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Analisi e modellazione dei transitori di esercizio di compressori industriali

Analysis and modelling of operational transients in industrial compressors

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"When you can measure what you are speaking about, and express it in numbers, you know something about it; but when you cannot measure it, when you cannot express it in numbers, your knowledge is of a meagre and unsatisfactory kind: it may be the beginning of knowledge, but you have scarcely, in your thoughts, advanced to the stage of science, whatever the matter may be."
Lord Kelvin

Abstract

This doctoral thesis addresses the complex challenge of lifecycle management for centrifugal and reciprocating compression systems operating in critical industrial environments. Historically, the energy sector has suffered from a structural dichotomy between engineering departments, responsible for theoretical design, and operational workshops, tasked with overhaul and repair (O&R). This separation has generated an information asymmetry where vital data regarding actual failure modes and wear patterns rarely feeds back into the design cycle, limiting the evolution of machinery reliability.

The research aims to bridge this gap through a multidisciplinary approach that integrates advanced predictive modelling with empirical field validation. The first part of the study focuses on the theoretical analysis of the dynamic behaviour of compression trains, through the use of 1D gas-dynamic modelling and coupled CFD/FEM analysis, acoustic resonance phenomena in piping, to identify systemic criticalities that precede physical failure. Parallel to this, an in-depth analysis was conducted on material selection for turbomachinery and the flow distribution of progressive lubrication systems, essential for correct operation. The results of these theoretical investigations were made available through the publication of scientific articles specifically addressing the fluid dynamics of progressive divider valves and the dynamic analysis of centrifugal compressors.

Beyond the immediate technical enhancements (related to safety, as well), this research carries significant implications for industrial sustainability. The optimization of overhaul procedures and the scientific validation of component repairability through advanced metrology effectively promote a circular economy model within the heavy machinery sector. By extending the operational lifespan of high-value assets, such as piston rods, cylinder valves, and seal cartridges, through engineered restoration rather than routine replacement, the proposed methodology not only reduces the economic burden of maintenance but may also help in reducing or control the carbon footprint associated with the manufacturing and logistics of new spare parts.

The experimental core of the research was conducted during an international placement at Blackstone Industrial in Canada (Edmonton, Alberta). In this operational context, the study improved the overhaul process through crucial documentation. The implementation of new governance protocols via Inspection and Test Plans (ITP), combined with the adoption of digitalization technologies such as 3D laser scanning and quantitative non-destructive testing, allowed for the creation of a "digital thread" connecting the actual wear state of the component to original design parameters. Furthermore, the codification of tacit decentralized knowledge into structured training

manuals ensured the standardization of technical competencies, reducing human variability in the maintenance process. Simultaneously with these standardization efforts, the research findings were consolidated into extensive internal reporting and technical presentations to facilitate knowledge transfer among colleagues.

Finally, the thesis presents a specific case study on the safety of dry gas seal systems. Through lumped parameter fluid dynamic modelling of the venting system, the resilience of the tertiary barrier seal was analysed under catastrophic failure scenarios. The parametric analysis highlighted the limitations of standard plant configurations, demonstrating that the integration of an auxiliary vent valve is necessary to decouple plant safety from the degradation state of the primary seals.

In conclusion, after identifying several specific areas of particular interest for applications, through original activities of analysis, modelling, and design, it was possible to establish a strong connection between the engineering and maintenance sectors, highlighting the usefulness of a management approach based on this relationship and providing an original contribution to the field of industrial compressors (and, more broadly, heavy machinery).

Sommario

La presente tesi di dottorato affronta la complessa sfida della gestione del ciclo di vita dei sistemi di compressione centrifughi e alternativi operanti in ambienti industriali critici. Storicamente, il settore energetico ha sofferto di una dicotomia strutturale tra i dipartimenti di ingegneria, responsabili della progettazione teorica, e le officine operative, deputate alla revisione e riparazione (overhaul & repair - O&R). Tale separazione ha generato un'asimmetria informativa in cui i dati vitali relativi alle reali modalità di guasto e ai pattern di usura raramente sono riportati al ciclo di progettazione, limitando l'evoluzione dell'affidabilità delle macchine.

La ricerca mira a colmare questo divario attraverso un approccio multidisciplinare che integra la modellazione predittiva avanzata con la validazione empirica sul campo. La prima parte dello studio si concentra sull'analisi teorica del comportamento dinamico dei treni di compressione. Mediante modellazione gasdinamica 1-D e analisi accoppiate CFD/FEM, sono stati indagati i fenomeni di risonanza acustica nelle tubazioni, identificando le criticità sistemiche che precedono il guasto fisico. Parallelamente, è stata condotta un'analisi approfondita sulla selezione dei materiali per turbomacchine e sulla distribuzione del flusso nei sistemi di lubrificazione progressiva, essenziali per il corretto funzionamento.

Oltre agli immediati miglioramenti tecnici (che indirettamente incidono anche sulla sicurezza), questa ricerca comporta implicazioni significative per la sostenibilità industriale. L'ottimizzazione delle procedure di revisione e la validazione scientifica della riparabilità dei componenti, ottenuta tramite metrologia avanzata, promuovono efficacemente un modello di economia circolare nel settore dei macchinari pesanti. Estendendo la vita operativa di asset ad alto valore, quali steli dei pistoni, valvole dei cilindri e cartucce di tenuta, attraverso un ripristino ingegnerizzato piuttosto che la sostituzione, la metodologia proposta riduce non solo l'onere economico della manutenzione, ma permette di incidere anche sulla carbon footprint associata alla produzione e alla logistica delle nuove parti di ricambio.

Il nucleo sperimentale della ricerca è stato condotto durante un periodo di mobilità internazionale presso la sede canadese di Blackstone Industrial (Edmonton, Alberta). In questo contesto operativo, lo studio ha migliorato il processo di revisione attraverso l'implementazione di nuovi protocolli operativi, tramite piani di ispezione e collaudo (Inspection and Test Plans - ITP), combinata con l'adozione di tecnologie di digitalizzazione quali la scansione laser 3D e i controlli non distruttivi quantitativi. Ciò ha permesso la creazione di una "connessione digitale" che collega lo stato di usura reale del

componente ai parametri di progetto originali. Inoltre, la formalizzazione della conoscenza pratica e decentralizzata in manuali di formazione strutturati ha garantito la standardizzazione delle competenze tecniche, riducendo la variabilità umana nel processo di manutenzione.

Infine, la tesi presenta un caso studio specifico sulla sicurezza dei sistemi di tenuta a gas (Dry Gas Seals). Attraverso una modellazione fluidodinamica a parametri concentrati del sistema di sfiato (venting), è stata analizzata la resistenza della tenuta di barriera terziaria in scenari di guasto catastrofico. L'analisi parametrica ha evidenziato i limiti delle configurazioni impiantistiche standard, dimostrando che l'integrazione di una valvola di sfiato ausiliaria è necessaria per disaccoppiare la sicurezza dell'impianto dallo stato di degrado delle tenute primarie.

In conclusione, individuati alcuni specifici ambiti di particolare interesse per le applicazioni, attraverso originali attività di analisi, modellazione e progettazione, è stato possibile stabilire una forte connessione tra i comparti di ingegneria e manutenzione, evidenziando l'utilità di un approccio alla gestione basato su tale relazione, e fornire un contributo di ricerca originale nel settore dei compressori industriali (e, più in generale, delle macchine pesanti).

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1. Introduction

1.01. Funding Framework

This doctoral thesis is funded by the National Recovery and Resilience Plan (NRRP) (Italian: Piano Nazionale di Ripresa e Resilienza, PNRR), a massive investment initiative designed to drive post-COVID recovery and promote long-term structural competitiveness. Within the broader context of the NextGenerationEU program, a central pillar of this strategy is the ecological transition and the modernization of critical energy infrastructures.

To operationalize these strategic goals within the academic framework, the NRRP allocates specific resources to bridge the gap between theoretical research and industrial application, mandating international mobility and close collaboration with the private sector. Consequently, this doctoral path was structured around two distinct industrial placements to cover the entire spectrum of machinery management, from the initial design phase to operational maintenance.

The first phase, lasting twelve months, was conducted in Italy at the headquarters of Compression Service Technology (CST) in Florence. CST is an independent high-tech Design and Maintenance Engineering company with over 50 years of experience in the field of rotating machinery for gas compression. This placement provided deep exposure to sophisticated static and dynamic simulation studies, including pressure pulsations and structural calculations, as well as root cause analysis and troubleshooting. A key takeaway from this experience aligns with CST's core philosophy: the capability to work across the machinery life cycle, allowing for a continuous transfer of experience from the field to the design of new equipment, and vice versa.

The second phase, covering six months, was performed abroad at Blackstone Industrial, a company specialized in executing turnaround and overhaul work on critical rotating assets. This mobility was primarily based at the company's headquarters in Sherwood Park (Alberta, Canada) for five months, where the research focused on Blackstone's methodical and practical approach to maintenance in large-scale facilities. The experience concluded with a one-month tenure at the European branch in Asua-Erandio (Bilbao, Spain). This final segment was crucial to observe the application of these maintenance procedures within the specific constraints of the European regulatory framework.

This dual structure – combining the advanced engineering and simulation capabilities of CST with the practical, execution-oriented expertise of Blackstone – allowed for a holistic view of the machinery. In compliance with the NRRP mandate, this work

aims to contribute to industrial innovation in the energy sector, specifically focusing on enhancing efficiency and reliability (and safety) of critical centrifugal and reciprocating compressor assets.

1.02. Structure of the thesis

This doctoral thesis is organized into distinct sections that mirror the multidisciplinary nature of the research project, bridging the gap between theoretical mechanical design and operational reality. The document follows a logical progression, moving from system-level dynamic analysis to component-specific material science, and finally arriving at the practical execution of maintenance and safety optimization in an industrial setting.

The content is structured as follows:

Introduction, Objectives, and Research Project Timeline These opening sections defines the scope of the research, highlighting the primary goal: closing the information loop between the engineering department (design) and the repair workshop (operation). It outlines the specific industrial context of the project, detailing the timeline of the international research placement at Compression Service Technology (Florence) and Blackstone Industrial (Canada and Spain) and the collaborative framework established to mitigate geographical and operational disconnects.

Literature Review A comprehensive review of the state-of-the-art regarding compression systems. This chapter analyses existing academic and industrial literature concerning reciprocating and centrifugal compressors together with their auxiliary systems, providing the theoretical foundation for the subsequent technical chapters.

Industrial Plant Acoustic This chapter investigates the macro-level impact of compression machinery. It focuses on the acoustic characterization of the industrial plants, analysing pulsation generation mechanisms, acoustic propagation paths and surge onset causes. This section sets the theoretical baseline for part I and Part II.

Part I: Dynamic System Analysis Moving from the environment to the machine system, Part I addresses the fluid-dynamic response of a centrifugal compression plant.

It focuses on the surge prevention during emergency shutdown by modifications of the plant layout.

Part II: Reciprocating Compressors This section digs into the integrated techniques and tools utilized to analyse pulsations and shaking forces in reciprocating compressors plants, utilizing simulation tools to predict and mitigate vibration risks that could compromise structural integrity.

Part III: Progressive Divider Focusing on auxiliary subsystems, Part III analyses the lubricant supply system, specifically the progressive divider valves. This chapter investigates the fluid mechanics of lubricant distribution under the reciprocating compressor operative conditions, which is vital for preventing seizure and minimizing wear rates.

Part IV: Turbomachinery Material Selection This part addresses the materials science aspect of the research. It evaluates the selection criteria for critical turbomachinery components, balancing mechanical properties with chemical resistance. It discusses the standard requirements related to materials for severe service applications.

Part V: Overhaul and Repair Marking the transition to the field research conducted in Canada and Spain, Part V details the optimization of the Overhaul and Repair (O&R) phase. It describes the implementation of advanced diagnostic protocols (3D scanning, NDT) and the standardization of quality governance through Inspection Test Plans (ITP) and training manuals. This section demonstrates how the repair workshop serves as a critical feedback node for engineering design.

Part VI: Dry Gas Seals The final technical section presents a focused case study on the safety systems of centrifugal compressors. It analyses the venting capacity of the venting piping system connected to Dry Gas Seals. Through a lumped parameter modelling, this part assesses the risk of barrier seal failure under catastrophic conditions and proposes design optimizations to enhance plant safety.

Conclusions The thesis concludes by synthesizing the results obtained across the various domains. It summarizes the value of integrating theoretical design with field-based maintenance data and outlines the future developments for reliable and safe compression system management.

1.03. Industrial compressors

While the industrial market encompasses a broad spectrum of fluid dynamic machinery, four principal compressor typologies dominate large-scale industrial applications: Reciprocating compressors (including specialized hyper-compressors), Centrifugal compressors, Screw compressors, and Axial compressors. The selection envelope for these units is fundamentally delimited by their characteristic performance requirements, typically defined by the volumetric flow rate (Q) versus the isentropic head (H_a). This relationship is formally represented in a compressor selection map (Figure 1), which outlines the specific operational domains where one machine architecture is technically preferred over others.

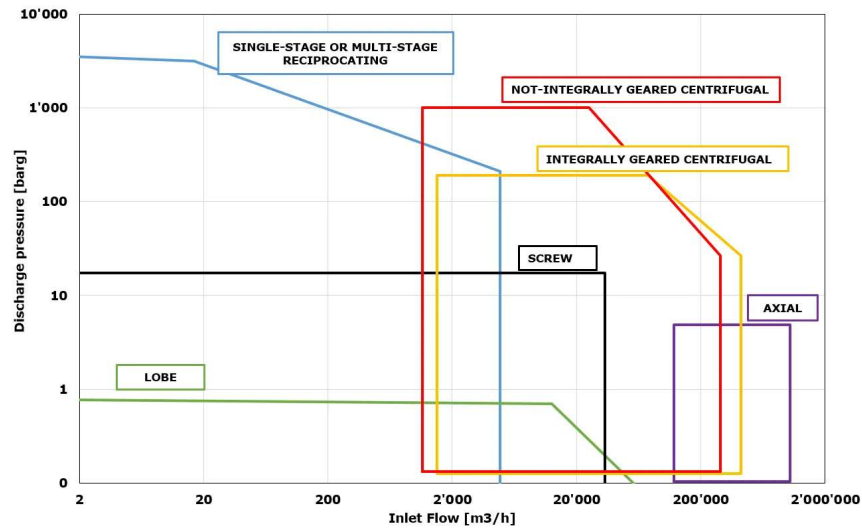


Figure 1: Typical application range per compressor type

Reciprocating compressors, which function on the principle of positive displacement, are inherently suited for processes demanding high-pressure ratios coupled with moderate-to-low volumetric throughput. Conversely, dynamic compressors, including centrifugal and axial machines, are generally selected for high flow rates, offering advantages in flow rate adjustability and handling large gas volumes. However, centrifugal machines exhibit an increased sensitivity to fluctuations in operational parameters, such as inlet pressure and gas molecular weight, compared to their reciprocating counterparts.

