial displacement, and radial vibration probes are used.
Both are non-contacting probes, working with the eddy current principles and are usually connected to a Bently Nevada system for monitoring.
Axial displacement probes are installed on the Thrust Bearing shaft end and are a minimum of 2.
Radial vibration probes are installed on each journal bearing and work as a couple measuring vibration amplitudes on X and Y axes .

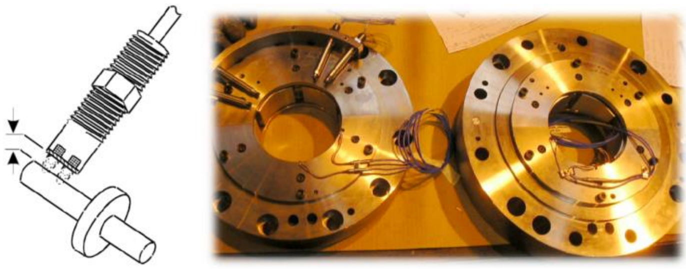

Figure II.10: Instrumentation

II.2.2 Rotating part:

As the rotating part, we identify all those components assembled to the shaft. This will include the Coupling Hub, Balance drum, Spacers, Impellers, and Thrust collar.

❖ **The shaft:**

It consists of a central part with a constant diameter where the impeler and intermediate bushings are mounted. It is made of steel with better mechanical properties 40NCD7.

Figure II.11: The shaft

❖ **Thrust collar:**

The thrust collar transmits the rotor thrust to the thrust bearings and fixes the axial position of the rotor.

It is hydraulically shrink-fitted to the shaft, making the assembly/disassembly procedure easier.

The thrust collar can also be threaded for balancing.

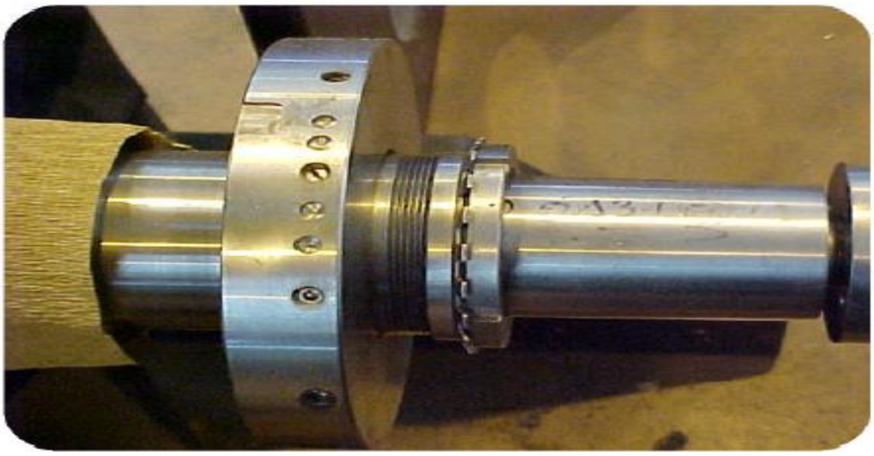

Figure II.12: Thrust collar

❖ Balance drum:

The balance drum is a steel disk installed after the last impeller.
Is interference fitted and keyed to the shaft, with threaded holes on one side to help rotor balancing?
The function of the balance drum is to compensate for the majority of axial thrust generated within the compressor.

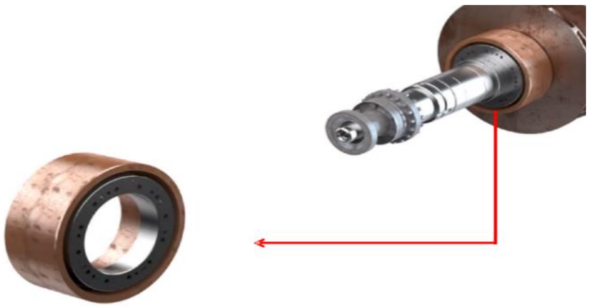

Figure II.13: Balance drum

II.3 Lube oil system :

The lubricating oil is supplied by the lubricating oil of the turbine which drives the compressor.
The filtered oil at the required temperature arrives at the head of the bearing where it is operated. The pressure is kept constant at approximately 1.75 bar g by the turbine lubrication system (PGT 10).
The bearing lubricating oil pressure is controlled using a calibrated orifice in each oil inlet line to the bearings.
This pressure is of the order of 0.9 to 1.3 bars g at the carrier bearings and 0.3 to 1.3 at the thrust bearings.

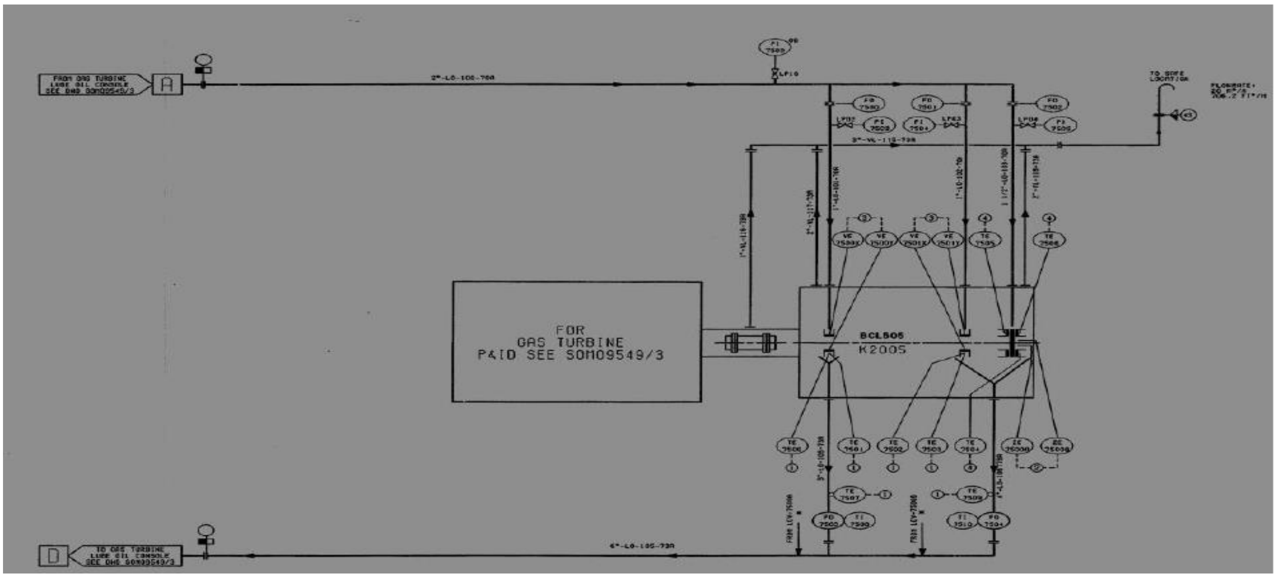

Figure II.14: Lube oil system

II.4 Gas Path & Axial Thrust :

According to the section arrangement of a centrifugal compressor, the internal components can be resumed in different sections.

This section nomenclature is used to identify the different areas where the gas passes and this is also called GAS PATH.

The gas flows through the compressor along the gas path.

Prior to starting, an overview is in order: a compression stage consists of the impeller, which is the rotating part, and the diffuser, which is the stationary part.

We can split a compressor gas path into different main sections:

• Suction stage

• Discharge stage

• Intermediate stage/s (multistage compressors)

Related to the operation of the machine, and also explained in this section, there is the case called Axial thrust[6].

- **Suction:**

Starting from the suction end, we can say that the gas flow from the pipe arrives at the suction stage traveling in a radial direction and must be reoriented into an axial direction to enter the impeller.

The purpose of the suction stage is to convey the gas into the first impeller with low load losses and uniform pressure and velocity.

Usually, the suction stage has a fin positioned at the opposite side of the suction flange, which splits the gas flow to avoid gas recycling in the inlet volute.

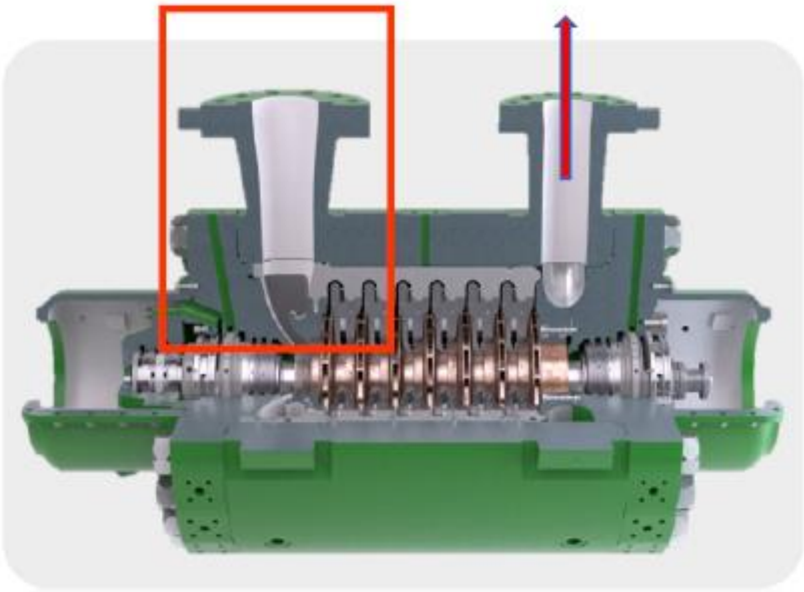

Figure II.15: Gas Path

- **Intermediate stages:**

The intermediate stages are the stages between the suction and the discharge; their number depends on how many impellers are installed on the shaft.

Because each compressor is fully customized, the geometry of the intermediate stages will change from compressor to compressor, according t operating specifications.

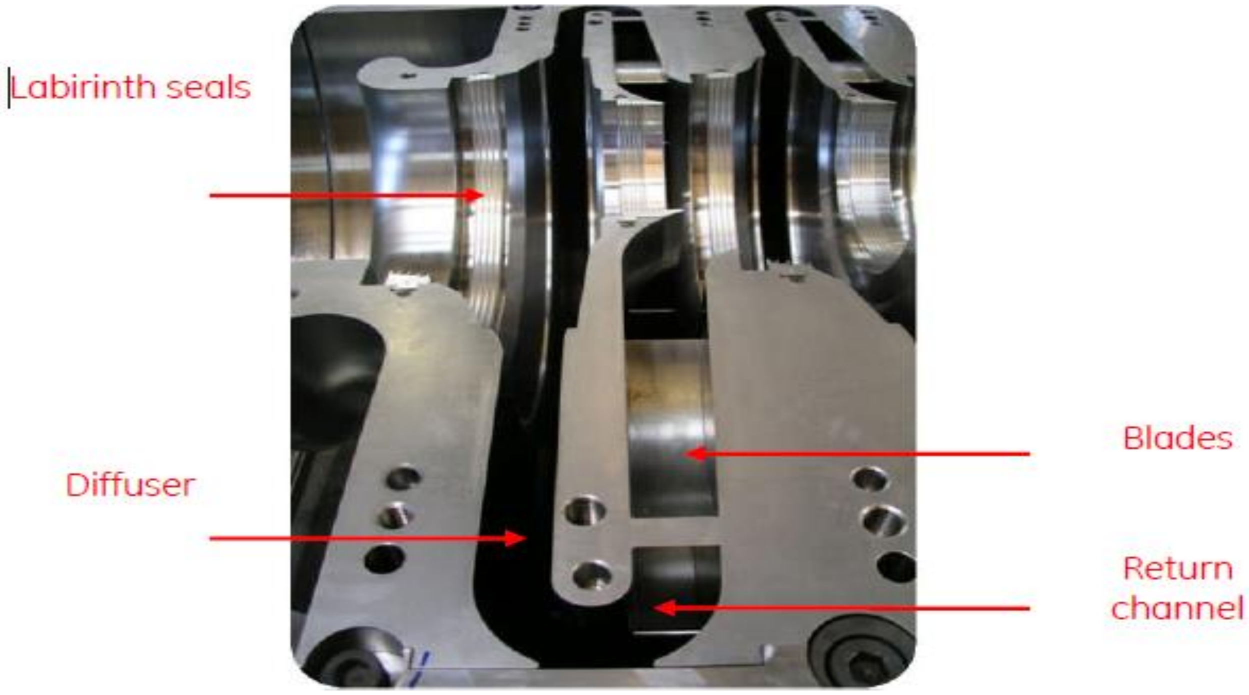

Figure II.16: Intermediate stages

- **Compressor stage:**

A compressor stage consists of the impeller, the diffuser, and the return channel.

The impeller rotates with the shaft, increasing the gas's velocity and pressure.

Regarding the pressure increase, we can say that it occurs within the impeller and the diffuser and that the magnitude of this increase depends on the design of both.

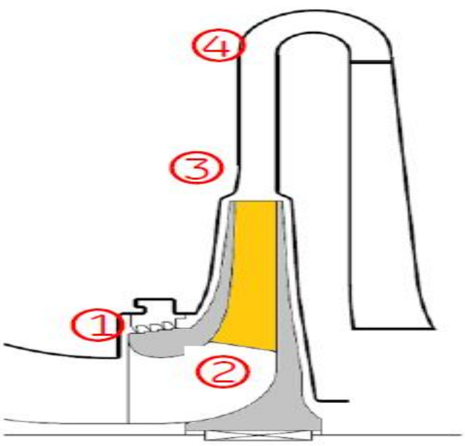

Figure II.17: Compressor stage

-As shown in the diagram below: in the impeller, both the velocity and pressure increase; in the diffuser the pressure increases and the velocity decreases due to the convergent desi, and in the return channel there is no increase in pressure because the convergent design is chosen only to avoid load losses.

The return channel is bladed to more effectively convey the gas toward the next impeller inlet, thus reducing load losses, but the. Still, therefore increase of pressure down the diffuser outlet.

All the pressure increase realized within a compressor stage is made between the blade inlet and the diffuser outlet.

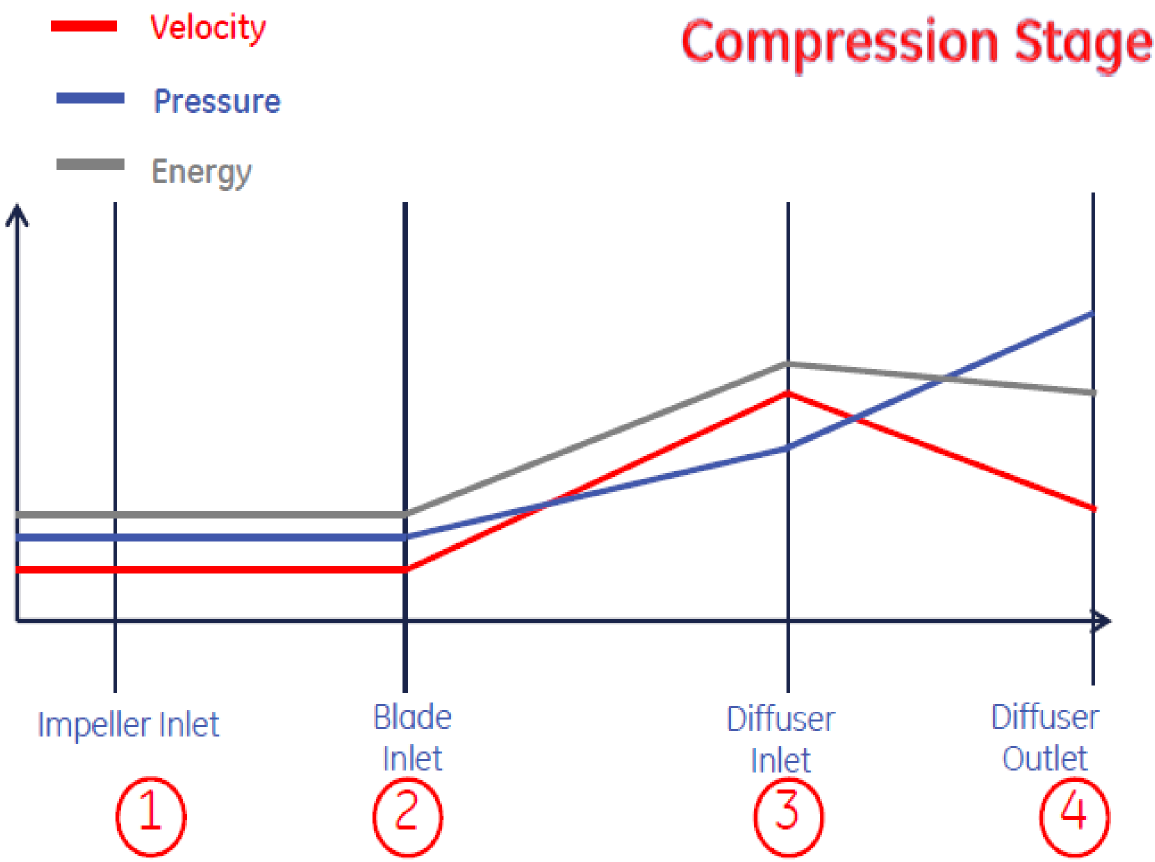

Figure II.18: Compression stage

- **Discharge volute:**

The discharge section of the gas path is the connection between the inner parts of the compressor and the piping. The most important part is the Discharge volute.