This research specifically concentrates its efforts on reciprocating and centrifugal compressors. Their dominance in critical industrial processes and their inherent susceptibility to complex dynamic and transient phenomena justify this dedicated analytical focus. It is essential to underscore that within this critical industrial domain, where machinery manages power fluxes often exceeding the megawatt range, the assets are predominantly custom-engineered. They are designed ad hoc to meet stringent and unique process chemistry, pressure, and flow specifications, thus effectively precluding the deployment of mass-produced, standardized solutions typical of smaller-scale applications. This intrinsic criticality fosters a highly conservative market approach. Given the immense economic implications of unplanned downtime and the catastrophic risks associated with failure, operators and OEMs are historically reluctant to adopt novel techniques or unproven technologies unless they are backed by extensive validation. This risk-averse culture paradoxically amplifies the necessity for rigorous, case-specific analysis. Since "trial and error" is not an option in the field, advanced simulation and robust

validation protocols become the only viable pathway to innovate or troubleshoot within such a constrained environment.

Industrial compressors are frequently subjected to operation under transient conditions, which necessitate focused engineering attention across all phases, including initial design, capacity modification (re-rating), process optimization (upgrading), and systematic diagnostics (troubleshooting). Transient events, such as rapid start-ups, emergency shutdowns, abrupt load excursions, and upstream/downstream network upsets, induce rapid, localized fluctuations in pressure, flow rate, and resultant mechanical loads.

Although frequently of short duration, these dynamic events typically impose the most severe cyclic and dynamic stresses on both the rotating assembly and the associated process piping architecture. A critical engineering gap persists because these complex, short-lived behaviours remain insufficiently characterized and are often only partially mitigated by existing industrial design codes and standards. Addressing this deficiency necessitates the deployment of specialized predictive modelling tools and advanced diagnostic expertise.

Reciprocating and centrifugal compressors present distinct criticalities during these dynamic phases. As positive-displacement units, the inherent intermittent nature of the flow delivered by reciprocating machines generates high-amplitude pulsations (cyclic oscillations in flow and pressure). The resulting acoustic and fluid-dynamic interaction with the complex piping architecture induces significant dynamic forces. These forces can lead to excessive mechanical vibration, accelerate material fatigue, compromise the accuracy of instrumentation, and elevate the systemic risk of pipework failure. Moreover, they affect the machine performances as they influence the compression cycle and they may alter the valve's dynamic, as they often are automatic type, reducing their expected life duration. Effective risk mitigation for the plant necessitates highly specialized pulsation and vibration studies coupled with efficient dampener and acoustic filter design. Dynamic compressors are highly sensitive to the fluid dynamic boundary conditions at the inlet and discharge. Their operational range is fundamentally constrained by two primary limits: the Surge line and the Stonewall limit, correlated directly with flow rate, pressure ratio, and gas molecular weight. The surge limit line specifically represents the stability boundary. Operation proximate to this boundary can induce two critical instability modes: Stall, a localized aerodynamic flow separation leading to performance degradation, and Surge. Surge represents the most critical, system-wide instability, involving axial flow reversal and violent pressure oscillations that can precipitate rapid, catastrophic mechanical failure of rotor-bearing systems and seals. In essence, while transient conditions in reciprocating compressors primarily pose a risk to plant piping and

instrumentation, the equivalent dynamic phenomena in centrifugal machines represent a risk to the integrity of the machine itself.

Transient dynamics affect the auxiliary systems as well, which are vital for machine reliability and safety. For reciprocating compressors, the lubrication system plays a crucial role; accurate oil delivery is mandatory to ensure correct operation, as a deficiency results in increased wear, while excessive delivery contaminates the process gas.

Current market trends, driven by both economic optimization and environmental compliance, strongly demand the minimization of cylinder lubricant consumption. This requirement has led to the widespread adoption of progressive dividers, precise metering devices capable of handling minute flow rates. However, ensuring the correct operation of these systems is challenging due to the extremely low flow rates involved and the fluctuating back-pressure of the cylinder.

For centrifugal compressors, the dry gas seal (DGS) systems are responsible not only for the efficiency of the machine, but also for safety. They guarantee that the process gas does not leak in the environment surrounding the device and avoid cross-contaminations between process gas and lubricant. These seals are highly sensitive to transient-induced forces; consequently, the associated venting piping must be designed to ensure safety even in the event of a seal failure.

While steady state models are widespread for design and efficiency analysis, they are inadequate for prediction of the transient behaviour of the machines. Hence, the necessity for a deeper, physics-based understanding and rigorous predictive modelling of these transient phenomena is therefore of crucial scientific, economic, and safety importance. Furthermore, the sustained exposure to the severe fluid-dynamic stresses induced by these transients (pulsations, surge, flow instabilities) accelerates material fatigue and wear, mandating a proactive approach to Lifecycle Management. This includes procedures such as re-rating (modifying the unit for new operating conditions) and overhauling (detailed repair, restoration, and dimensional verification). Overhauling is particularly critical as it involves ensuring that repaired or replaced components meet or exceed the original stringent design specifications. The selection of appropriate materials for replacement parts is not merely a logistical step but a critical engineering function that directly mitigates the damage accumulated during dynamic operation. This involves meticulous evaluation of material properties, heat treatments, and failure modes to ensure the restored component can withstand the original operating envelope.

To navigate these complexities and ensure a uniform level of safety and reliability across the industry, engineers rely on a rigorous normative framework. These standards provide

the essential guidelines for design, testing, and safety validation, forming the backbone of the methodologies explored in the following section.

1.04. Industry standards

API Standards

The American Petroleum Institute (API), established in 1919, remains the most influential trade association for the energy sector. While its origins are deeply rooted in the oil and gas industry, API standards have evolved into a global benchmark for engineering quality, extending their relevance far beyond fossil fuel applications. In the context of the current energy transition promoted by the NRRP, these standards continue to be strictly required and applied even in green energy projects and sustainable industrial processes (such as hydrogen compression, carbon capture and storage, ...). In these novel applications, the stringent safety margins imposed by API standards become even more critical due to the hazardous nature and high diffusivity of the process fluids. This cross-sectoral adoption ensures maximum safety and performance reliability regardless of the application field. Specifically, for the commercialization of centrifugal and reciprocating compressors, compliance with API standards is a mandatory prerequisite. Consequently, this study establishes its normative framework upon four key documents: the API 617 standard [1] for centrifugal machines, the API 618 standard [2] and API 688 recommended practice [3] for reciprocating compressors, and the API 692 standard [4] for dry gas sealing systems.

API 617 standard

Regarding centrifugal machinery, the API 617 standard (Axial and Centrifugal Compressors and Expander-compressors) represents the primary reference for design, testing, and mechanical integrity. Beyond structural requirements, this regulation places a critical emphasis on operational safety and the prevention of aerodynamic instability, specifically the surge phenomenon. While steady-state calculations are sufficient for preliminary sizing purposes, they cannot adequately predict the system's behaviour during rapid transients. Therefore, to validate the effectiveness of the protective systems, the standard emphasizes the necessity of performing dynamic simulations.

In this context, Annex H of the standard serves as a guideline for the analyst. Rather than prescribing specific software capabilities or computational algorithms, this annex provides a comprehensive list of typical operating scenarios that must be investigated, such as Emergency Shutdowns (ESD), start-up from settle-out conditions, or fast valve closures, along with the specific input data required to build a reliable model. By simulating the time-dependent interaction between gas volumes, piping acoustics, and the anti-surge

control loop response based on these inputs, the analysis aims to verify that the compressor remains within its safe operating envelope under all the prescribed conditions.

API 618 standard

The API 618 standard provides comprehensive guidelines for reciprocating compressor plants. It defines rigorous design requirements for both the machinery and the associated piping systems, covering geometric, physical, and operative parameters. Specifically, clause 7.9 regulates pulsation and vibration control. To this end, the standard outlines three distinct design approaches, categorized by the compressor's power, size, and discharge pressure:

- Design Approach 1: The most fundamental level, it relies on empirical or proprietary techniques to size pulsation suppression devices without requiring formal acoustic simulation.
- Design Approach 2: This level necessitates a formal acoustical analysis alongside a study of piping restraints. The acoustic simulation is mandatory to verify preliminary design parameters and ensure compliance with standard limits. While it allows for the design of suppression devices without full knowledge of the piping layout, this often leads to a conservative and oversized system. The piping restraint analysis utilizes tables and simplified calculations to determine maximum clamp spacing and the natural frequencies.
- Design Approach 3: The most advanced level, it supplements the second approach with a comprehensive mechanical analysis of the compressor and piping system, potentially including a forced response study. This enables a detailed evaluation of system stresses and dynamic mechanical behaviour.

Given the complexity of the systems analysed within this doctoral work, the focus is primarily placed on Design Approach 2 and 3. These approaches are the only ones ensuring a rigorous prediction of the fluid-structure interaction, which is essential to prevent fatigue failures in high-performance piping systems. Depending on the selected approach, the standard prescribes strict limits for pressure pulsations, pressure drops, shaking forces, and vibrations. Crucially, the standard mandates the determination of pulsations at the compressor cylinder flanges and at the line side nozzle of the pulsation suppression device; consequently, these specific locations are the primary focus of the analyses within this work.

API 688 Recommended practice

The API 688 Recommended Practice serves as the technical supplement to Clause 7.9 of the API 618 standard. This document effectively bridges the gap between the normative limits

set by API 618 and the physics of fluid-structure interaction. It expands upon the regulatory requirements by providing the essential theoretical background, addressing critical design challenges, and establishing methodological guidelines. Structurally, the document is organized into two primary sections.

The first section addresses the fundamental principles of pulsation and vibration control. It details the acoustics of duct systems and mechanical vibrations, analysing how various operating parameters impact the compression cycle, machine performance, and valve dynamics. Furthermore, this section defines three distinct Design Philosophies available to the engineer:

- Acoustic Control: Focuses on applying acoustic techniques to attenuate pressure pulsations and, consequently, shaking forces.
- Shaking Force Control: Utilizes specific shaking force limits as the primary constraint for sizing pulsation suppression devices.
- Vibration Control: Employs forced mechanical response analysis to predict and manage the comprehensive acoustic and mechanical behaviour of the system.

The second section is specifically dedicated to reciprocating compressor systems. It provides a detailed explanation of the three Design Approaches introduced in API 618, illustrating the procedural steps through flowcharts. These diagrams are essential for correctly executing the compliance verification process.

API 692 Standard

Complementing the machinery standards, the API 692 standard places a significant emphasis on operational safety and the management of critical failure scenarios. Section 6 of Part 1 outlines the general design requirements, mandating an approach that ensures system integrity even under catastrophic conditions. A fundamental aspect of this safety philosophy is the requirement to protect the machine's lubrication system. The standard stipulates that in the event of a dry gas seal failure the separation seal must be designed to maintain sealing integrity. Its primary function during a failure is to strictly limit the amount of process gas entering the bearing housing, thereby preventing lubrication system contamination and hazardous releases into the operating environment. To ensure this safety requirement is met, the standard mandates a comprehensive identification of potential failure modes within the entire seal system. The design process must not be limited to standard operations but must explicitly address combinations of failures. Specifically, the standard highlights the necessity to evaluate scenarios including the simultaneous failure of both primary and secondary seals.

The practical validation of these safety requirements is achieved through the Vent Study, as defined in the standard's annexes. This study is the critical analytical tool used to establish the venting system's capacity during the identified seal failures. By analysing the pressure and flow profiles during these rupture scenarios, the Vent Study ensures that the system is designed to minimize potential damage and uncontrolled leakage to the atmosphere. It verifies that the piping and component design can accommodate the increase in flow and pressure, preventing the compromise of the separation barrier even during the most critical failure combinations.

However, while the standard defines the acceptance criteria and the scope of the analysis, it does not prescribe the specific numerical methods or computational models required to accurately capture the transient gas dynamics. Consequently, a significant portion of this research is dedicated to developing and validating a modelling approach compliant with Annex A, capable of handling the complex compressibility effects during critical seal failures.

1.05. Context Synthesis and Research Motivation

The industrial landscape outlined in this introductory chapter, characterized by strict safety mandates and a conservative technological approach, defines the perimeter of the present research. The analysis highlights a critical dichotomy: while API standards rigorously define safety targets and acceptance criteria, they do not prescribe the specific advanced numerical methods required to predict complex transient behaviours. This gap between normative requirements and available predictive capabilities constitutes the primary driver of this doctoral work. Consequently, the following chapter will articulate the project's multifaceted objectives, encompassing not only the scientific advancements aimed at bridging this technical divide but also the goals related to professional development and the fostering of strategic industrial cooperation.

2. Objectives

2.01. Overall Aim of the Thesis

The fundamental aim of this research is to advance both fundamental and applied knowledge regarding transient fluid dynamics in industrial compressors. The specific topics addressed in this thesis were selected in close collaboration with the partner companies, CST and Blackstone Industrial. The selection criteria focused on areas presenting either significant scientific gaps, thus offering high academic value, or technological disparities that currently limit the competitiveness of the involved companies.

By developing detailed simulation models and incorporating real-world field data, this work aims to deliver predictive tools and engineering insights directly applicable to design, operation, and diagnostic activities. Consequently, the research is designed not only to contribute to the scientific community but also to solve practical limitations in current industrial workflows.

2.02. Scientific objectives

To ensure a comprehensive analysis, the scientific objectives have been categorized into three main areas: fluid dynamic transients, auxiliary systems, and lifecycle & materials. Firstly, a primary focus was placed on flow instabilities in centrifugal compressors, leading to the development of simulation tools capable of analysing these plants under transient conditions. Specifically, the objective was to quantify the onset of surge during transient scenarios. A second major goal is the study of pulsation phenomena in reciprocating compressor plants, prioritizing the identification of key parameters that drive acoustic resonance and vibration. To evaluate system-level behaviour, a comprehensive dynamic model was developed to simulate the interactions among compressors, piping, heat exchangers, and other process components. Additionally, a blowdown analysis was conducted to investigate the rapid depressurization dynamics within these systems. Then, the research aimed to evaluate the critical subsystems that support compressor operation. A significant objective is the evaluation of progressive divider-type lubricant supply systems used in reciprocating compressors. The aim is to assess their performance under fluctuating load conditions to identify critical operating states. Furthermore, a focused investigation was performed on dry gas seals systems. In this context, the objective was to analyse the performance of the associated piping and components, specifically in the event of failure occurrence, to ensure safety and containment effectiveness. Lastly, to support a broader understanding of machine integrity and operational availability, the study addressed material selection best practices for turbomachinery

components. The research also focused on the overhaul and repair of reciprocating compressors, establishing criteria to optimize maintenance interventions. This includes defining engineering guidelines for component life extension and evaluating the trade-offs between repair and replacement.

2.03. Professional Development Objectives

Beyond technical research, this PhD is structured to reinforce professional competence in fluid machinery engineering through targeted developmental objectives.

A primary goal is to cultivate the ability to structure and manage the research process, breaking it down into defined phases and aligning it with a realistic project timeline.

Another crucial objective is the application of advanced research methodologies within an industrial context, bridging the gap between academic investigation and real-world engineering practice. The research activities also aim to build a comprehensive understanding of the simulation, measurement, and calculation tools commonly used in the sector. Finally, gaining exposure to international industrial practices and workflows is a key aim of this PhD. This experience contributes to a broader understanding of how fluid machinery systems are approached across different cultural and regulatory environments.

2.04. Industrial Cooperation

Close cooperation with partner companies is essential to ensure that the scientific outcomes of this research address real-world engineering challenges.

A central objective is optimizing the interaction between the engineering department (CST) and the workshop floor (Blackstone Industrial). This collaboration faces challenges not only due to differing operational approaches but also because of geographical distance and time zone differences. The research aims to streamline this workflow, ensuring that engineering specifications are effectively translated into practical execution and that field feedback is correctly interpreted by design engineers.