The discharge volute is installed after the last diffuser and conveys the gas into the discharge piping.

The volute has a variable section to impart low load losses to avoid decreasing the pressure.

A fin is usually installed near the discharge flange to prevent gas recirculation within the volute.

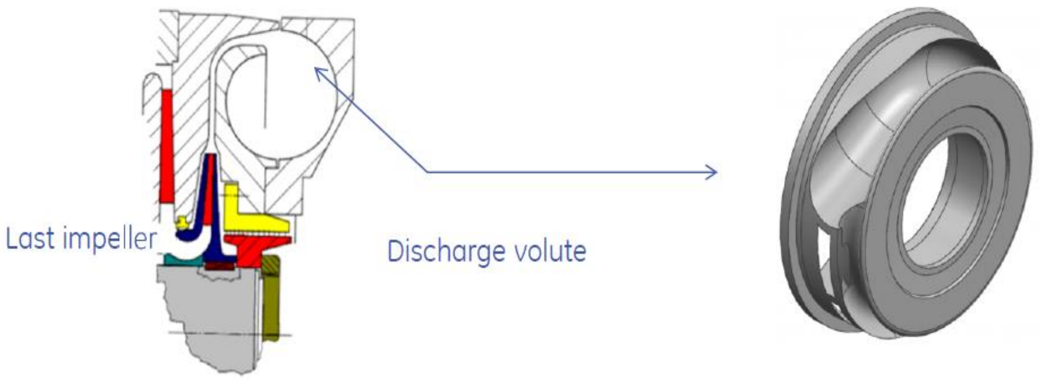

Figure II.19: Discharge volute

- **Axial thrust:**

-During operation, an axial thrust is generated inside the compressor along the entire length of the rotor.

This axial thrust is caused mainly by pressures acting against the impellers.

The disc side of the impeller is exposed to discharge pressure, while the hub side is exposed partly to the discharge pressure and partly to suction pressure.

This pressure distribution generates a force generally in the direction of suction. The total value of this axial force depends on pressure levels and the surface area of the impeller upon which these pressures act.

Because these forces do not mutually compensate each other, an axial load is generated upon the compressor rotor.

This axial load may reach a very high value in multistage, high-pressure compressors.

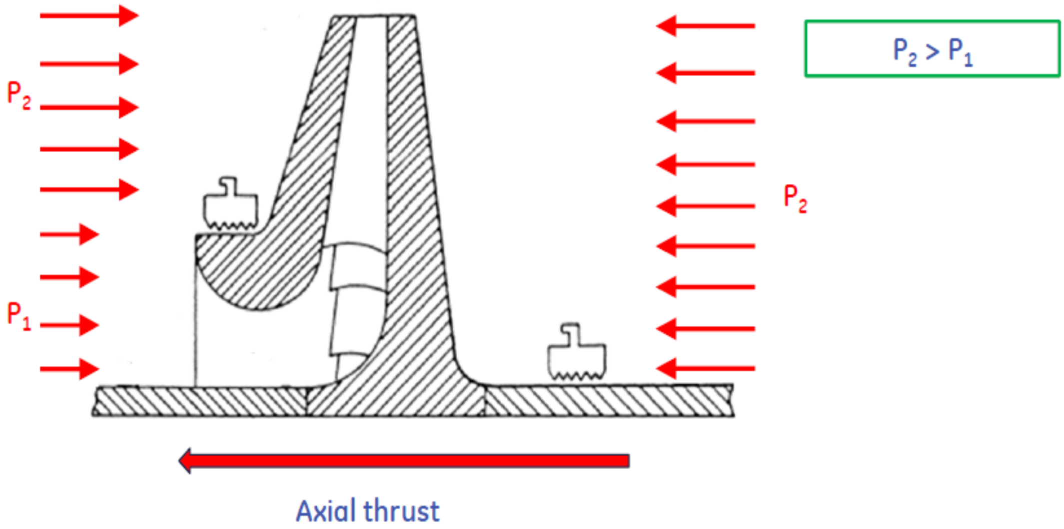

Figure II.20: axial thrust

The amount of axial thrust is not constant; it varies depending on the pressure value and the amount of mass flow that the compressor is elaborating.

The gas flow produces a thrust towards the compressor's discharge side, and, its amount depends on the mass flow.

- **Balancing drum:**

The thrust bearing is always installed on all compensate for axial load.

Additionally, to reduce the axial load applied to the thrust bearing to a more convenient value and reduce the size of the thrust collar, which is, an over hangover hanging ten, a balancing drum is provided.

This balancing drum is placed after the last impeller of the rotor, at the discharge end of the compressor, so the journal bearings support its weight.

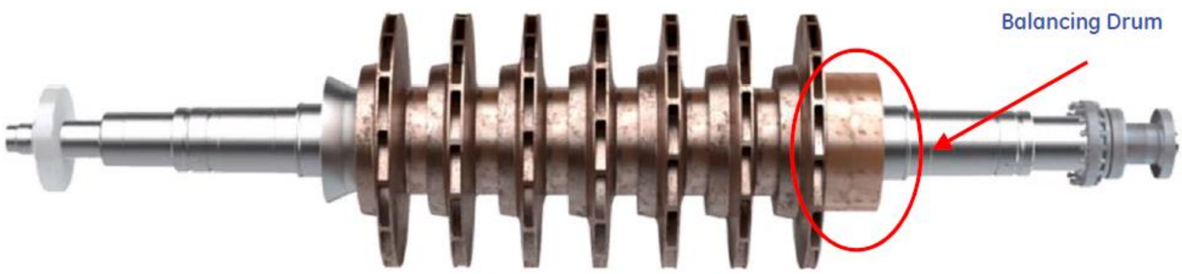

Figure II.21: Balancing drum

-One side of the balance drum is exposed to discharge pressure, while the other side is exposed to suction pressure through a

balancing line connected to the suction volute.

The diameter of the balance drum is designed to produce a thrust from the suction side to the discharge side of the

compressor.

This reduces the load from impellers placed upon aree thrust bearing.

The compensation normally is incomplete, thus avoiding any axial instability in the rotor.

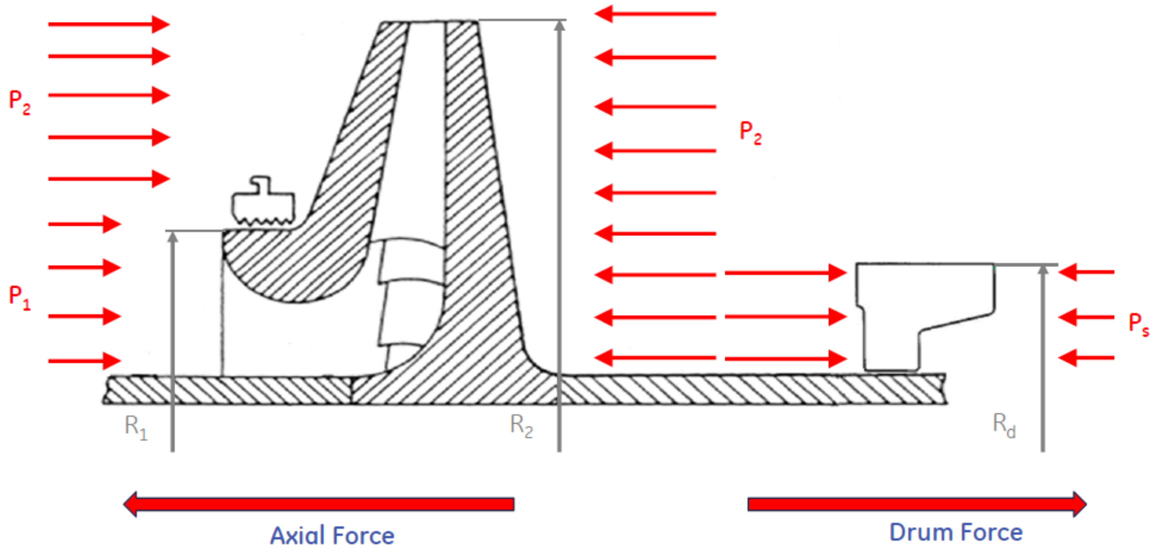

Figure II.22: axial thrust and drum force

Chapter III

Performance calculation

III.1 Theoretical study :

III.1.1 Thermodynamic study of gas compression:

after thermodynamics, the amount of energy supplied to the gas, namely the work "$\boldsymbol{dW}$" and th amount of heat"$\boldsymbol{dQ}$" can be expressed as the variation; "$\boldsymbol{dh}$" and that of the kinetic energy "$\boldsymbol{d(\omega^2/2)}$for the unit of mass"
$\boldsymbol{m = 1}$ **kg** . It is expressed in j/kg or in meters if divided by the force of gravity(g), this is called «Height».

$dW + dQ = dh + d\left(\frac{\omega^2}{2}\right)$ $\boldsymbol{\omega}$: Gas angular velocity in[rad/s]

This equation represents one of the forms of the equation of the first principle of thermodynamics relative to gas flow.

The heat is always italice for the compressors, for formed($\mathbf{d}\boldsymbol{\omega} = \mathbf{0}$) because the gas velocities at the inlet and outlet of a compressor are approximately equal an, d the work required to compress the gas can be calculated from the following form:
We have: $d(\omega^2/2) = 0$
And therefore:

$$dh = dw + |dq| \qquad (1)$$

The thermodynamic study of the compression often carried out by means of diagrams($\boldsymbol{h}, \boldsymbol{s}$) , allows to determine the variation of the enthalpy in compressor.

$$(\Delta \boldsymbol{h} = \boldsymbol{h}_2 - \boldsymbol{h}_l)$$

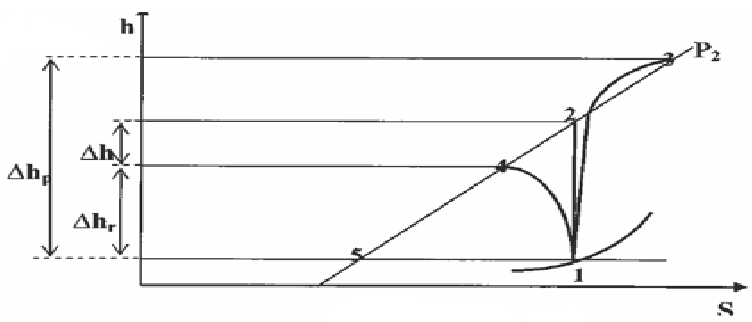

Figure III.1. Graph of variation the enthalpy in compressor

- the reversible adiabatic compression in an ideal compressor without loss of energy is represented by the right **(1-2)** because in this case($\boldsymbol{Q = 0}$) and the variation of the enthalpy($\boldsymbol{S_2 - S_1 = 0}$) .$\Delta h = W$.
- actual compression without cooling takes place according to the curve (1-3) and is always accompanied by losses $\boldsymbol{\Delta h_p}$ as well as the increase of entropy $(dS > 0)$.

- compression with cooling **(1-4)** for which, according to equation (2) $\Delta h = W - Q$.

- for perfect gases**Δh** is calculated from the specific heat at constant pressure $\mathbf{C}_\mathrm{p}$.

$$W = \Delta h = C_p(T_2 - T_1) \qquad (2)$$

Where:

- $\mathbf{T_1}$: suction temperature
- $\boldsymbol{T_2}$: discharge temperature

There is sometimes more control to anal italic cooperation italic Cresson us informer diagram $(\boldsymbol{P}, \boldsymbol{V})$ because the area in this diagram corresponds to the value of the work.

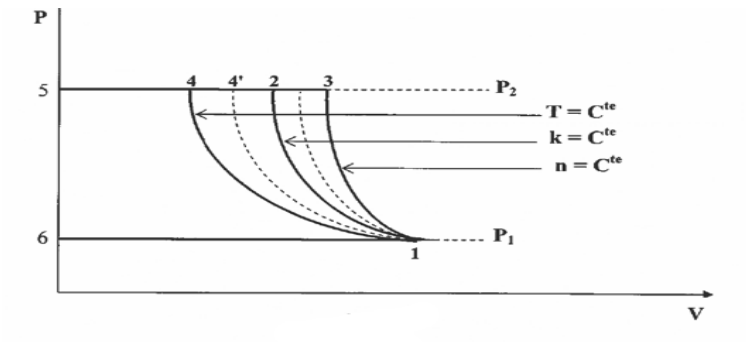

Figure III.2. **Graph of diagram** $(\boldsymbol{P}, \boldsymbol{V})$

Theo expresses the work, **W**„ cording to the pressure **P** and the specific volmeformerr gas:(V) the relation for the enthalpy must be used.

$$h = U + Pv \qquad (3)$$
$$dh = du + PdV + VdP \qquad (4)$$

Based on the equation of the first principle of thermodynamics for a variable volume system such as$(dP = 0)$; constant pressure.

We have: $dQ = dU + PdV$ (5)

By reporting (5) and (6) to (1) we obtain:

$$dW = VdP \tag{6}$$

$$W = \int VdP \tag{7}$$

And so, the work is represented informeriagram$(P - V)$ by the area limited by the thermodynamic transformation curve.

- For adiabatic compression (1-2) the work$(\boldsymbol{W})$ corresponds to the area (1-2 - $5 - 6 - 1$) which is between the adiabatic $(1 - 2)$ with the adiabatic exponent $(\boldsymbol{K = cst})$ and the two straight lines (1-6) and (2-5).

The actual compression with internal losses is performed with the polytropic exponent.

- The isotherm represents the isothermal compression (1-4).

III.1.2 The work of gas compression:

In turbochargers, adiabatic compression is usually used without cooling the machine's body; the adiabatic work can be calculated from the expression (3) for perfect gases.

But it is sometimes more commands to express the value of W as a function of compression ratio that is usually known.

$$\varepsilon = \frac{p_2}{p_1} \tag{8}$$

In the case of adiabatic transformation:

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\left(\frac{k-1}{k}\right)} \tag{9}$$

Replacing equation (10) with equation (3), we find:

$$W_{ab} = z \cdot c_p T \left(\varepsilon^{\left(\frac{k-1}{k}\right)} - 1 \right) \qquad (10)$$

C_p : explained on Kj/Kg · °K

$roundround$

$$C_p - C_v = R \qquad (11)$$

and $K = C_p/C_v$

$$C_p = \left(\frac{k}{k-1}\right) \cdot R \qquad (12)$$

Replacing equation (13) in (11), we find:

$$W_{ad} = \left(\frac{k}{k-1}\right) \cdot z \cdot R \cdot T_1 \left(\varepsilon^{\left(\frac{k-1}{k}\right)} - 1 \right) \qquad (13)$$

This equation calculates the adiabatic work of the reversible transformation without internal losses saying: "isentropic work."

By replacing the adiabatic exponents**k** with the polytropic exponent all adiabatic equations can be used to calculate polytropic transformations. It emerges that the polytropic work of a reversible transformation without energy loss without Tirith(**n = var)** is calculated from the equation:

$$W_p = \left(\frac{n}{n-1}\right) \cdot z \cdot R \cdot T_1 \cdot \left(\varepsilon^{\left(\frac{n-1}{n}\right)} - 1 \right) \qquad (14)$$

The actual adiabatic work $\mathbf{W}_r$ of an irreversible transformation with(**n = var**) according to the expression (3) is equal to:

$$W = c_p (T_{2r} - T_1) \qquad (15)$$

Where: $\mathbf{T_{2r}}$: Actual discharge gas temperature.