The work seeks to provide a rigorous technical-scientific framework to competencies primarily established through operational practice. Efficiency improvements were targeted specifically within overhaul, maintenance, and repair contexts. This involved integrating specialized modern techniques, such as 3D scanning and Positive Material Identification (PMI), and traditional metrology methods, including mechanical measurements, Prussian blue checks, and dye penetrant inspections. The objective is to link these diagnostic results to robust engineering decisions, specifically guiding strategies for component replacement, rerating, and upgrading. By applying the scientific method to these industrial processes, the work creates a bridge between empirical know-how and formal engineering analysis.

2.05. Goals summary

To summarize, the main scientific objective of this thesis is to investigate the fluid dynamic behaviour and operational integrity of industrial compressors. The specific goals are grouped as follows:

- Fluid Dynamic Transients
 - Investigating flow instabilities and surge onset in centrifugal compressor plants.
 - Analysing pulsations and acoustic resonance in reciprocating compressor plants.
 - Performing blowdown analysis for rapid depressurization.
- Ancillary Systems
 - Analysing lubricant supply systems for reciprocating compressors under fluctuating loads.
 - Evaluating dry gas seal piping performance during failure scenarios.
- Lifecycle and Materials
 - Defining best practices for material selection in turbomachinery.
 - Optimizing overhaul and repair strategies for reciprocating compressors.
- Professional & Industrial Goals
 - Optimizing the interface between engineering teams and workshop operations (CST & Blackstone Industrial).
 - Combining advanced diagnostic techniques (e.g., 3D scanning) with traditional methods to inform rerate and upgrade decisions.
 - Applying advanced research methodologies to solve competitiveness-limiting technological gaps.

3. Research Project Timeline

The doctoral research project spanned a three-year period, systematically designed to progress from foundational theoretical development through specialized industrial application to international empirical validation and, ultimately, the final academic synthesis.

3.01. First Year: Research Initiation

The initial year was fully dedicated to research activities undertaken at the home university, with the primary objective of establishing the robust theoretical framework and consolidating the initial methodologies essential for the subsequent phases.

The first semester focused on preparatory activities that yielded a critical synthesis of the state-of-the-art. The core achievement was the comprehensive literature review targeting both reciprocating and centrifugal compressors, with particular emphasis on characterizing their complex transient behaviours. This theoretical grounding was immediately complemented by foundational empirical investigations in the laboratory, specifically targeting the operational dynamics of progressive divider lubrication systems and cavitation phenomena within poppet valves. A key result of this initial work was the development of a complementary numerical modelling framework to interpret the experimental data related to the lubrication study. The semester culminated with participation in the EFRC (European Forum for Reciprocating Compressors) workshop, which provided essential early-stage exposure to the industrial landscape through guided facility visits.

Building upon this foundational knowledge, the second semester focused on the critical development and consolidation of the initial findings. The research attained external validation through the successful dissemination of initial results at the ATI conference, through a proceeding. A technical visit to SIAD facilities further enhanced the practical understanding of operational systems. A significant expansion of scope was realized when this phase culminated in the initiation of a distinct and major research stream: the formulation of a detailed centrifugal compressor plant model. This specialized model was engineered specifically to conduct comprehensive stability analyses and derive critical insights for the design of effective surge prevention systems.

3.02. Second Year: Industrial Immersion

The subsequent year marked a strategic shift towards comprehensive industrial immersion, with the entire period conducted at the headquarters of Compression Service

Technology S.r.l. (CST). This phase was indispensable for translating theoretical concepts into verifiable industrial practices.

Transitioning from theoretical exploration, the third semester was dedicated to the systematic assimilation of industry-mandated specifications. Key efforts involved the intensive study of normative frameworks (such as API and ASME) and the intricate principles of piping design. This acquired expertise was fundamental, serving to significantly refine and enhance the fidelity of the plant modelling initiatives established previously. Close interaction with the company facilitated the acquisition of comprehensive industrial insights into how detailed pulsation studies are executed and how their mitigation strategies are implemented within complex systems.

Parallel investigations explored the acoustic behaviour of centrifugal compressor installations, providing critical understanding of fast transient dynamics. Furthermore, a new collaboration with Dropsa was established to advance the empirical studies on lubrication systems. This period was academically capped by attending the Global Power and Propulsion Society (GPPS) seminar, which provided crucial networking opportunities with leading domain experts.

Deepening the industrial focus, the fourth semester was dedicated to intensive characterization of critical ancillary systems and operational variables. Attention was directed towards the detailed functional analysis of dry gas seals; specifically, the study focused on evaluating the performance of associated piping networks during critical failure scenarios. Concurrently, blowdown analyses were performed to investigate the rapid depressurization dynamics, ensuring system integrity under emergency conditions. Simultaneously, a crucial research outcome was achieved through the systematic investigation of the effects of composition variations on centrifugal compressor performance, leading to a detailed understanding of stability and transient response under varied operating conditions. This period strategically culminated with extensive organizational and logistical preparation for the forthcoming international mobility, a necessary step to ensure the seamless continuation of the ambitious research objectives in a new, specialized environment.

3.03. Third Year: International Specialization and Finalization

The final year inaugurated the international specialization phase while simultaneously focusing on the crucial task of synthesizing all research findings into the doctoral thesis. The fifth semester began with complementary analytical activities at the university, where analyses were conducted through 3D-CFD simulations on orifices and valves, alongside the task of supervising master's thesis students. Subsequently, research transitioned to the

Canadian facilities of Blackstone Industrial. This stay represented a crucial methodological expansion, shifting the focus towards practical maintenance, overhauling, and repair. Key achievements included the practical implementation of techniques such as 3D scanning and manual measurements, alongside the development and execution of reverse engineering protocols specifically applied to reciprocating pump valves. Theoretically, this placement facilitated an in-depth study of specialized procedures, including the critical elements of material selection, heat treatment, re-rating, and start-up protocols for centrifugal compressors. A comprehensive understanding of both the analytical and practical dimensions of turbomachinery lifecycle management was successfully established.

The sixth and final semester was primarily dedicated to the synthesis and consolidation of all the main research threads into the final doctoral dissertation document and through the publication of papers in the relevant journals. To ensure the completeness of the doctoral analysis, two highly focused, short-term research deployments were executed. A two-week visit to the CST headquarters allowed for a final, specialized investigation into centrifugal compressor geometry selection. This was immediately followed by a productive one-month period at the Spanish repair shop of Blackstone Industrial. This final stage provided indispensable, direct exposure to practical overhaul procedures applied to a centrifugal pump, including specific diagnostic techniques such as the Prussian blue test and penetrant liquid inspection. This final practical validation successfully linked the theoretical modelling and industrial application phases with tangible, on-site repair diagnostics. The last three months were dedicated to finalizing the manuscript and concluding the doctoral research.

4. Literature review

To establish the state of the art and provide the theoretical foundation for this thesis, a comprehensive literature review was conducted across the main areas relevant to the study. The review addresses the following key topics:

- Centrifugal compressor dynamics and associated flow instabilities.
- Dry Gas Seal (DGS) systems in centrifugal applications.
- Reciprocating compressors, pulsation phenomena, and fluid-structure interaction.
- Progressive divider lubrication technology for reciprocating compressors.
- Material selection and surface treatments for turbomachinery

4.01. Centrifugal Compressors

Centrifugal compressors are dynamic machines inherently subject to aerodynamic instabilities, most notably surge and rotating stall. Ensuring stability during both continuous and emergency operations is a primary responsibility of the designer. Specifically, these phenomena can occur during transient operations, such as start-up, emergency shut down (ESD), partial loads and parameters variations (e.g., gas composition, intake and discharge pressure fluctuations).

Foundational textbooks [5-9] provide the essential theoretical explanations of these phenomena. Sparks [10] details the critical interaction between the compressors and the piping system, emphasizing the acoustic dynamics that govern the machine's behaviour. Greitzer and Moore [11, 12] introduced a mathematical model capable of describing the transient response of compression systems; this work remains the cornerstone for modern computer-based simulations of plant behaviour.

Extending the research beyond these fundamental texts, the literature covers a vast array of specific aspects. To provide a structured overview, the available documents have been categorized into four key areas:

- Phenomenon modelling and simulation
- Influence of design and plant parameters
- Anti-surge solutions and control strategies
- Design guidelines and sizing criteria

The first category encompasses studies focused on mathematical modelling, root causes investigation, and prediction of onset conditions. Powers et al. [13] developed a quintic characteristic curve (flow-head) that enables the differentiation between mild surge and deep surge. Toyama et al. [14] investigated the surge onset under a flow perturbations during conditions otherwise considered operationally stable. To gain high-fidelity insights, Sundstrom et al. [15] utilized Large Eddy Simulation (LES) to study the internal gas flow

behaviour, clarifying how surge develops across different flow-head conditions. Similarly, Fink et al. [16] provided valuable experimental data by inducing both stall and surge in a turbocharger compressor to analyse the transition dynamics. Focusing on system-level dynamics, Macdougall et al. [17] developed a model capable of simulating compressor behaviour during transient conditions. Similarly, Morini et al. [18] proposed a 1D simulation model specifically for predicting surge occurrence. From a mechanical integrity perspective, Klaus et al. [19, 20] analysed the stresses acting on the compressor train during surge. They develop a lumped parameter model to determine the transient forces generated, which is critical for shaft and bearing design. Finally, regarding damage assessment, Munari et al. [21] introduced a coefficient to quantify surge severity. This metric provides operators with a criterion to determine whether maintenance or overhaul is required following a surge event.

The second category includes studies analysing how specific plant layouts and compressor geometries influence the machine's stability, either facilitating or preventing surge onset. Research by Sun et al. [22] highlights how inlet piping geometry, specifically bent pipes versus straight sections, affects transient phenomena. Tamaki [23] further investigated the piping system's influence on the surge line using a lumped parameters model. Regarding machine geometry, Danlos et al. [24] demonstrated how pre-swirl devices affects the machine flow-head characteristic, effectively shifting the surge line, while Zou et al. [25] analysed the impact of the diffuser vanes on both rotating stall onset and surge line position. Of particular relevance to complex plants, Brun et al. [26] studied the interaction between different compressor types, specifically analysing how the pulsations of a reciprocating compressor affects the stability of a centrifugal compressor sharing the same process network.

The third category includes studies regarding anti-surge solutions and control strategies. The most common solution in industrial applications is a dynamically controlled recirculation valve (commonly referred to as a bypass or anti-surge valve). This approach is favoured for its simplicity, reliability, and effectiveness.

While standard control strategies are widespread [27-29], alternative solutions have been explored in the literature to enhance performance or address specific limitations. Pinsley et al. [30] developed a suppression system for small perturbations based on a close-coupled valve (CCV) placed on the plenum outlet, designed to extend the operational range for low-flow rates. Gravdahl et al. [31] expanded on Pinsley's work by integrating a speed control on the compressor drive. Other active control methods include the use of Active Magnetic Bearings (AMB) to modulate tip clearance, as proposed by Yoon et al. [32] and the optimization of a discharge vessel piston system by Uddin and Gravdahl [33].

The final category collects works providing practical design techniques, primarily focusing on bypass valve systems. Almasi [34] outlines critical parameters and considerations for sizing and design of effective bypass systems. To support decision-making, Botros defines a dimensionless parameter, the inertia number [35], to determine whether a hot gas bypass is required to ensure safe operations during rapid transients [36]. Building upon this, Yoon et al. [37] introduced the surge number, a new dimensionless parameter that improves prediction accuracy in cases where the inertia number yielded inconclusive results regarding the necessity of a hot bypass.

4.02. Dry gas seals

Dry gas seals (DGS) are non-contacting mechanical seals, widely adopted in modern centrifugal compressors as a superior alternative to (or in combination with) labyrinth, brush, or honeycomb seals. Their primary function is to minimize process gas leakage from the casing along the rotor shaft. Comprehensive descriptions of their operating physics, fluid dynamic principles, and standard industrial configurations are detailed in foundational textbooks [38].

Current research literature primarily focuses on two critical aspects of DGS technology:

- Leakage gas recovery and emission mitigation
- Component reliability and dynamic stability

The first area addresses the industry-wide imperative to lower emissions, a necessity driven by increasingly stringent environmental regulations, particularly in natural gas applications. Research in this field aims to optimize systems for capturing and repurposing seal leakage rather than venting it to the atmosphere. Suomilammi [39], Betti et al. [40] and Conforti et al. [41] have extensively studied and compared different engineering solutions for natural gas recovery, evaluating their efficiency and economic feasibility.

The second research stream responds to the high sensitivity of these components to operating conditions. Given the critical role of DGSs, their failure often leads to immediate machine shutdowns. Day and Allison [42] conducted a statistical analysis of field data to identify the primary failure modes and root causes leading to seal damage. From a theoretical perspective, Zhang et al. [43] performed a comprehensive review of the research progress regarding the dynamic stability of DGS systems. The analysis of transient behaviour is particularly crucial in this context: predicting the fluid film stiffness and damping is essential to ensure proper seal lift-off (face separation), thereby avoiding

catastrophic friction between rotating and stationary parts during start-ups, coast-downs, or process upsets.

While the intrinsic behaviour of the seal faces is well-documented, a significant divergence exists between academic research and industrial necessity regarding the auxiliary support systems. The recent enforcement of API 692 standards has introduced mandatory requirements for the Vent Study, necessitating a rigorous verification of the piping network's capacity to handle rapid depressurization during seal failures. However, standardized computational tools capable of accurately modelling these compressible flow transients – accounting for choke flow and gas expansion – are not widely established in the engineering practice. This represents a critical technological gap: while the theoretical governing equations are known, the industry lacks robust, applied methodologies to perform these safety verifications efficiently. Addressing this disparity by developing specific calculation tools is a primary objective of the industrial partner (CST) and a key focus of this thesis.

4.03. Reciprocating compressors

Reciprocating compressors are positive displacement machines inherently characterized by unsteady flow delivery. While in internal combustion engines this flow irregularity can be exploited to achieve supercharging effects [44, 45], in industrial gas compression applications, the primary design objective is to mitigate the resultant phenomena. Due to their cyclic operating principle, these machines generate high-amplitude pressure pulsations which propagate through the piping network. If uncontrolled, these pulsations induce shaking forces, high noise levels, and flow-induced vibrations (FIV), potentially leading to fatigue failures.

Comprehensive theoretical treatments of duct acoustics are provided in foundational textbooks by Munjal [46] and others [47, 48]. Specifically focusing on compressor applications, numerous studies [49-53] expand upon the guidelines set by the API standards [1-4] detailing the mechanisms of pulsation generation and propagation.

To mitigate these issues, Pulsation Suppression Devices (PSDs) are employed to attenuate pressure wave amplitude within the piping system. Warzyńska [54] proposes a systematic classification of these devices, summarized in Figure 2.1, which categorizes the available design choices into three principal typologies: Dissipative, Reactive, and Hybrid dampers.

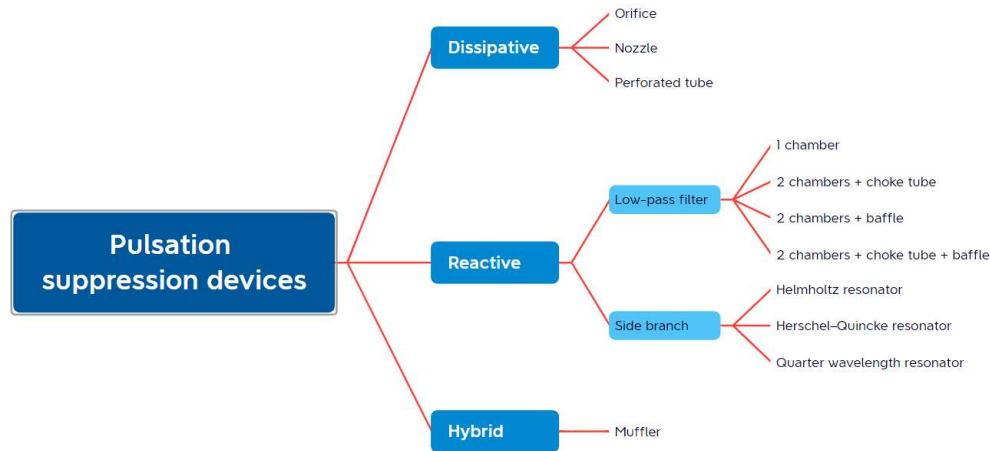


Figure 2: Damper classification

Dissipative devices operate as resistive components that induce a localized pressure drop, thereby converting acoustic energy into heat and turbulence. The simplest forms include mono-hole orifices and nozzles. These elements dampen pulsations through three concurrent mechanisms: limiting the peak flow rate, generating turbulence, and introducing energy dissipation. More complex configurations, such as multi-hole orifices and perforated tubes, enhance performance by introducing additional wave-scattering effects.

The literature on orifices is extensive due to their widespread industrial adoption. Notably, Bagade et al. [55] analysed various multi-hole configurations, while Adair [56] developed a variable orifice design capable of optimization across different compressor speeds. Regarding nozzles, Mlynarczyk [57-62] and Cyklis [63-67] conducted in-depth studies, including the design of a tuned mass damper nozzle [62]. Perforated tubes, widely utilized in internal combustion engine exhaust mufflers [68]; are frequently discussed in general acoustic textbooks [44-46] or analysed as components within hybrid systems. Reactive dampers function by exploiting acoustic compliance (volume) and inertance (tube) to create an impedance mismatch that reflects wave energy, thereby filtering specific frequencies without significant pressure loss. These components are generally arranged in two configurations: side-branch resonators or low-pass filters (in-line)

The side-branch resonators include Helmholtz resonators, Quarter-Wave tubes, and Herschel-Quincke resonators. Selamet et al. [69-76] provided extensive characterization of Helmholtz resonators. Specialized applications include Mattarelli's [77] intake system for Formula Student engines and Paiva's [78] analysis of resonator trees. Regarding Quarter-Wave resonators, Lin [79] demonstrated their attenuation capacity, while Cambonie [80] investigated curved geometries. The characteristics of Herschel-Quincke resonators have been explored in multiple works by Poirier [81-83].

The low-pass filters are constructed by combining volumes, baffles, and choke tubes. The simplest configuration is the single-chamber surge volume (empty vessel). In more demanding applications, the volume-choke-volume configuration is preferred. This can be realized as two separate vessels or a single vessel internally divided by a baffle, with chambers connected by a choke tube. Warzyńska [53, 54, 84] compared the behavior of various low-pass filter configurations. Additionally, Sadamoto [85] investigated the acoustic properties of slit-like short expansion devices.

Lastly, hybrid dampers combine the principles of both dissipative (flow resistance/perforation) and reactive (geometric volume) elements to achieve broadband attenuation. While heavily utilized in the automotive industry for exhaust noise reduction, their application is also relevant in industrial scenarios where broad frequency ranges must be targeted. Denia [86-92] and Selamet [93, 94] have extensively studied the transmission loss characteristics of these combined systems.

4.04. Progressive dividers

Precise cylinder lubrication is a critical operational requirement for reciprocating compressors, demanding a strict balance between component integrity and process purity. On one hand, insufficient lubrication leads to accelerated wear of rings and packings; on the other, over-lubrication results in excessive oil consumption and the contamination of the process gas, which is unacceptable in sensitive applications (e.g., hydrogen or food-grade processes).

Current industrial standards employ two primary architectures to achieve this delivery. The first approach, known as the Pump-to-Point (PTP) system, relies on a bank of independent volumetric pumps, each dedicated to a specific injection point. While this allows for individual flow regulation, it typically results in complex and bulky installations. The alternative solution is the Progressive Divider (PD) system, which utilizes a single centralized pump feeding a series of volumetric flow divider blocks. The fluid is distributed to the various points through the sequential movement of internal metering pistons. The PD system offers significant advantages in terms of compactness and centralization. However, its operation is inherently coupled: the motion of one piston hydraulically triggers the next, making the system sensitive to fluid properties and back-pressure fluctuations.

Despite their widespread industrial adoption, scientific literature regarding the dynamic behaviour of progressive dividers is remarkably scarce. Most available knowledge is confined to commercial technical manuals [95, 96], which provide steady-state sizing guidelines but lack dynamic insight.

Among the limited academic contributions, Maffei et al. [97] described the development of a hybrid lubrication system for a hydrogen reciprocating compressor, integrating a PD system with pre-existing PTP infrastructure. In a parallel effort, Sun et al. [98] investigated an oil mist PD system by developing a simulation model within the Simcenter Amesim environment to predict flow distribution. Furthermore, Veprek [99] applied genetic algorithms to assist the design process, focusing on optimizing the module configuration to meet specific lubrication requirements.

Consequently, a detailed understanding of the transient behaviour of these devices under pulsating back-pressure remains a largely unexplored area in the open literature.

4.05. Turbomachinery material selection

Material selection constitutes a fundamental pillar of the design process, directly influencing the machinery's ability to withstand the cyclic stresses induced by the transient fluid dynamic phenomena discussed in previous sections. Consequently, defining robust selection criteria is essential not only for the initial design but also for ensuring operational integrity throughout the machine's lifecycle.

Foundational guidelines for manufacturing and metallurgy were established by Cameron [100, 101], whose work provided critical considerations on impeller material selection.

These concepts were later expanded by Dowson et al. [102], who offered a comprehensive overview of material and process selection criteria for both centrifugal compressors and steam turbines, subsequently extending their analysis to complex industrial turbine applications [103]. Complementing these design-oriented studies, Boddenberg [104] evaluated the correlation between available manufacturing technologies and specific impeller typologies.

Crucially, the selection process extends beyond initial design to the lifecycle management of the asset. Clayton et al. [105] applied these selection principles specifically to impeller repair strategies, addressing the challenge of restoring mechanical properties in damaged components. This is closely linked to the work of Radgolchin and Anbarsooz [106], who analysed the fatigue behaviour of centrifugal compressor impellers, a critical failure mode driven by the transient loads described earlier in this thesis.

Regarding specific alloy treatments and microstructural properties, the literature offers detailed investigations into the materials commonly employed in critical components. Sanij et al. [107] performed comparative studies on double versus single quenching and tempering protocols for AISI 4140 steel, a standard low-alloy steel widely used for compressor shafts and impellers. For more extreme operating environments, particularly in gas turbines where creep and oxidation resistance are paramount, the focus shifts to

superalloys. Salwan et al. [108] provided a comparison of casting techniques for turbine graded materials, distinguishing between conventional casting, directionally solidified, and single-crystal manufacturing. Konter et al. [109] and Benini et al. [110] produced extensive overviews of material and coating selection, highlighting recent innovations in the sector. Complementing this broad perspective, Tejedor [111] specifically reviewed different material technologies tailored for high-temperature applications. On the metallurgical level, Bhadeshia [112] effectively explained the composition and crystalline microstructure of Nickel-based superalloys. Similarly, Sato et al. [113] and Coutsouradis et al. [114] focused on the composition and properties of Cobalt-based superalloys. Finally, Akca et al. [115] investigated the property variations of Inconel 718 (IN718) depending on the specific heat treatment applied, underscoring the critical link between manufacturing processes and final material performance.

5. Industrial plants acoustic

Before addressing the specific methodologies employed in the subsequent parts of this work, a semantic and physical clarification is necessary regarding the use of the term "acoustics" within the context of industrial compression systems. In strict physical terms, classical acoustics refers to the propagation of infinitesimal pressure perturbations ($p' \ll p_0$), characterized by small amplitudes where wave behaviour remains linear, isentropic, and non-intrusive to the mean flow.

This definition typically aligns well with the steady-state operation of turbomachinery. In centrifugal compressors, strictly "acoustic" phenomena are primarily associated with aerodynamic noise and high-frequency harmonics (such as the Blade Pass Frequency), while high-energy pressure fluctuations are anomalous events restricted to aerodynamic instabilities like rotating stall or surge.

Conversely, in the specific domain of reciprocating compressors, the intermittent volumetric displacement inherently generates finite-amplitude pressure waves as part of normal operation. Unlike sound waves, these pulsations often reach magnitudes that are significant fractions of the static line pressure (typically ranging from 1% to over 10% peak-to-peak). Although these high-intensity waves theoretically introduce non-linearities that deviate from classical acoustics, the industry nomenclature (API 618 standard) retains the term "acoustical simulation" to describe the study of plane wave propagation and resonance in the piping network. In this broader engineering acceptance, encompassing both the macroscopic transients of centrifugal surge and the cyclic pulsations of reciprocating machinery, extending the concept beyond the limit of small perturbations.

5.01. Centrifugal compressors fast transient response

A fundamental distinction must be drawn between the quasi-steady response of a compression system and its behaviour during rapid transients. While slow modulations in operating conditions allow the system to track the steady-state system resistance curve, fast transients, such as those induced by rapid valve closures, emergency shutdown (ESD), or high-frequency pulsations, decouple the compressor from this steady-state path. In these dynamic scenarios, the instantaneous relationship between pressure and flow is governed by the acoustic impedance of the attached piping. Figure 3 shows a comparison between the slow and fast transients on the steady map.

As originally postulated by Sparks [10], the interaction between a centrifugal compressor and its piping during a fast transient can be visualized by superimposing an impedance line onto the steady-state compressor map.

In steady-state conditions, the discharge pressure P_D is a function of the piping friction and follows a square law relative to flow velocity v :

$$P_D = R_x \Delta v^2 \quad (1)$$

where R_x is the flow resistance coefficient.

However, during a rapid perturbation (where the time scale of the disturbance is comparable to the acoustic wave propagation time), the change in pressure (Δp) and the change in flow velocity (Δv) are linearly related by the characteristic acoustic impedance of the duct:

$$\Delta p = (\rho c) \Delta v \quad (2)$$

or, more generally for a specific piping geometry:

$$\Delta p = Z_{ac} \Delta v \quad (3)$$

where:

- ρ is the gas density,
- c is the acoustic velocity,
- Z_{ac} is the piping network impedance.

This implies that the operating point does not slide along the parabolic system resistance curve during a fast transient. Instead, it moves along a trajectory defined by the slope Z_{ac} passing through the initial steady-state operating point.

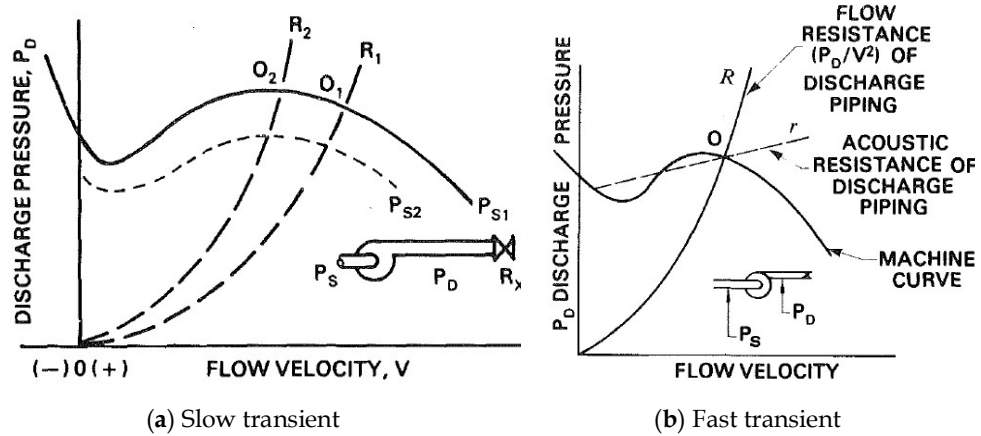


Figure 3: Transient response on steady state map [10]

This divergence is critical for stability analysis. As illustrated in Figure 4, a high-impedance discharge line (represented by steep acoustic resistance line) implies that a small reduction in flow velocity results in a larger variation in discharge pressure.

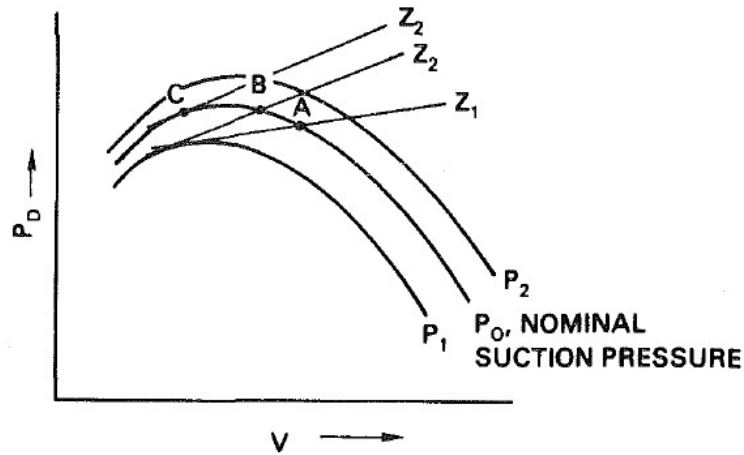
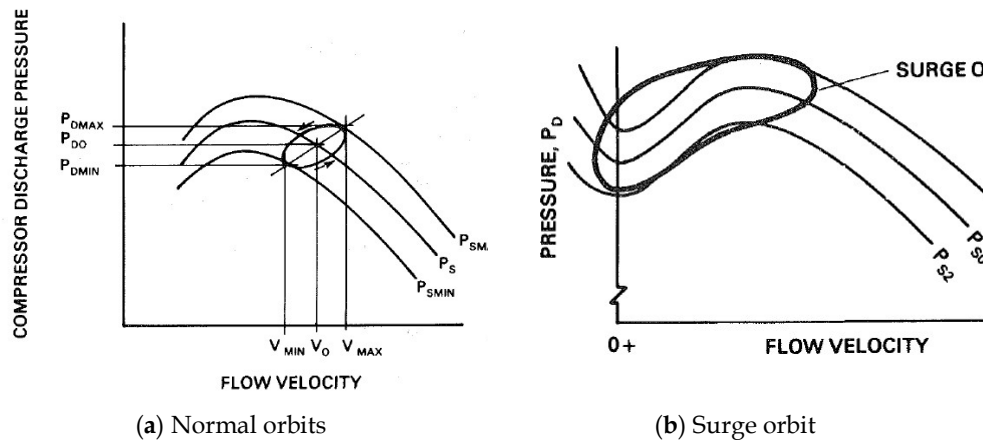


Figure 4: impedance lines variations [10]

If the piping is purely resistive (assuming infinite pipe length), the operating point oscillates along a straight line with slope Z_{ac} . In real industrial systems, as noted by Sparks and further analysed by Brun et al., the piping contains reactive elements (acoustic mass and stiffness). This introduces a phase shift between pressure and flow, causing the transient trajectory to open into an elliptical orbit rather than a straight line as represented in Figure 5a. If the orbits extend into the surge region, the nonlinearities will deform the elliptical shape (Figure 5b).