If we know that$\left(\mathbf{n} = \mathbf{c}^{\mathrm{te}}\right)$ the exponent$(\mathbf{1} - \mathbf{3})$ of the polytrope (fig.2) passing through point 3 corresponds to the actual compression we can write:

$$\frac{T_{2r}}{T_1} = \left(\varepsilon^{\left(\frac{n-1}{n}\right)}\right) \qquad (16)$$

Relating equations (13) and (17) to equation (16), we obtain the following:

$$Wr = \left(\frac{k}{k} - 1\right) Z \cdot R \cdot T1 \cdot \left(\varepsilon^{\left(\frac{n-1}{n}\right)} - 1\right) \qquad (17)$$

III.1.3 Thermodynamic efficiency of a compressor:

In order to calculate the actual work $\boldsymbol{W_r}$ required to compress (**1kg**) gas, one of the thermodynamic efficiencies of the compressor is used:

$$\eta^{th} = \frac{W_{th}}{W_r} \qquad (18)$$

$\boldsymbol{W_{th}}$: Thermodynamic work of the compressor

Are must be adapted according to the particularities of the italic, often without work $\boldsymbol{W_{ad}}$ and adiabatic efficiency
are generally used.

$$\eta_{ad} = \frac{W_{ad}}{W_r} \qquad (19)$$

$$\eta_{iso} = \frac{W_{is}}{W_r} \qquad (20)$$

$$\eta_p = \frac{W_p}{W_r} \qquad (21)$$

This polytropic yield does not accurately account for actual losses but only characterizes the difference between actual work and thermodynamic work with $\left(\boldsymbol{n} = \mathbf{c}^{\mathbf{th}}\right)$.

Relations (11), (16), and (20) for compressors without cooling have:

$$\eta_{ad} = \frac{\left(\varepsilon^{\left(\frac{\mathrm{n}-1}{n}\right)} - 1\right)}{\left(\frac{T_{2r}}{T_1} - 1\right)} \qquad (22)$$

According to (15), (18) and (22):

$$\eta_p = \left(\frac{n}{n-1}\right)\left(\frac{\mathrm{k}-1}{k}\right) \qquad (23)$$

In practice the polytropic exponent is not known and must be replaced by the known parameters:

$$n = \frac{\log \varepsilon}{\log(\frac{Z1.T1.P2}{Z2.T2.P2})} \qquad (24)$$

III.2 Operating parameters :

TableIII.1: table showing operating parameters[7]:

1^{st} stage	
Control room	
Suction pressure; in [bar]	Pa = 2,1
Discharge pressure; in [bar]	Pr = 9,8
Suction temperature; in [K]	Ta = 300,15
Discharge temperature; in [K]	Tr = 414,09
Manufacturer	
Mass flow; in [Kg/h]	Qm= 4460,4
Number of impeler	N = 5
Compressibility coefficient	Z = 0,983
Specific heat; in [KJ/Kg.C °]	Cp= 2,35
Rotation speed; in [rpm]	n =13060
2^{nd} stage	
Control room	
Suction pressure; in [bar]	Pa = 9,8
Discharge pressure; in [bar]	Pr = 33,5
Suction temperature; in [K]	Ta = 349,15
Discharge temperature; in [K]	Tr = 440,67
Manufacturer	
Mass flow; in [Kg/h]	Qm = 22748,4
Number of impeler	N = 5
Specific heat; in [KJ/Kg.C °]	Cp = 2,42
Rotation speed; in [rpm]	n = 13060

III.3 Calculation of work on the 1st stage:

Gas characteristics:

TableIII.2: characteristics of the gases present in the compressor

Components	Molecular concentration X%	Molecular concentration (μ) [Kg/K°.mol]	Specific heat at constant pressure Cp [KJ/Kg.K°]	Critical temperature [K°]	Critical pressure [bars]
N_2	4,19	28,02	1,621	126,2	33,92
CO2	0,6	44,01	1,031	304,1	73,84
CH4	67,34	16,04	2,204	190,56	45,96
C2H6	20,02	30,07	1,714	305,33	48,72
C3H8	6,08	44,09	1,624	369,85	42,48
C3H8	0,44	58,12	1,678	407,85	36,41
nC4H10	1,02	58,12	1,620	425,16	37,97
iC5H12	0,12	72,15	1,625	460,4	33,82
nC5H12	0,19	72,15	1,601	469,7	33,70

III.3.1 Molar mass of the gas mixture:

$$M = \sum M_i Y_i$$

M=M(N2) Y (N2) + M (CO2) Y (CO2) + M(CH4) Y (CH4) + M(C2H6) Y (C2H6) + M(C3H8) Y (C3H8) + M(C3H8) Y (C3H8) + M(nC4H10) Y(nC4H10) + M(iC5H12) Y (iC5H12) + M(nC5H12) Y(nC5H12)

$\mathrm{M} = 22.01\mathrm{Kg/k°.\ mole}$

- **determination of the specific gas constant:**

$$\mathrm{r} = \frac{R}{M} = \frac{8{,}314}{22.01} \Rightarrow r = 0{,}38[\mathrm{kj/kg \cdot k°}]$$

- **calculation of adiabatic coefficients:**

$$r = Cp - Cv$$

$$K = \frac{Cp}{Cp - r}$$

$$K = \frac{2.16}{2.16 - 0{,}38} = 1.21 \Rightarrow K = 1.21$$

- **calculation of compression ratio:**

$$\varepsilon = \frac{P_r}{P_a} = \frac{9{,}8}{2{,}1} = 4{,}67 \Rightarrow \varepsilon = 4{,}67$$

- **polytropic coefficient:**

$$\frac{n}{n-1} = \frac{\log \varepsilon}{\log \frac{T_r}{T_a}}$$

$$n = \frac{\log \varepsilon}{\log \varepsilon - \log \frac{T_r}{T_a}}$$

$$n = \frac{\log 4{,}67}{\log 4{,}67 - \log \frac{414{,}09}{300{,}15}} = 1.21 \Rightarrow n = 1.21$$

III.3.2. Adiabatic work:

The compressibility coefficient and the temperature Ta is given by the manufacturer in the table where $\mathrm{T_a} = 300{,}15$ K and $Z = 0{,}983$

$$W_{ad} = \frac{K}{K-1} Z \cdot r \cdot T_a \left(\varepsilon^{\frac{K-1}{K}} - 1\right)$$

$$W_{ad} = \left(\frac{1.21}{1.21 - 1}\right) \times 0{,}983 \times 0{,}38 \times 300{,}15 \times \left(4{,}67^{\frac{1.21-1}{1.21}} - 1\right)$$

$$\Rightarrow W_{ad} = 197.01[\mathrm{KJ/Kg}]$$

III.3.3. polytropic work:

$$W_p = \frac{n}{n-1} \cdot Z \cdot r \cdot T_a \cdot \left(\varepsilon^{\frac{n-1}{n}} - 1\right)$$

$$W_p = \frac{1{,}26}{1{,}26 - 1} \times 0{,}98 \times 0{,}38 \times 300{,}15 \times \left(4{,}66^{\frac{1{,}26-1}{1{,}26}} - 1\right)$$

$$\Rightarrow W_p = 202.5[KJ/Kg]$$

III.3.4. Actual work:

$$W_r = \frac{K}{K-1} \cdot Z \cdot r \cdot T_a \cdot \left(\varepsilon^{\frac{n-1}{n}} - 1\right)$$

$$W_r = \frac{1.21}{1.21-1} \times 0,98 \times 0,38 \times 300,15 \times \left(4,67^{\frac{1,26-1}{1,26}} - 1\right) = 241.93[\text{kj/kg}]$$

$$\Rightarrow W_r = 241.93[\text{KJ/Kg}]$$

.The results obtained are shown in Table

Table III. 3: the work of the first stage of the compressor

r	N	K	Z	T_a	ε	W_{ad}	W_p	W_r
0,38	1,26	1,21	0,983	300,15	4,67	197.01	202.5	241.93

III.4 Calculation of work on the 2nd stage:

- ✓ **Compressibility coefficient of the 2nd stage:**

a) suction:

•Critical temperature of the mixture: $T_{cr} = \sum X_i T_{cri} \Rightarrow Tcr = 226,62[K°]$

•Critical pressure of the mixture: $P_{cr} = \sum X_i P_{cri} \Rightarrow P_{cr} = 45,80$ [bars]

$$T_{\text{rea}} = \frac{T_a}{T_{cr}} = \frac{349,15}{226,62} = 1,54[K°]$$

$$P_{\text{rea}} \frac{P_a}{P_{cr}} = \frac{9,8}{45,80} = 0.21[\text{ bars }]$$

$$Z_a = 1 - 0,4273 \cdot \frac{0,21}{(1,54)3,66} = 0,98$$

b) discharge:

$$T_{\text{rea}} = T_r/T_{cr} = \frac{440,67}{226,62} = 1.54[K°]$$

$$P_{\text{rea}} = \frac{P_r}{P_{cr}} = 0.21[\text{bar}]$$

$$Z_r = 1 - 0,4273 \frac{P_{\text{rea}}}{(T_{\text{rea}})3,66} = 1 - 0,4273 \frac{0.21}{5.64} = 0.96 \Rightarrow Z_r = 0,96$$

$$Z_{\text{moy}} = \frac{Z_a + Z_r}{2} = \frac{0,98 + 0,96}{2} = 0,97 \Rightarrow Z_{\text{moy}} = 0,97$$