(a) Normal orbits

(b) Surge orbit

Figure 5: Examples of orbits on steady state map [10]

The impedance line formulation explains why compressors can surge during fast transients even if the steady-state operating point appears stable. A rapid external pulse can drive the instantaneous operating point along the impedance line (or orbit) across the surge limit line, triggering deep surge, even if the mean flow remains to the right of the surge line. Brun, Nored, and Kurz [116] highlighted that reducing discharge volume to minimize stored energy often results in higher acoustic impedance. While this reduces the

energy available for a full surge cycle, the steeper acoustic load line amplifies pressure pulsations, potentially causing the compressor to violate vibration limits or trip on high discharge pressure before aerodynamic reversal occurs.

While Sparks' impedance lines describe the path permitted by the piping acoustics, Botros demonstrates that the speed at which the compressor traverses this path during an ESD event is controlled by the system's mechanical and fluid inertia. In low-inertia systems, the rotor speed decays rapidly, and the operating point effectively "crashes" into the surge line following a trajectory dictated by the acoustic relief capacity (impedance) of the recycle system. Therefore, the behaviour of a centrifugal compressor during fast transients is not an aerodynamic property of the impeller alone, but a system-dependent trajectory constrained by the acoustic load lines of the suction and discharge piping. This highlights how the phenomenon is intrinsically a characteristic of the system behaviour, rather than solely of the machine.

5.02. Reciprocating compressors pulsations response

While in centrifugal systems the acoustic impedance primarily acts as a boundary condition defining stability limits, in reciprocating compressor systems, the interaction is active and harmonic. Here, the machine does not generate a steady flow but functions as an intermittent flow modulator, and the piping system acts as a frequency-selective amplifier.

Pressure pulsations within piping systems constitute an acoustic phenomenon characterized by the propagation of unsteady pressure waves. These disturbances travel through the fluid medium at the local speed of sound. Assuming the fluid behaves as an ideal gas and the process is isentropic; the sonic velocity is defined by the thermodynamic relationship:

$$c = \sqrt{\frac{\gamma RT}{M}} \quad (4)$$

Where:

- γ is the adiabatic index,
- R is the universal gas constant,
- T is the absolute gas temperature,
- M is the molecular mass of the gas.

The acoustic response of a piping segment is governed by its geometric configuration and boundary conditions. A pipe behaves as an acoustic resonator, amplifying pressure waves when the excitation frequency coincides with the system's natural frequency (or eigenfrequency). The mechanism of resonance is driven by wave

reflection at the pipe boundaries. An incident pressure wave traveling at sonic velocity will reflect upon reaching a boundary. The phase of the reflected wave depends on the impedance mismatch: an open end results in a phase inversion (positive incident wave becomes negative reflected wave), whereas a closed end reflects the wave in phase (positive remains positive). When a reflected wave encounters a subsequent incident wave, the principle of superposition applies. Constructive interference occurs when waves of the same sign coincide, potentially leading to high-amplitude standing waves. The natural frequency for a specific pipe harmonic mode is derived from the longitudinal wave equation:

$$f_n = \frac{nc}{kl} \quad (5)$$

Where:

- c is the speed of sound [m/s].
- l is the pipe length [m].
- n is a harmonic number ($n = 1, 2, 3...$ for symmetric boundaries; $n = 1, 3, 5...$ for asymmetric boundaries).
- k is the boundary condition factor ($k=2$ for open-open or closed-closed; $k=4$ for open-closed)

It is important to note that while Eq. 5 provides a fundamental understanding of resonance in simple geometries, industrial piping networks possess complex impedance landscapes. Consequently, accurate prediction in real-world scenarios requires numerical methods, such as the transfer matrix method or time-domain solvers (1D-CFD), to account for damping, complex branches, and interactions.

Figure 6 illustrates the acoustic mode shapes for these configurations, visualizing the pressure nodes and antinodes associated with the resonance frequencies.

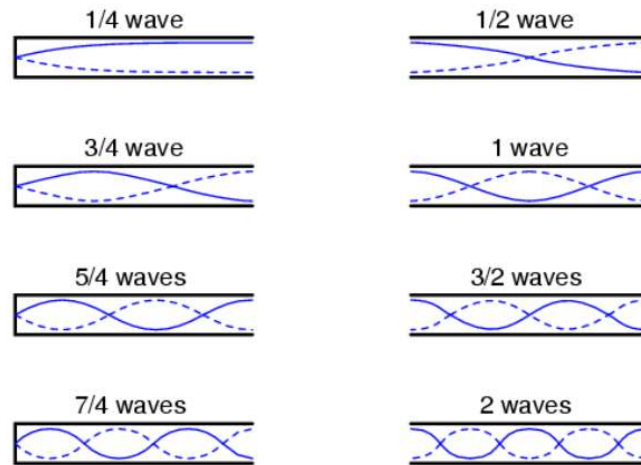


Figure 6: Pipes wave shape

Reciprocating compressors act as the primary source of flow-induced pulsations. The spectral content of these pulsations is intrinsically linked to the compressor's rotational speed and the kinematics of the crank-slider mechanism.

The instantaneous piston velocity, which drives the unsteady mass flow into the piping, is governed by the geometry of the crank radius and the connecting rod length. It can be expressed as:

$$u_p(t) = R\omega[\sin(\omega t) + \frac{R}{2l}\sin(2\omega t)] \quad (6)$$

Where:

- ω is the angular velocity of the crankshaft.
- T is time.
- R is the crank radius
- l is the connecting rod length

As demonstrated in Figure 7, the piston velocity profile closely approximates a sinusoidal waveform. The deviation from a pure sine wave is dictated by the ratio R/l ; as the connecting rod length l increases relative to the crank radius, the motion approaches simple harmonic motion, reducing the amplitude of higher-order harmonics.

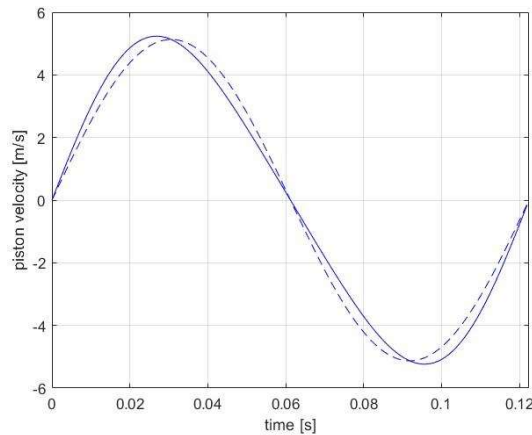


Figure 7: Piston speed (continuous) vs sin wave (dashed)

The harmonic content of the generated pressure pulsations varies significantly based on the cylinder's compression cycle and geometric symmetry. In a single-acting configuration, compression occurs on only one side of the piston. Consequently, the flow profile is discontinuous. The Fourier decomposition of this signal, as shown in Figure 8, reveals a spectrum containing the fundamental frequency (corresponding to the shaft speed) and a significant presence of higher-order harmonics.

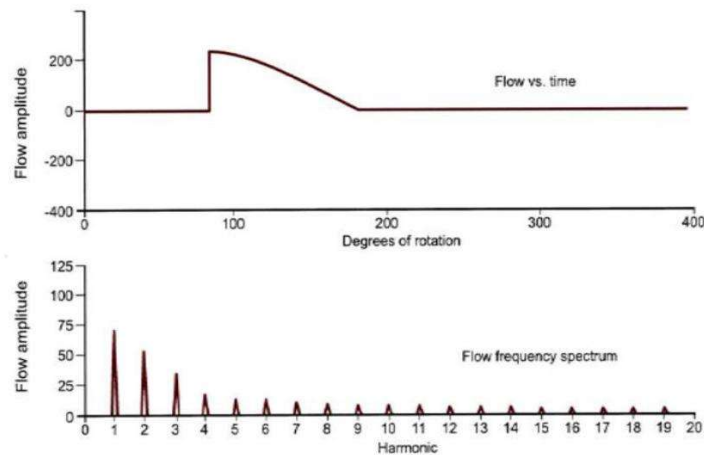


Figure 8: Single acting compressor pulsations

In a symmetrical double-acting cylinder, compression occurs on both the head-end and crank-end. Due to the inherent phase shift of the crank mechanism, the pressure pulses generated by the two chambers are separated by a specific crank angle (180°). In a perfectly symmetrical scenario, this phase relationship causes destructive interference for specific frequencies. Notably, all odd harmonics represent opposing phases and cancel out, leaving only the even harmonics in the frequency spectrum (Figure 9).

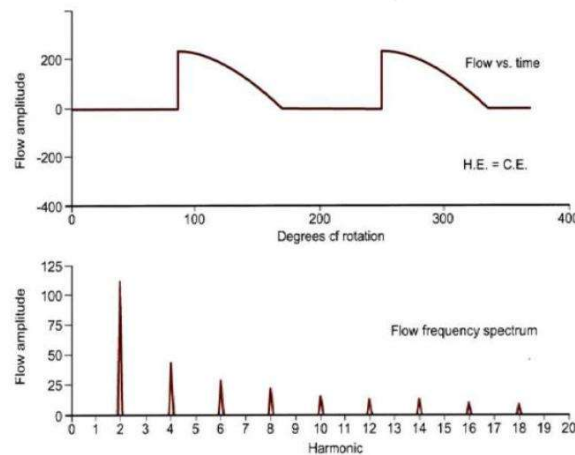


Figure 9: Symmetrical double acting compressor pulsations

In industrial applications, double-acting cylinders are usually non-symmetrical due to the presence of the piston rod on the crank-end side. The piston rod reduces the effective area of the crank-end face, resulting in a lower displacement volume compared to the head-end. The flow rates generated by the two chambers are unequal. Consequently, the destructive interference observed in the symmetrical case is incomplete. While the even harmonics remain dominant, the odd harmonics re-emerge in the spectrum (Figure 10), although with lower amplitudes compared to the single-acting configuration.

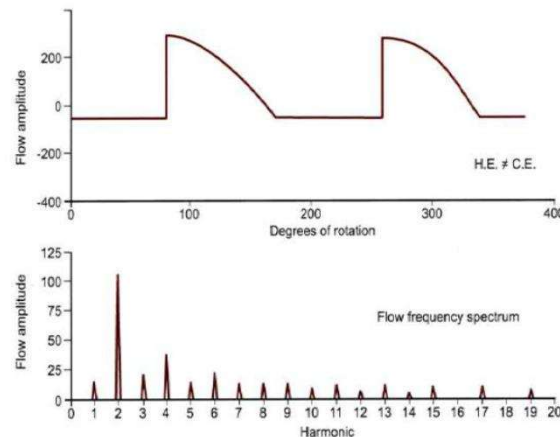


Figure 10: Non-symmetrical double acting compressor pulsations

The analysis of pressure standing waves is not an independent matter but a prerequisite for evaluating mechanical integrity. While pulsations don't interact mechanically with straight pipes, a standing wave generates a pressure differential across piping discontinuities such as elbows, reducers, or closed valves. This differential results in a dynamic force, commonly referred to as a "shaking force" or "unbalanced force", which acts as the excitation source for the mechanical system. The shaking force can be defined

from the momentum conservation principle as the surface integral of the time-dependent pressure field $p(t)$ acting on the internal piping surface S :

$$F_{\text{Shaking}} = \oint_S p(t) \cdot \hat{n} \, dA \quad (7)$$

API 688 [3] provides simplified versions of Eq. 7 for each piping discontinuity.

When the frequency of these acoustic shaking forces coincides with a mechanical natural frequency of the piping structure, high-amplitude vibrations occur, leading to cycle fatigue and potential failure. Thus, acoustic resonance is the direct precursor to structural resonance.

In summary, this chapter has highlighted that acoustic response is not merely a fluid-dynamic artifact but a critical system parameter for both centrifugal and reciprocating technologies. Whether determining the surge margin through impedance lines in centrifugal compressors or defining the shaking force magnitude in reciprocating units, the piping acoustic characteristics dictate the operational limits and reliability of the machinery.

Part I: Dynamic system analysis

1. Introduction

Centrifugal compressors have diverse applications, such as air separation units, chemical plants, and vehicles. During the design of these systems, particular attention must be paid to the surge phenomenon, as it can lead to significant damage to the compressor. Surge is a critical event, as the compressor works in unstable flow conditions, especially during transient operations. This instability occurs at low flow rates and high-pressure ratios where the compressor performance curve (p-Q curve) exhibits a shallow decline or an increasing slope. The occurrence of surge is also influenced by the discharge volume as the pressure differential over the compressor requires more time to decrease in presence of bigger volumes. The surge phenomenon may occur during transient operations, and it's marked by a rapid dynamic as it evolves in few seconds. So, any analysis tool has to consider the transient nature of the phenomenon in order to be effective in surge prediction. Many solutions are available to address the surge phenomenon. The main ones involve the installation of one or two recirculation valves. The antisurge control system usually includes a cold-gas recycle (CGR) valve, connecting the suction side with the discharge section downstream the cooler. Depending on the plant arrangement (volumes, valves opening times, etc.), the addition of a hot-gas recycle (HGR) valve may be necessary. This solution is more efficient for antisurge control as it bypasses the cooler's volume. To evaluate the effectiveness of a chosen solution, a Dynamic System Analysis (DSA) of the compressor plant is necessary. Moreover, these analyses are also used for prevention of overheating during recycle loop operations. Despite the literature presenting many studies it is still not always sure for which cases a cold-gas recycle is enough for surge prevention [36]. On the other hand, the installation of a hot-gas recycle is a challenging and expensive solution. So, the development of a model is a necessary step to verify a system compliance to anti-surge requirements, especially for emergency shutdown (ESD) scenarios [117].

Several contributions on the topic are available in literature. Many textbooks such as [9] provide an effective explanation and definition of surge, useful for the understanding of the phenomenon. Sparks [10] explains how the piping system affects the compressor transient behaviour. Kurz et al. [118] summarize different approaches for estimating the compressor's train speed. MacDougall et al. [17] develop a model able to simulate a centrifugal compressor behaviour during transient conditions. Morini et al. [18] propose a 1D simulation model to predict the surge occurrence extending the performance map to the backflow region (negative mass flow rate). Pak et al. [28] show an interesting

approach for a surge analysis during emergency shutdown operations. Almasi [34] highlights several important parameters and considerations useful for recycle valve system design and sizing. Haeckel et al. [119] propose a new compressor model aimed at enhancing the numerical definition of the surge limit. Marsano et al. [120] offer a comprehensive review of existing centrifugal compressor limits related to flow instabilities. Furthermore, Qiao et al. [121] investigate the occurrence of a secondary surge following the recovery from an initial surge event. Botros defines a dimensionless parameter, named as inertia number [35], useful to decide if a hot-gas recycle is required for safe operations [36]. Yoon et al. [37] further develop Botros's work by establishing another dimensionless parameter, the surge number. This new model should provide higher certainty whereas the previous one is not able to predict whether a hot-gas recycle is necessary. Brun et al. [19, 20] study the stresses acting on the compressor train during the surge phenomenon. They develop a lumped parameter model useful to determine the forces that appear during surge or any transient operation. Epstein et al. [122] compare the effectiveness of active and passive suppression system for compressor instabilities. Tamaki [23] investigate the piping system influence on the surge line through a lumped parameters model. Powers et al. [13] develop a quintic characteristic curve (flow-head) that allows to differentiate between mild and severe surge. Pinsley et al. [30] develop a suppression system for small perturbations based on a servo valve placed on the plenum outlet, expanding the operational range for low-flow rates. Gravdahl et al. [31] further develop Pinsley's work by adding a speed control on the compressor drive. Blanchini et al. [123] propose an effective control strategy for surge suppression assessed through experimental test. Zou et al. [25] analyse how the diffuser vanes presence influences the rotating stall onset and the surge line position.

The current chapter focuses on the dynamic analysis of a centrifugal compressor useful to assess the occurrence of the surge phenomenon. Initially, key plant parameters are gathered to construct a 1D fluid dynamic (CFD) model. The handling of experimental data is crucial in developing an accurate compressor performance map. With the model in place, simulations of the shutdown operation, considered the most severe transient condition, are conducted. The primary goal is to assess the occurrence and severity of the surge phenomenon. While simplified dimensionless models offer initial guidance for surge protection, they present limitations when applied to complex multi-stage machines and typically exclude the consideration of alternative solutions. This motivates the need for a comprehensive dynamic analysis. Subsequently, the analysis is broadened to evaluate the impact of various parameters on the system dynamics and to explore different layouts. Specifically, the focus is on the vent ones, which are considered cost-effective solutions for

the present case if compared to the HGR. Moreover, the analysis of these layouts would expand the literature, which is mainly focused on the hot-gas recycle. So, the comparison of results provides useful insights into how different solutions address the surge phenomenon and highlights key parameters that need consideration. Consequently, this comprehensive study not only aims to examine the surge phenomenon during shutdown operations but also to assess the influence of different parameters and layouts, in order to offer valuable understanding for optimizing compressor performance.

2. Materials and Methods

2.1. System Description

The case study is a two-sections compressor with inter-cooler and after-cooler (Figure 12). The machine is integrally geared. The basic version of the plant includes a slow 8" cold-gas recycle valve (S-CGRV). A check valve is placed in the discharge line, after the cooler, to prevent backflow and to facilitate a pressure drop in the discharge side during the recycle valve opening. The processed gas is nitrogen, which has a safe nature. This feature makes the vent valves a feasible solution for addressing the surge phenomenon during transient operations. An electric motor drives the compressor through a gearbox. An inlet guide vane system adjusts the machine's performance according to the inlet and the discharge parameters. The gas enters the inlet port at 8°C and 6 barA. The discharge port conditions are, nominally, 40°C and 28.5 barA. The compressor nominally processes 56000 kg/h. Table 1 reports the main parameters used in the model. Then, referring to Botros [35] and Yoon [37], both the inertia number and the surge number are calculated for this system: $N_I=3.64$, $N_S=153291$. According to these numbers, Yoon's model claims that the cold-gas recycle is not adequate and that the system requires a hot-gas recycle valve to prevent surge occurrence. The system is deemed unsafe without an HGRV as the value of N_I is lower than 30 and N_S is larger than 1678. These thresholds are indicated in the model's map *Figure 11*.

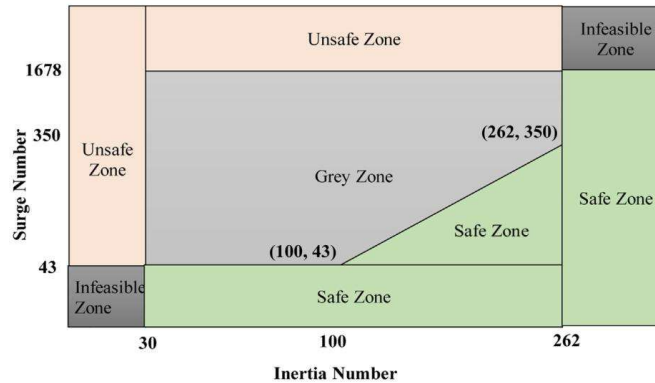


Figure 11: Yoon's model map [37]

However, one should notice that Yoon’s model is developed for single stage compressors, while the present study deals with a two-sections machine. Indeed the calculations were performed for reference, rather than testing the reliability of the model.

Table 1. Case study parameters.

Parameter	Value	Unit
S-CGR valve diameter	8	inches
S-CGR valve flow coefficient	276	m ³ /h
Discharge volume	6.1	m ³
Interstage volume	8.8	m ³
Intake volume	11.1	m ³
Nominal mass flow rate	56000	kg/h
Inlet pressure	6	barA
Inlet Temperature	8	°C
Discharge pressure	28.5	barA
Discharge Temperature	40	°C
Compressor train equivalent inertia	16.277	kg m ²

Fluid	Value	Definition
Reference fluid	Nitrogen	Semi-perfect

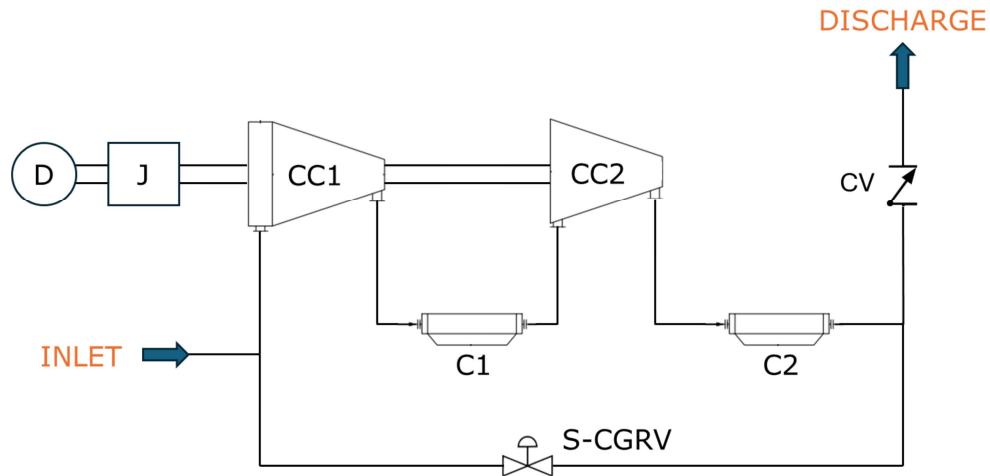


Figure 12. Sketch of the compression unit basic layout (D: Driver, J: Inertia, CC1: First Stage, CC2: Second Stage, C1: Interstage Cooler, C2: Discharge Cooler, S-CGRV: Slow Cold Gas Recycle Valve, CV: Check Valve).

2.2. Virtual Model

To represent the compressor unit, it is necessary to model both the mechanical and the fluid-dynamic components. For the mechanical sub-system, a speed trend derived from literature [118] is imposed. Meanwhile, for the fluid-dynamic components a 1D-CFD representation is chosen to take into account wave propagation phenomena, which are

essential for a realistic transient analysis. This approach is critical for the assessment of surge occurrence, as this system-wide in-stability is dependent on the acoustic dynamics and transient wave propagation throughout the piping system [10]. Consequently, Simcenter Amesim version 2504 [124] is chosen for developing the model as it is able to perform 1D-CFD simulations [125]. For the shaft speed trend (Figure 13) the following equation [118] is obtained, assuming a constant Q/N ratio, the fan law and neglecting friction losses:

$$N(t) = \frac{J N_0^3}{J N_0^2 + P_0 t} = \frac{J N_0^2}{J N_0 + T_0 t} \quad (8)$$

Where:

- $N(t)$ is the shaft speed [rad/s]
- N_0 is the initial shaft speed [rad/s]
- J is the compressor train inertia [kg m²]
- P_0 is the initial power consumed by the compressor [W]
- T_0 is the initial torque on the shaft [Nm]
- t is the time from the shutdown event start [s]

Then, a comparison with constant speed decays trends [118] is performed:

$$N(t) = e^{\ln(1-k)*t} \quad (9)$$

Where:

- k is the speed decay [1/s]

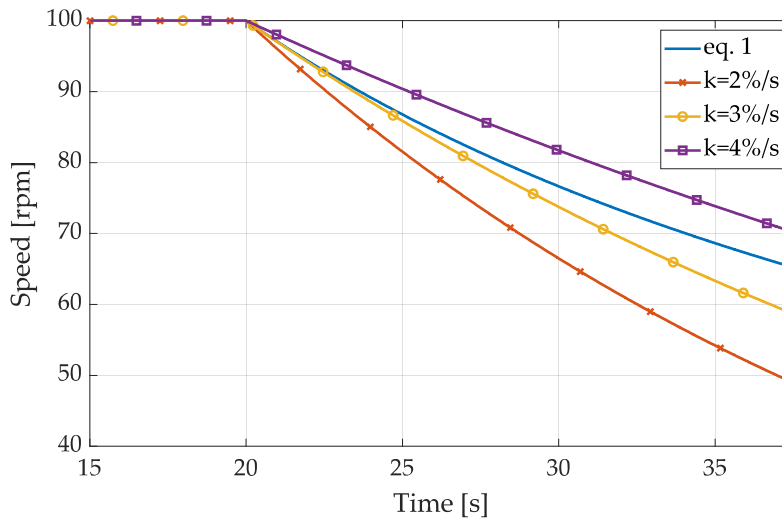


Figure 13. Speed trends used as input.

The non-linear decay model is the primary input for the main transient simulations as it provides a more realistic speed input for the severe shutdown transient as it is derived from the conservation of angular momentum and consider the non-linear dependency between speed and aerodynamic resistance (Fan Law). While constant decay rates are useful for preliminary sensitivity analyses or for more conservative assumptions. For the fluid-dynamic, the 1D-CFD approach involves the use of continuity (eq. 3), momentum (eq. 4), and total energy (eq. 5) equations to simulate each fluid-dynamic element.

$$\frac{\partial \rho S}{\partial t} + \frac{\partial \rho u S}{\partial x} = 0 \quad (10)$$

$$\frac{\partial \rho u S}{\partial t} + \frac{\partial (\rho u^2 + p) S}{\partial x} - p \frac{\partial S}{\partial x} + \frac{1}{4} f |u| \rho u \sqrt{\pi S} = 0 \quad (11)$$

$$\frac{\partial \rho E S}{\partial t} + \frac{\partial (\rho E + p) u S}{\partial x} - 2(h_c(T_{wall} - T) + h_r(T_{wall}^4 - T^4))\sqrt{\pi S} = 0 \quad (12)$$

Where:

- ρ is the gas density
- u is the gas speed
- S is the pipe section
- t is the time
- x is the abscissa
- p is the pressure
- f is the friction coefficient
- T is the gas temperature
- T_{wall} is the pipe's wall temperature
- h_c is the convective heat transfer coefficient
- h_r is the radiative heat exchange coefficient

The discretization of the governing equations above was achieved by dividing the pipe network into N cells, each having a uniform spatial step, Δx . The selection of the cell size is intrinsically linked to the numerical scheme adopted and the requirement for stable computation. In this study, the Lax-Wendroff two-step scheme, as proposed by Richtmyer and Morton [126] and modified by Bassett et al.. [127], was employed.

For this model, numerical stability and convergence are ensured by satisfying the Courant-Friedrichs-Lewy (CFL) condition (eq. 6), which dictates the relationship between the spatial step, the time step and the maximum absolute velocity of the fluid flow ($|u|$).

$$\frac{(a + |u|)\Delta t}{\Delta x} = CFL \quad (13)$$

To maintain a balance between computational efficiency and accuracy, a CFL number equal to 0.7 was specified, following established best practices. A uniform spatial discretization of $\Delta x = 100$ mm was selected. Given this fixed spatial step and using the maximum expected characteristic velocity, the software automatically calculates the time step for the working fluid, which is Nitrogen (N_2). For the defined system conditions, the resulting time step was of the order of $1.5 \cdot 10^{-4}$ s, ensuring the stability and high-frequency resolution required for the transient simulation.

The boundary conditions are imposed as fixed temperatures and pressures, Table 2 reports the values for inlet, discharge and vent ports.

Table 2. Boundary conditions

Port	Temperature [°C]	Pressure [barA]
Inlet	8	6
Discharge	40	28.5
Vent	20	1.01

For the fluid-dynamic, the 1D-CFD approach involves the use of continuity, momentum, and total energy equations to simulate each fluid-dynamic element. Then, to describe the centrifugal compressor behaviour, performance map data are collected. This map is represented through a table containing the shaft speed, compression ratio, volume flow rate and efficiency. Missing iso-speeds curves are extrapolated using the fan law. To simulate the shutdown operation and check the surge occurrence a complete map is necessary, while the iso-speed curves are limited inside the operating range. Consequently, the data was extrapolated in four regions of the map:

- Low speed,
- Maximum speed,
- Surge,
- Choke.

A Riemann solver and the mass conservation equation were used to find the operative conditions of the compressor on the map, correcting the parameters according to the intake temperature and pressure. The 1D-CFD approach is used for pipes and coolers, while valves are represented as pressure loss elements. The valves' flow coefficient vs time trends are considered linear during opening (Figure 16). Their behaviour is described through a constant discharge coefficient (C_v) and a maximum area. Pipes and cooler dimensions are computed according to measured length, diameters, and volumes. The piping and coolers are considered ideal; thus, they are considered rigid. The gas, pure Nitrogen, is modelled as semi-perfect.

All valves are evaluated using standard diameters (even numbers from 2" to 12"), while the S-CGRV does not change size (8"). Table 4 reports the flow coefficients, and the opening and delay times for each diameter. The flow area is assumed to be linear over time during opening. However, real fast-acting valves may possess a non-linear flow area characteristic with respect to stem travel and time.

Table 3. Valves parameters.

	S-CGRS	Fast acting valves					
Valve size	8"	2"	4"	6"	8"	10"	12"
Flow coefficient (Cv)	320	45	188	367	578	918	1180
Opening time	5 s	1 s	1.2 s	1.8 s	2.4 s	3 s	3.6 s
Delay time	0.2s	0.2 s	0.2 s	0.2 s	0.2 s	0.2 s	0.2 s

Five different plant layouts have been modelled:

- Basic layout,
- Fast cold gas recycle (F-CGR),
- Single vent (1V),
- Double vent (2V),
- Hot gas recycle (HGR).

The first one (Figure 12) is the existing plant layout without any modifications. In the second one (Figure 14a), a fast-acting valve is installed in tandem with the S-CGR one. Here, the new valve is responsible for preventing surge occurrence during fast transient operations (such as the ESD), while the slow one is more accurate during partial load operations. The third layout (Figure 14b) considers one vent valve on the second stage discharge section after the cooler. This valve aims to depressurize this section of the plant during a shutdown operation to prevent the occurrence of surge. The fourth layout (Figure 14c) adds another vent valve to the single vent layout in the interstage piping before the cooler. Here, the objective is to reduce the interstage pressure and prevent the first stage from surging. Simulation runs are performed considering both valves are the same size. Lastly, a hot-gas recycle layout has been simulated (Figure 14d). Here, a hot-gas recirculation valve has been added for each stage in the base model. The simulations assumed both valves to have the same size. For the HGR valves have the same discharge coefficients and opening times as the fast acting valves in Table 3.