- **Calculation of adiabatic coefficients:**

$r = Cp - Cv$

$$K = \frac{Cp}{Cp - r}$$

$$K = \frac{2.284}{2.284 - 0{,}38} = 1.2 \Rightarrow K = 1.2$$

- **calculation of compression ratio:**

$$\varepsilon = \frac{P_r}{P_a} = \frac{33{,}5}{9{,}8} = 3.42 \Rightarrow \varepsilon = 3.42$$

- **polytropic coefficient:**

$$\frac{n}{n-1} = \frac{\log \varepsilon}{\log \frac{Tr}{Ta}}$$

$$n = \frac{\log \varepsilon}{\log \varepsilon - \log \frac{Tr}{Ta}}$$

$$n = \frac{\log 3{,}41}{\log 3.42 - \log \frac{440{,}67}{349{,}15}} = 1{,}23 \Rightarrow n = 1{,}23$$

III.4.1. Adiabatic work:

$$W_{ad} = \frac{K}{K-1} Z \cdot r \cdot T_a \left(\varepsilon^{\frac{K-1}{K}} - 1\right)$$

$$W_{ad} = \frac{1.2}{1.2-1} \times 0{,}97 \times 0{,}38 \times 349{,}15 \times \left(3{,}41^{\frac{1.2-1}{1.2}} - 1\right) = 174.34[\text{kj/kg}]$$

$$\Rightarrow W_{ad} = 174.34[\text{KJ/Kg}]$$

III.4.2. polytropic work:

$$W_p = \frac{n}{n-1} Z \cdot r \cdot T_a \cdot \left(\varepsilon^{\frac{n-1}{n}} - 1\right)$$

$$W_p = \frac{1{,}23}{1{,}23-1} \times 0{,}97 \times 0{,}38 \times 349{,}15 \times \left(3{,}42^{\frac{1{,}23-1}{1{,}3}} - 1\right) = 177.03$$

$$\Rightarrow W_p = 177.03[\text{KJ/Kg}]$$

III.4.3. Actual work:

$$W_r = \frac{K}{K-1} Z \cdot r \cdot T_a \cdot \left(\varepsilon^{\frac{n-1}{n}} - 1\right)$$

$$W_r = \frac{1,2}{1,2-1} \times 0,97 \times 0,38 \times 349,15 \times \left(3,41^{\frac{1,23-1}{1,23}} - 1\right) = 202.77[\text{KJ/Kg}]$$

$$\Rightarrow W_r = 202.77[\text{KJ/Kg}]$$

Table III. 4: the work of the second stage of the compressor

r	N	k	Z	T_a	ε	W_{ad}	W_p	W_r
0,38	1,23	1,2	0,97	349,15	3,42	174.34	177.03	202.77

III.5 Overall Compressor Work:

III.5.1. adiabatic work:

$W_{ad} = W_{ad1} + W_{ad2}$

$W_{ad} = 194.78 + 174.34 = 369,21[KJ/Kg]$

$\Rightarrow W_{ad} = 371.35[KJ/Kg]$

III.5.2. Polytropic work:

$W_p = W_{p1} + W_{p2}$

$W_p = 202.50 + 177.03 = 379.53[KJ/Kg]$

$\Rightarrow W_p = 379.53[KJ/Kg]$

III.5.3. Actual work:

$W_r = W_{r1} + W_{r2}$

$W_r = 241.93 + 202.77 = 444.69[\text{KJ/Kg}]$

$\Rightarrow W_r = 444.69[KJ/Kg]$

Table III. 5: the overall work of the compressor

Adiabatic work W_{ad}	polytropic work W_p	Actual work W_r
371.35	379.53	444.69

III.6 efficiency calculation:

- ✓ **adiabatic efficiency:**

$$\eta_{ad} = \frac{W_{ad}}{W_r}$$

$$\eta_{ad} = \frac{371.35}{444.69} = 0.84 \Rightarrow \eta_{ad} = 84\%$$

> polytropic efficiency 1^{st}stage

$$\eta_p = \frac{W_p}{W_r}$$

$$\eta_p = \frac{202.5}{241.93} = 0.84$$

> polytropic efficiency 2^{nd}stage

$$\eta_p = \frac{W_p}{W_r}$$

$$\eta_p = \frac{177.03}{202.77} = 0.87$$

overall polytropic efficiency:

$$\eta_p = \frac{W_p}{W_r}$$

$$\eta_p = \frac{379.53}{444.69} = 0.85 \Rightarrow \eta_p = 85\%$$

- ✓ **overall efficiency:**

The mechanical efficiency of the compressor: $\eta_{méc} = 0.95$.

Compressor volumetric efficiency: $\eta_{vol} = 0.98$.

$$\eta_g = \eta_{ad} \times \eta_{\text{vom}} \times \eta_{\text{méc}}$$
$$\eta_{\text{méc}} = (0,92 \div 0,96)$$
$$\eta_g = 0.84 \times 0,98 \times 0,95 = 0,78 \Rightarrow \eta_g = 78\%$$

III.7 Power calculation [9]

- **the power transmitted to the 1st stage gas:**

$P_u = Q_m * W_r$

$$\text{Pu} = 1.24 * 241.93 = 299.75[\text{kw}]$$

- **the power transmitted to the 2nd stage gas:**

$P_u = Q_m * W_r$

$\text{Pu} = 6.32 * 202.77 = 1281.28\ [\text{kw}]$

- **The power absorbed 1st stage:**

$p_\alpha = \frac{P_u}{\eta\text{p}} = \frac{299.75}{0.84} = 358.10\text{KW}$

- **The power absorbed 2nd stage:**

$$p_\alpha = \frac{P_u}{\eta\text{p}} = \frac{1281.28}{0.87} = 1467.57\text{KW}$$

- **Coupling power :**

We take into account the mechanical transmission losses which are between 2% and 4%, we will admit here 4%.

$P_{ac} = p_a * 1.04$

$P_{ac} = 3240,53 * 1,04 = 3370,15[\text{kw}]$

Chapter IV

Results and interpretation

IV.1 Calculation of intermediate pressures:

Knowing the compression ratio, we can easily calculate the intermediate pressures by the following formula:

$P_{i+1} = \zeta P_i$: stage number;

- P_i : pressure of stage (i);
- P_{i+1}: pressure of stage (i+1);
- ζ: compression ratio

. from Equation in calculus

$$\zeta = \sqrt[n]{\varepsilon_1}$$
$$\zeta_1 = \sqrt[5]{4,6} \Rightarrow \zeta_1 = 1,36$$
$$\zeta_2 = \sqrt[5]{3,41} \Rightarrow \zeta_2 = 1,27$$

The calculation results are given in table:

Table. IV.1: intermediate pressures in each impeller:

1st stage					
P_{as} [bar]	P_1 [bar]	P_2 [bar]	P_3 [bar]	P_4 [bar]	P_5 [bar]
2,1	2,85	3,88	5,28	7,18	9,8

2nd stage					
P_{as} [bar]	P_6 [bar]	P_7 [bar]	P_8 [bar]	P_9 [bar]	P_{10} [bar]
9,8	12,4	15,75	20,01	25,41	33,5

By observing the curve of the internal pressure change at each impeller, we conclude that each impeller has a different degree of pressure.

When the gas moves from the impeller to the impeller, its pressure increases; from impeller 1 to impeller 10, the pressure increases from 2.1[bar] to 33.5[bar].

In the case of cooling, we observe that the pressure stays fixed.

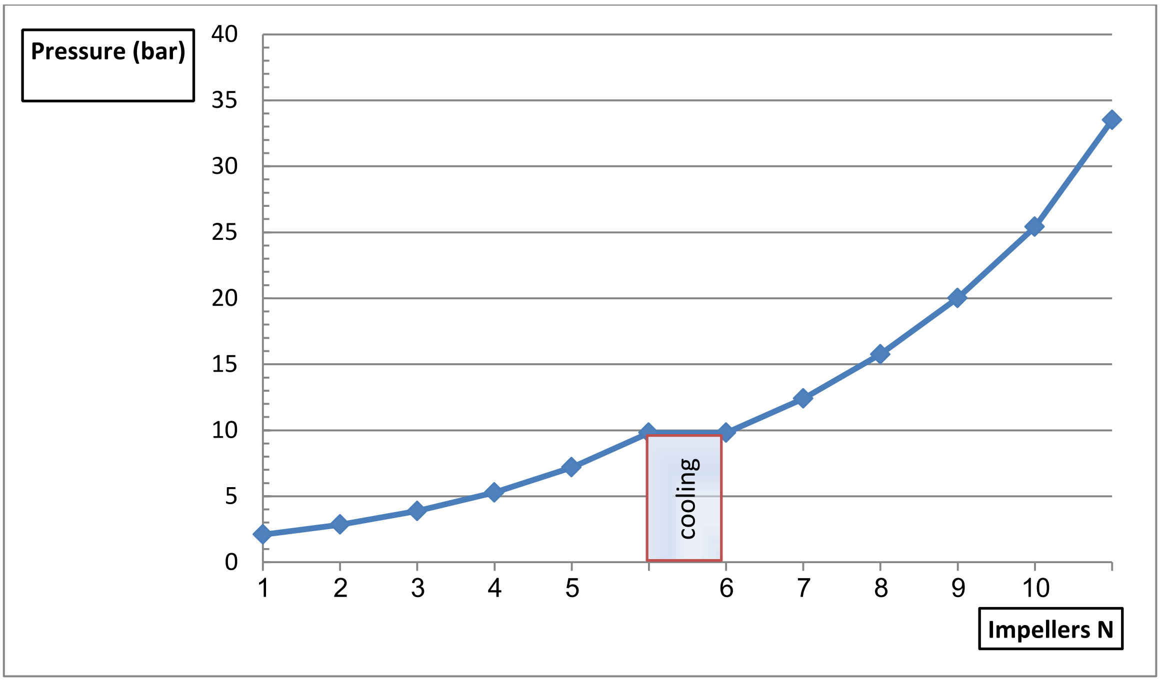

Fig IV.1: graph of intermediate pressures in each impeller

IV.2 Calculation of intermediate temperatures:

$$\frac{T_{i+1}}{T_I} = \left(\frac{P_{i+1}}{P_i}\right)^{\frac{n-1}{n}}$$

T_{i+1}: temperature of stage (i+1);

T_i: temperature of the stage (i)

The calculation results are shown in Table:

Table. IV.2: Intermediate temperatures in each impeller

1st stage					
T_{as} en [K]	T_1 [en [K]]	T_2[en[K]]	T_3 [en [K]]	T_4 [en [K]]	T_5[en[K]
300.15	319.67	340,68	363,04	386,82	414.09

2ND stage					
T_{as} [en [K]]	T_6[en [K]]	T_7[en [K]]	T_8[en [K]]	T_9[en [K]]	T_{10}[en [K]]
349.15	364.86	381.54	399	417.24	440,67

By observing the curve of the internal temperature change at each impeller, we conclude that each impeller has a different degree of temperature.

When the gas moves from the impeller to the impeller, its temperature increases; from impeller 1 to impeller 10, the temperature increases from 300.15 [K] to 414.09 [K] in the first stage, then after the cooling increased from 349.15[K] to 440.67[K] in the second stage.

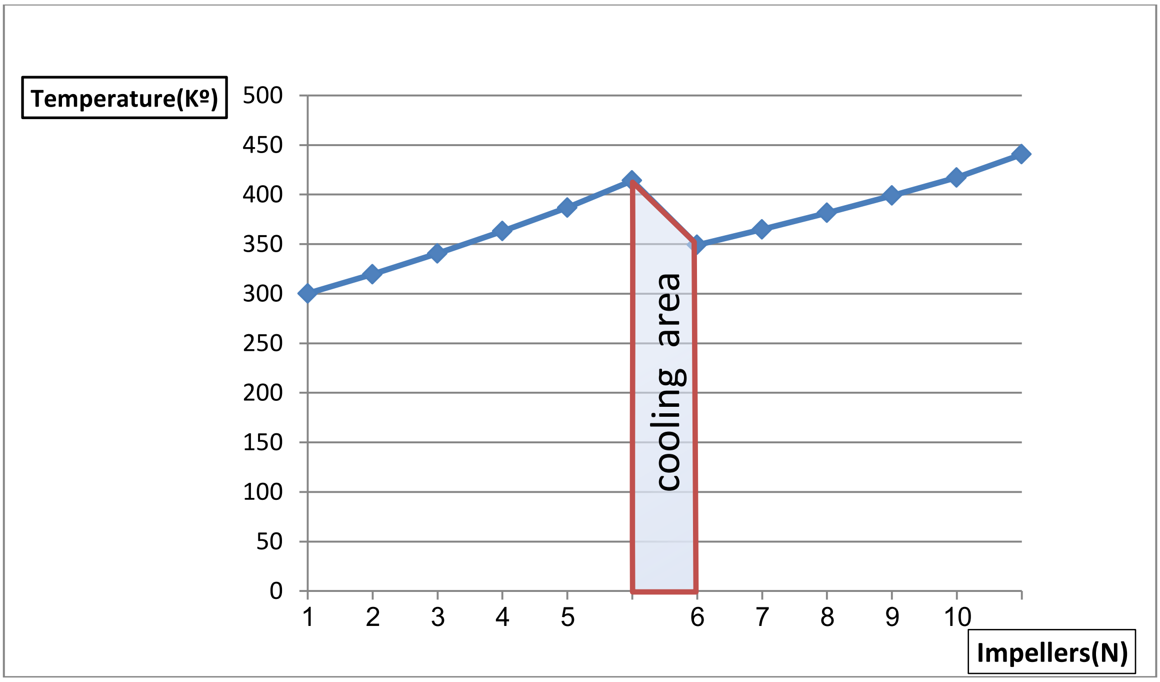

Fig IV.2: graph of intermediate temperatures in each impelle

IV.3 Interpretation of results:

Table. IV.3: Interpretation of results**:**

1st stage			
	Polytropic efficiency (%)	Actual work (KJ/Kg)	Absorbed power (KW)
Control room	85.00	223.00	343.00
Calculated	83.70	241.93	358.10
Relative gap	1.52	8.49	4.40
2nd stage			
	Polytropic efficiency (%)	Actual work (KJ/Kg)	Absorbed power (KW)
Control room	90.00	171.00	1326.00
Calculated	87.31	202.77	1467.57
Relative gap	2.99	18.58	10.68

The Relative gap of the (polytropic efficiency, real work, and absorbed power) from the Control room are notably weak; these Relative gaps are due to differences in the quality of the gas (temperature, molecular concentration% (our study case: CH_4: **67,34%**, C_2H_6: **20,02%**. Control room CH_4: **64,99%**, C_2H_6 :**16,21%**).

Therefore, the results are acceptable for an estimative study.

IV.4 Cooling effect on compressor performance:

After obtaining the previous results of the compressor **K101A**, we will study the effect of cooling $\Delta T(Td\text{-}Ts \geq 0)$ on the performance (polytropic efficiency (%), actual work(kJ/kg), and absorbed power(w)).

The effect of cooling on the polytropic efficiency (%):

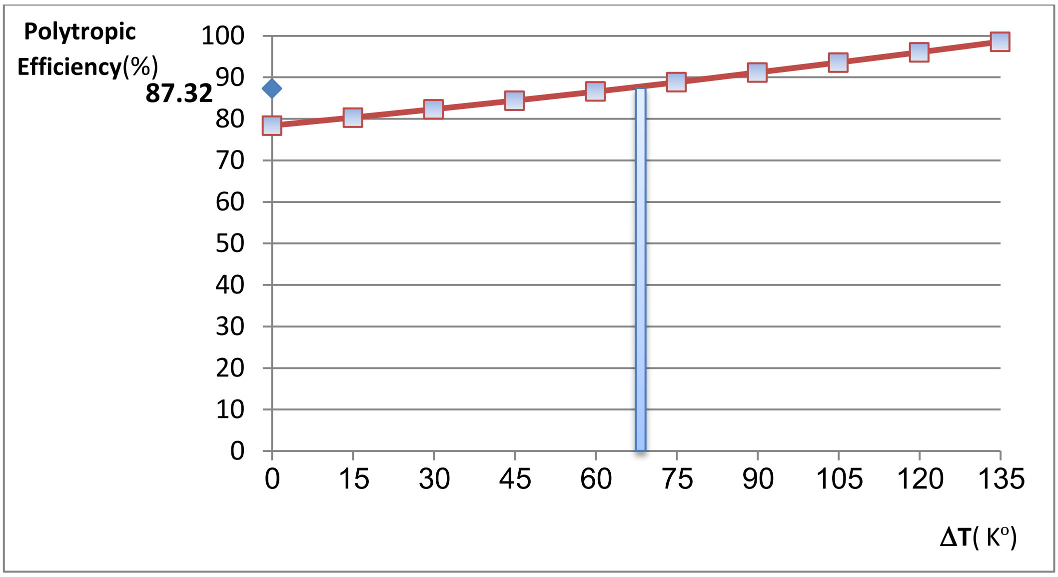

Fig IV.3: graph of the effect of cooling on the polytropic efficiency (%)

we observe the polytropic efficiency. It changes with the change in the temperature of the suction of 2nd stage.