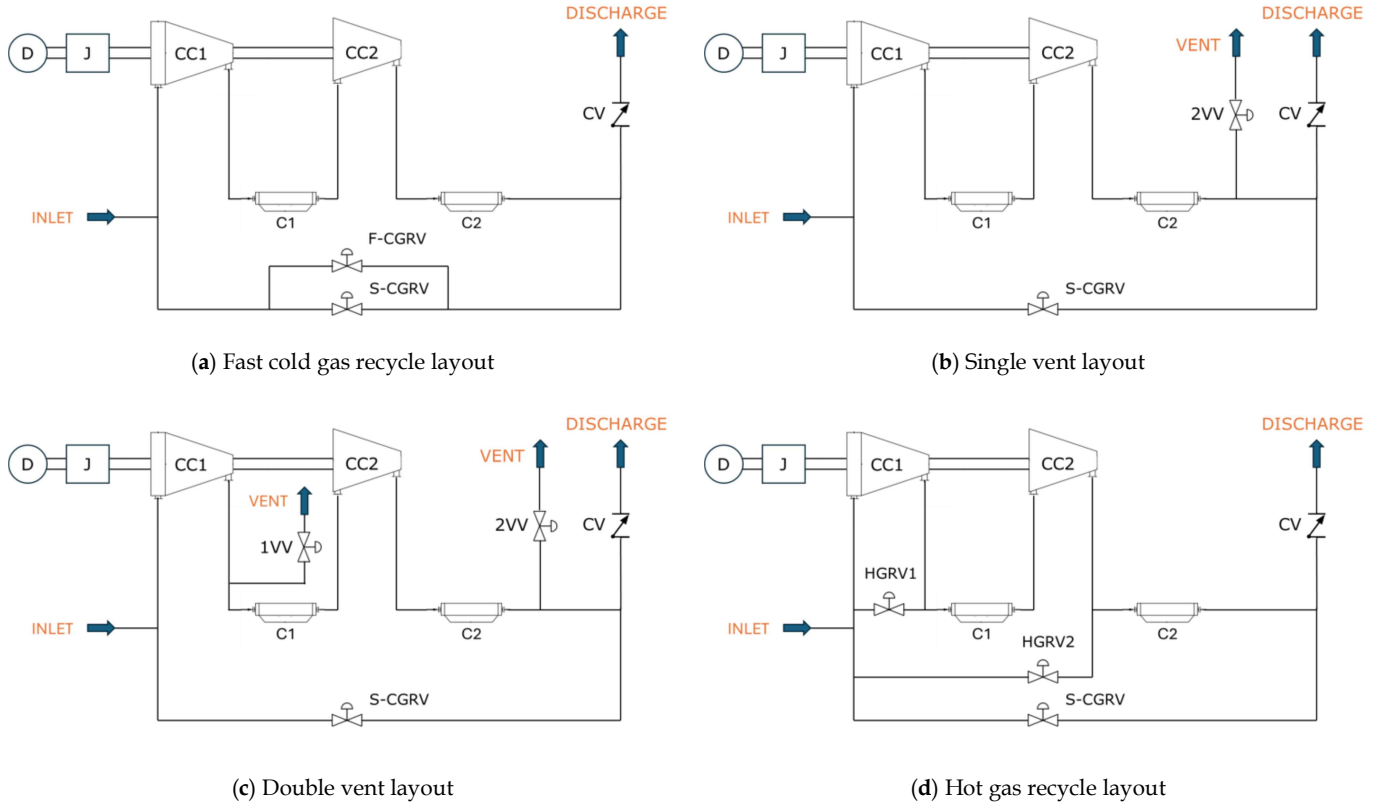


Figure 14. Sketch of the compression unit basic layout. (D: Driver, J: Inertia, CC1: First Stage, CC2: second stage, C1: Interstage Cooler, C2: Discharge Cooler, S-CGRV: Slow Cold Gas Recycle Valve, CV: Check Valve, F-CGRV: Fast Cold Gas Recycle Valve, 1VV: Interstage Vent Valve, 2VV: Discharge Vent Valve, HGRV1: First Stage Hot Gas Recycle Valve, HGRV2: Second Stage Hot Gas Recycle Valve).

3. Results and Discussion

Different layouts and cases have been investigated. For each one, a shutdown operation is simulated starting from the rated operative conditions. For each layout the conditions are represented on each stage map (Figures 4 and 6-9). Here, six iso-speed curves (grey), the operative point trajectory, and the surge line (red) are represented. In the beginning, the operative point is located on the nominal speed curve (100%). After the shutdown operation starts, it moves to lower speeds. If the operative point moves to the left of the surge line, the surge occurs, and damage may happen to the machine. In these cases, the trajectory is trimmed at the red line as the surge region is unstable. The assessment of this condition has to be done for both stages. Additionally, the main surge onset times are collected for each case (Table 4) and will be compared in the subsections below. A dash (-) is reported in the table if surge did not occur. The surge phenomenon can be considered worse if it happens earlier and at higher shaft speeds.

Table 4. Surge occurrence timing

	Basic layout				F-CGR layout					
	Eq. 1	K=2%/s	K=3%/s	K=4%/s	2"	4"	6"	8"	10"	12"
1 st stage	1.35 s	19.6 s	1.32 s	0.99 s	1.35 s	1.38 s	1.42 s	1.48 s	1.64 s	1.71 s
2 nd stage	1.55 s	2.25 s	1.51 s	1.13 s	1.55 s	-	-	-	-	-

	1V layout						2V layout					
	2"	4"	6"	8"	10"	12"	2"	4"	6"	8"	10"	12"
1 st stage	1.35 s	1.38 s	1.41 s	1.46 s	1.59 s	1.65 s	1.65 s	-	-	-	-	-
2 nd stage	1.55 s	-	-	-	-	-	1.32 s	1.26 s	-	-	-	-

	HGR layout					
	2"	4"	6"	8"	10"	12"
1 st stage	1.65 s	6.09 s	-	-	-	-
2 nd stage	1.33 s	1.18 s	-	-	-	-

3.1. Basic Layout

The first simulations concern the basic layout in order to assess whether the plant would surge without any additional anti-surge device. The batch run involves all the speed trends in Figure 13. The results highlight how the compressor is prone to surge. Even for a decay rate of 2 %/s, the surge occurs after about 2 s in both stages. Consequently, an additional anti-surge system is considered necessary to avoid surge during ESD operations. From the results of eq. 1 speed, we can assume that the additional antisurge system has about 1.3 s to effectively address the phenomenon. While the S-CGR takes 5.2 s from the shutdown to fully open. Figure 15 and Table 4 show how assuming a constant speed decay rate (3%/s) can provide a reasonable approximation for equation 1.

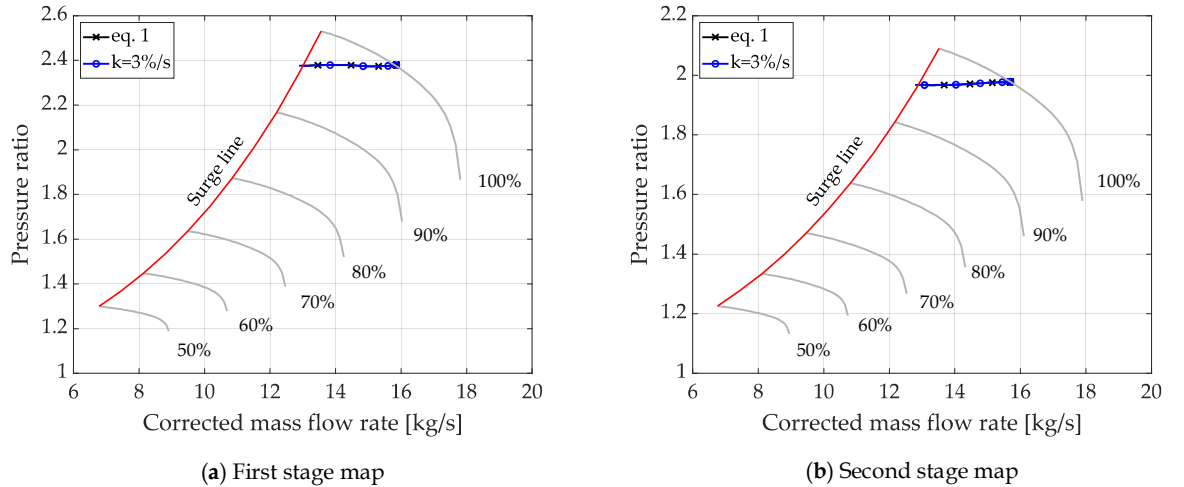


Figure 15. Basic layout results for Equation 1 and $k=3\%/s$ on compressor maps. Surge line in red and isospeed lines in grey.

3.2. F-CGR Layout

After the basic layout, the F-CGR one is analysed. Here the eq. 1 is used as input for the compressor shaft speed. As stated before, the simulations are performed for different values of the fast-acting valve size between 2" and 12". However, as two valves are opening in parallel the equivalent open area has been calculated over time as a percentage of the 8" flow area.

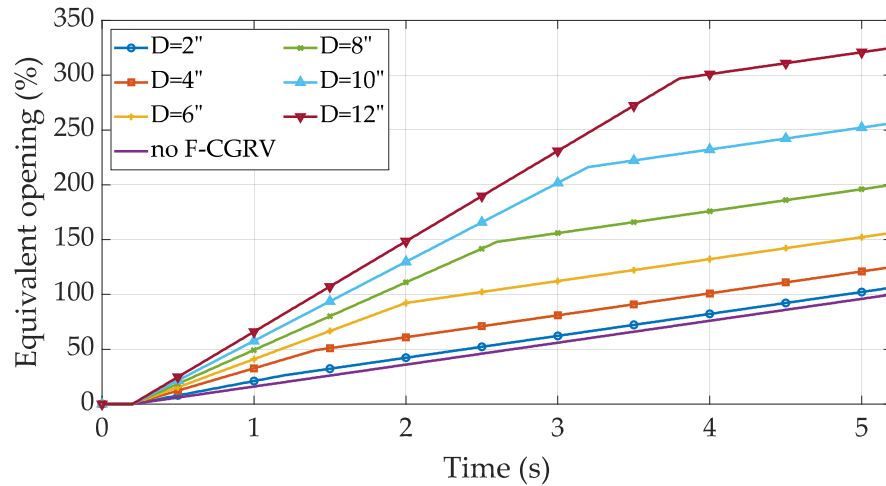
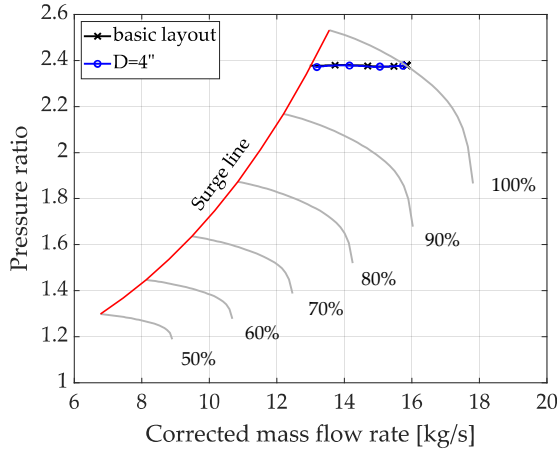


Figure 16. Equivalent recirculation flow area expressed as percentage of S-GRV flow area at fully open conditions.

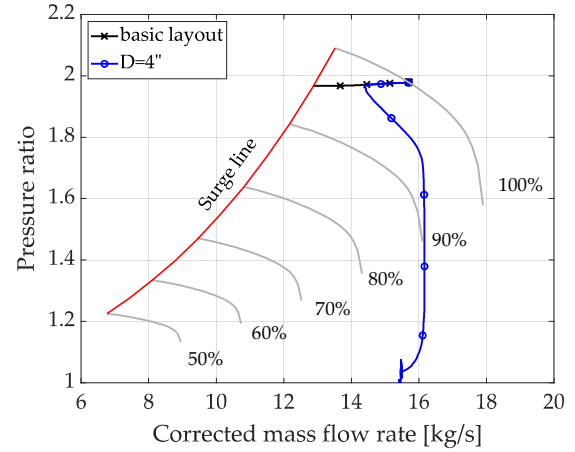
Here, it is possible to notice in Figure 16 how for bigger F-CGR valves the system achieves a 100% flow area earlier providing a better resistance to surge. As the cold recycle system is modified the new inertia and surge numbers have been recalculated (Table 5). It is possible to notice how the inertia number remains unchanged. Then, the results in figures 5 highlight how 4" size is the minimum value to prevent surge in the second stage. However, this solution is not adequate for addressing the phenomenon for the first stage. This happens because the interstage pressure remains relatively unchanged, maintaining the first stage pressure ratio high.

Table 5. Surge and inertia numbers calculated for each F-CGRV size.

	Basic layout			F-CGR layout			
Valve size	-	2"	4"	6"	8"	10"	12"
Inertia number N_I	3.64	3.64	3.64	3.64	3.64	3.64	3.64
Surge number N_s	153291	144093	113435	67448	49053	42921	36790



(a) First stage map

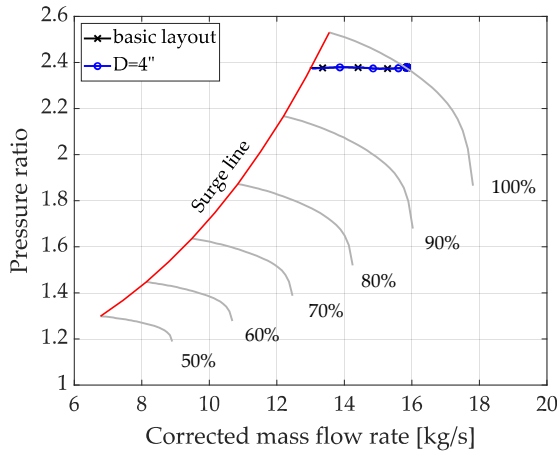


(b) Second stage map

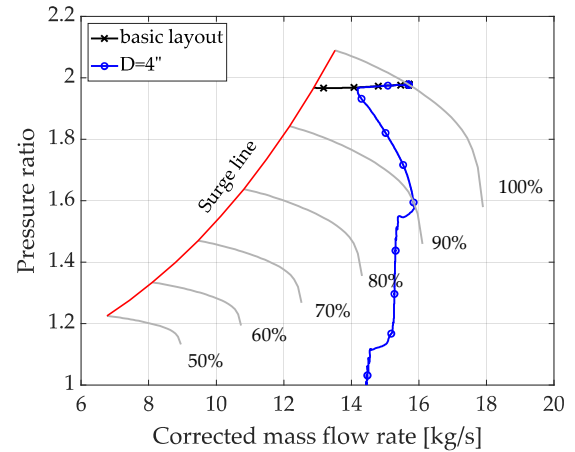
Figure 17. Comparison of F-CGR layout with a 4" valve with the basic layout results. Surge line in red and isospeed lines in grey.

3.3. 1V layout

Then, the single vent layout has been investigated. The results highlight how this solution proves to be very similar to the F-CGR one. Figure 17 shows a comparison between the basic layout and the 4" 1V layout. The operative conditions trajectories for both stages appear similar to the previous case. Moreover, the surge timing (Table 4) show how the 1V layout is slightly more effective, especially for larger valves. This effect is ascribed to the fact that the pressure drop over the vent valve is higher than the recirculation one. This implies that the mass flow rate is higher for the same flow coefficient. Consequently, also the discharge volume's pressure varies with the same trend.