We notice that the curve is rising from 78.36% to 98.55% between 0K° to 135 K°, which we obtained in our study case is on 87.32% when it is cooled to 65K°.

- when ΔT=0 (there is no cooling between stages) (Td1=Ts2) the polytropic efficiency = 78.36%.

- it means that the polytropic efficiency has increased with the decrease of the suction temperature in the 2nd stage.

The effect of cooling on the actual work(kJ/kg):

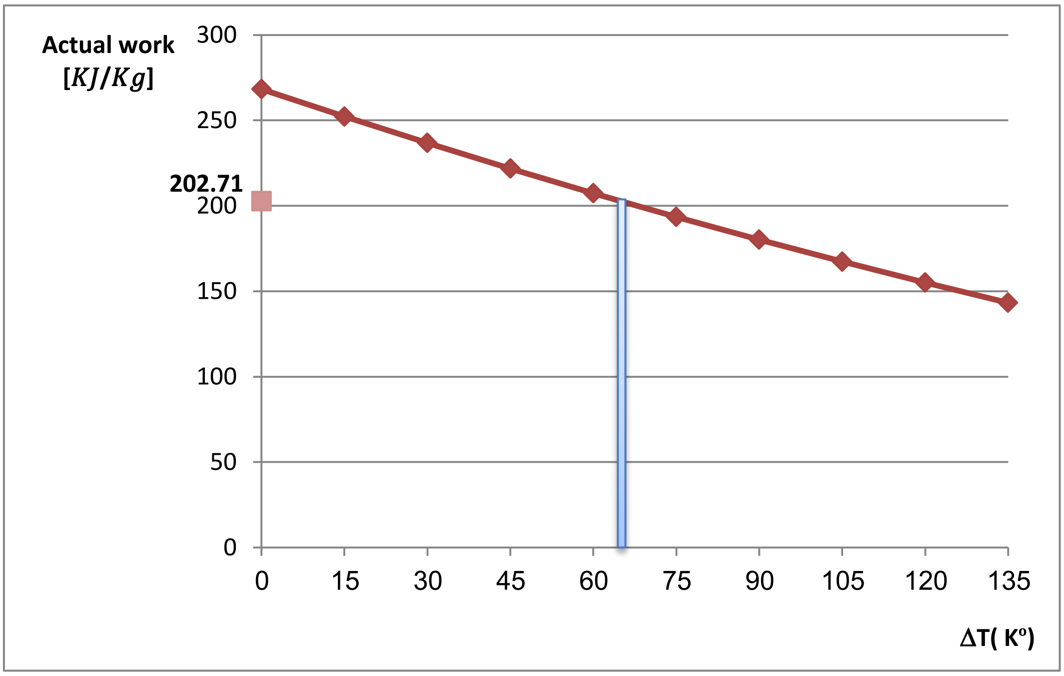

Fig IV.4: graph of the effect of cooling on the actual work(kJ/kg)

we observe the actual work. It changes with the change in the suction temperature of 2nd stage

Note that the curve is a rising curve from 268.29(kJ/kg) to 143.29(kJ/kg) between 0K° to 135 K°, where we obtained in our study case is on 202.71 when it is cooled to 65K°.

- when ΔT=0 (there is no cooling between stages) (Td1=Ts2) the actual work = 268.29(kJ/kg).

- it means that the actual work has decreased with the increase of the cooling of the discharge temperature in the 1st stage.

The effect of cooling on the absorbed power(w):

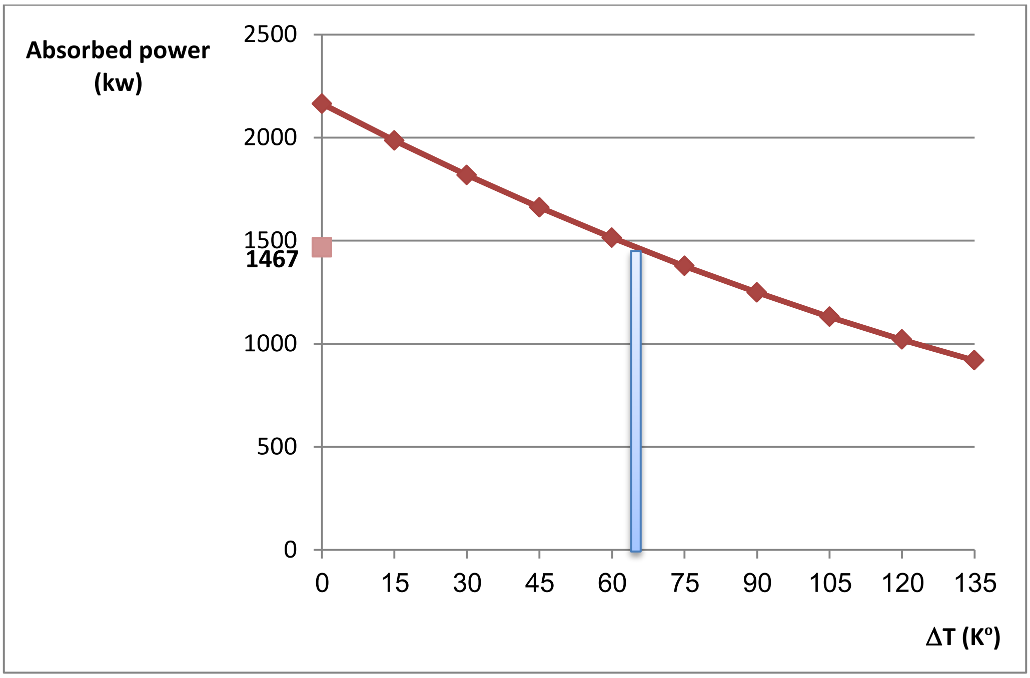

Fig IV.5: Graph of the effect of cooling on the absorbed power(w)

we will observe the absorbed power. It changes with the change of the suction temperature in 2nd stage.

We notice that the curve is a rising curve from 2163.52 (kJ/kg) to 918.8(kJ/kg) between 0K° to 135 K°, which we obtained in the case of the student on 1467 (kJ/kg) when cooled 65K°.

- when ΔT=0 (there is no cooling between stages) (Td1=Ts2) the absorbed power = 2163.52 (kJ/kg).

- it means that the absorbed power has decreased with the increase of the cooling of the discharge temperature in the 1st stage.

General conclusion

In this STUDY, we have dealt with the theme “study of the performance of K101 A multi-stage centrifugal compressor in the Oued station Noumer”. This subject is of great importance for the unit of production of Oud Noumer in the region of Hasi R'mel.

we performed a thermodynamic verification to verify the performance of the k101 centrifugal compressor.

First, we checked the thermodynamic parameters such as:

the efficiency (%), the actual work(kJ/kg), and the absorbed power (w). of the compressor, which was acceptable during our study at the station level، the temperature variation as a function of the pressure and the decrease in gas temperature between the two stages.

Finally, this study has been of great scientific use to us. It has enabled us to deepen our theoretical knowledge of the different equipment used in the Algerian oil industry, in particular on centrifugal compressors, which are very common in the latter.

We also hope for companies that give great importance to cooling because it increases the polytropic efficiency (%) and decreases the actual work(kJ/kg) and the absorbed power(w).

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Abstract

In this study, we performed a thermodynamic verification to verify the performance of the k101 centrifugal compressor. As a result we obtained that the gap with the control room is acceptable . We verify the evolution of the effect of the degree of cooling between the two stages (Td1 and Ts2) on the performance K101A centrifugal compressor .so that the increase in cooling produces the increase of the polytropic efficiency (%) and the decrease of actual work(kJ/kg), and the absorbed power (w).

Keywords

compressor centrifugal – performance - cooling