(a) First stage map

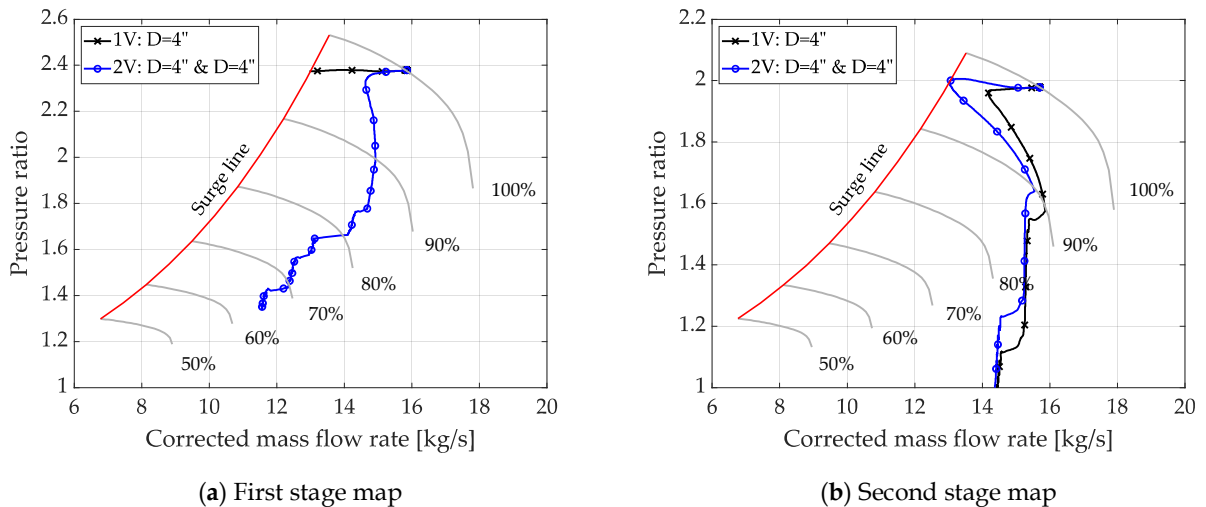


(b) Second stage map

Figure 18. Comparison of 1V layout with a 4" valve with the basic layout results. Surge line in red and isospeed lines in grey.

3.4. 2V layout

The double vent layout is then investigated in order to address the influence of an interstage vent valve on the surge occurrence. Here, the results highlight how the additional valve helps mitigating the phenomenon on the first stage of the compressor, while it worsens the situation on the second one. The compressor maps in Figure 19 show how the first stage compression ratio is reduced, while the second one is increased in the first interval. This is related to the interstage pressure reduction. The surge timing (Table 3) confirms this as the phenomenon may occur in the second stage even earlier than the basic layout. Surge prevention is obtained for the 2V 6" layout, as using 4" size for both valves implies that the second stage falls into surge.



(a) First stage map

(b) Second stage map

Figure 19. Comparison of 2V layout with two 4" valve with the 4" 1V layout results. Surge line in red and isospeed lines in grey.

3.5. HGR layout

Lastly, the results from the HGR layout are collected. Here, the surge is prevented when both valves' diameters are equal to 6". The first stage recirculation valve increases the second stage pressure ratio as it happened with the interstage vent valve. This phenomenon reduces the surge margin of the second stage. Another important parameter for this layout is the intake temperature, as the hot discharge gas is sent to the machine intake without cooling. Consequently, the suction temperature peaks at about 100 °C few seconds after the shutdown. With the installation of an intake check valve this value may further worsen.

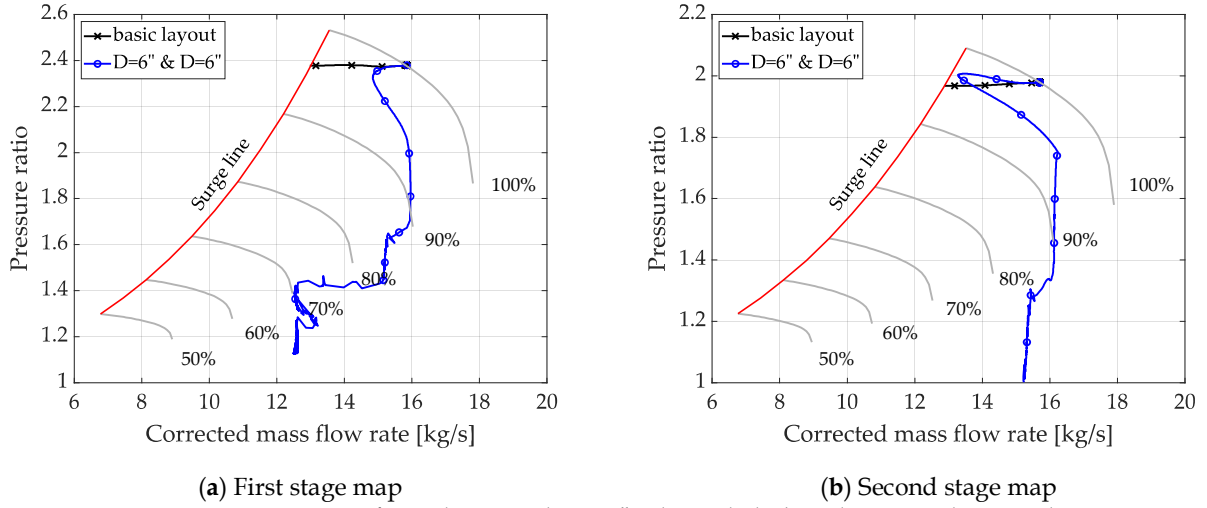


Figure 20. Comparison of HGR layout with two 6" valve with the basic layout results. Surge line in red and isospeed lines in grey

4. Conclusions

In conclusion, this study focuses on the assessment of vent valves as a suitable solution for surge occurrence in comparison to the traditional layouts. In order to achieve this, a virtual model is created using the Amesim environment. The model allows the simulation of transient operations, useful for the evaluation of the compressor plant performances. The analysis starts from a comparison of several speed trends, derived from literature. These offer a good approximation of the mechanical behaviour of the compressor and give the opportunity to implement more conservative conditions to the simulation. It allows to balance the uncertainties of the real system parameters. After, the surge occurrence has been assessed the focus moved to the available solutions. The model's results show how the 1V layout performances are similar to the F-CGR ones. While it is not always convenient (or possible) to release the processed gas into the environment, this solution is easier to implement. Then, the results highlight the importance of installing a second valve to avoid the surge occurrence in the first stage. Consequently, the 2V and the HGR layout are developed. The first one proves to be more efficient than the second one (Table 4).

Critically, the HGR layout leads to a peak suction temperature increase (reaching approximately 100°C), which poses a significant risk of compressor damage and material degradation, making this solution highly problematic in practice. Furthermore, the comparison reveals a key practical implication: to prevent the surge phenomenon from occurring, the HGR layout requires large 6" valves, whereas the superior 2V configuration only requires 4" valves. This, substantial difference in layout and in required valve sizing is

a major finding, as it translates directly into lower system cost, reduced physical footprint, and simplified piping complexity for the 2V system.

Moreover, the second layout leads to an intake temperature increase, which may result in compressor damage. So, to prevent the surge phenomenon from occurring the 2V or the HGR layout with 6" and 6" size valves are necessary. Then, the present model proved to be consistent with Yoon's one. Despite the latter is able to predict if a hot-gas recycle is required to prevent surge during ESD, it does not provide information about the minimum HGR valves size. A detailed fluid-dynamic model is necessary for choosing the valves diameter, evaluate different strategies (vent valves) and explore the limits of simplified dimension-less models. In essence, this research sheds light on the challenges associated with surge control in compressor units, emphasizing the need for a comprehensive approach that considers various parameters. The results provide a roadmap for future investigations and system redesigns, ultimately contributing to the advancement of compressor technology.

Part II: Reciprocating compressors

Introduction

Reciprocating compressors are positive displacement machines inherently characterized by unsteady flow delivery. Unlike continuous flow machinery, the intermittent nature of the piston motion inevitably generates pressure pulsations that propagate throughout the entire piping network.

If not adequately controlled and dampened, these pulsations give rise to complex acoustic phenomena with multifaceted consequences that extend beyond simple noise generation. From a structural perspective, the primary concern is the generation of unbalanced dynamic forces, commonly referred to as shaking forces, acting on piping elbows and reducers. These forces trigger Flow-Induced Vibrations (FIV), which pose a severe risk to the plant's integrity, potentially leading to fatigue failures of main piping, supports, or sensitive small-bore connections.

The research activities presented in this section were conducted entirely within the industrial context of Compression Service Technology (CST). The primary objective extended beyond the mere application of simplified design methodologies, aiming instead at the critical analysis and optimization of the workflow prescribed by the API 618 and 688 standards (Design Approach 3). This comprehensive approach necessitates an integrated multi-physics perspective to bridge the gap between two engineering domains often treated in isolation: the acoustic analysis, based on 1D fluid dynamics for predicting pulsations and exciting forces, and the mechanical analysis, performed via Finite Element Method (FEM) to evaluate structural response, natural frequencies, and fatigue stresses.

Crucially, it must be highlighted that industrial compressors do not operate at a single fixed point but are subject to a broad operational envelope. Even under steady-state mechanical operation, the thermodynamic boundary conditions, such as gas composition, pressure, and temperature, vary significantly due to seasonal environmental changes or process fluctuations. These variations directly alter the physical properties of the gas, most notably the speed of sound, which shifts the acoustic natural frequencies of the piping system. To account for this variability, the machine's datasheet defines a comprehensive matrix of operating cases. Consequently, the numerical analysis is not a single-point verification but a multi-case iterative process: simulations are performed for every scenario defined in the datasheet, and the design of countermeasures (supports and pulsation dampers) is driven by a worst-case envelope strategy, ensuring structural integrity under the most severe combination of operating parameters.

Materials and Methods

CFD1D: Gas modelling

To predict the acoustic behaviour of the piping network, the Simcenter Amesim simulation platform was selected, ensuring methodological continuity with Part I. However, while the previous analysis focused on macroscopic transient events, this section shifts the focus to the periodic nature of the pulsation phenomenon. Unlike fast transients, reciprocating machines typically operate at low rotational speeds; consequently, the critical aspect lies not in the speed of the event, but in the continuous interaction between the cyclic excitation and the frequency response of the piping geometry.

The choice of the calculation code is driven by the need for a strongly multiphysics approach. The pulsation phenomenon is not purely fluid-dynamic but results from a complex interaction between distinct physical domains: the mechanical behaviour of the machine (crank-slider kinematics), the dynamics of the valve moving elements (mass-spring systems), and the wave propagation within the fluid. Only a simulation environment capable of instantaneously coupling these physics allows for reliable predictive results.

Constructing the digital twin of a plant requires translating every physical component into a mathematical equivalent capable of replicating its acoustic and thermodynamic behaviour. The following details the modelling strategy adopted for the main subsystems: cylinders, automatic valves, plenums and pulsation dampers, heat exchangers and piping network.

The compressor cylinder is not modeled merely as a flow source, but as a variable-volume thermodynamic chamber. The instantaneous volume variation $V(\theta)$ is governed by the kinematic law of the standard slider-crank mechanism. The mechanical sub-model receives as input the angular velocity imposed by the driver (typically constant) and the machine geometry (stroke, connecting rod length, bore, and rod diameter). Inside the chamber, the gas undergoes polytropic transformations, exchanging mass with the exterior through valve ports and heat with the cylinder walls. The model explicitly accounts for the clearance volume, which is fundamental for determining volumetric efficiency and the slope of the re-expansion curve.

Figure 21 illustrates the schematic representation of the cylinder thermodynamics coupled with the crank mechanism block used in the simulation environment.

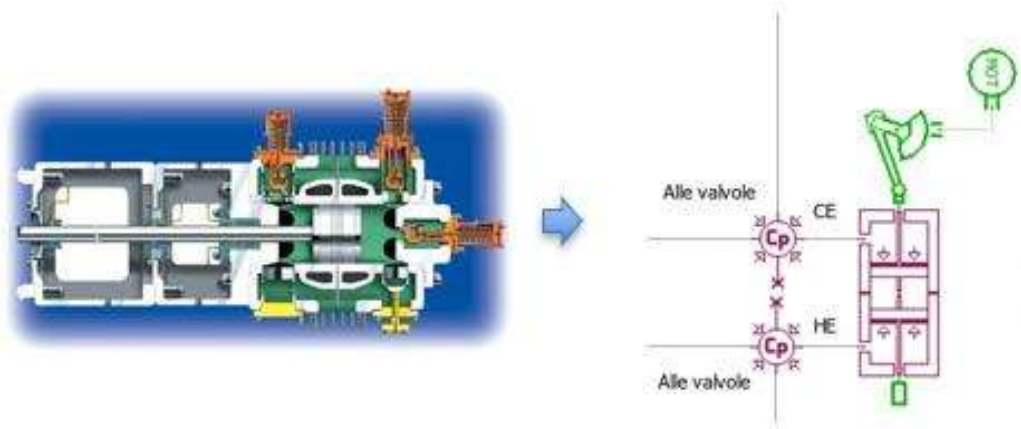


Figure 21: Cylinder sub-model

Suction and discharge valves represent the non-linear coupling element between the cylinder and the piping. Each valve is modeled as a single degree of freedom mass-spring system. The equation of motion ((14) solves the instantaneous position of the sealing element (plate or poppet) by balancing the differential pressure ($S\Delta p$), the gas drag force (F_{drag}), the spring restoring force (kx), and the inertia of the moving mass (ma).

$$ma = S\Delta p + F_{\text{drag}} - kx \quad (14)$$

The effective flow area varies as a function of the calculated instantaneous lift. This allows the model to capture complex phenomena such as delayed closing or valve flutter. The translation from the physical valve cross-section to the dynamic lumped-parameter model is depicted in Figure 22.

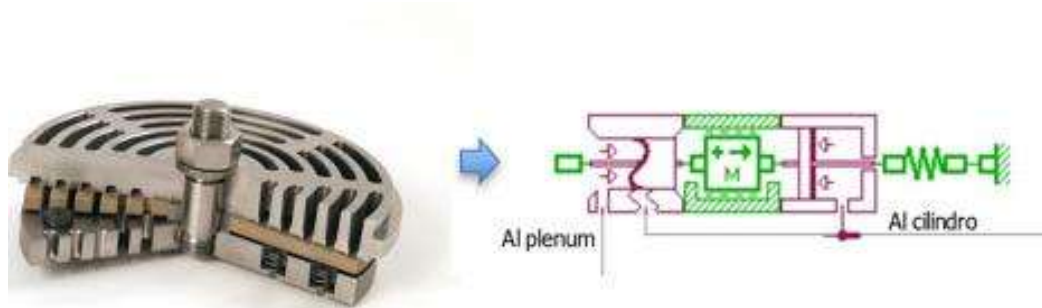


Figure 22: Valve sub-model

Geometrically complex volumes, such as internal cylinder chambers (plenums/manifolds) and pulsation dampener bottles, cannot be geometrically represented in a 1D code. Therefore, the equivalent duct method is adopted. The 3D volume is discretized into a network of pipes and lumped volumes that trace the mean flow path (streamline). Dimensions and connections are calibrated to preserve the original acoustic impedance (inductance and compliance) of the component. For volume-choke-volume dampeners, the

choke tube and internal baffles are modelled to replicate the low-pass filtering effect. Figure 23 and Figure 24 demonstrate the decomposition of a typical plenum and pulsation dampener, respectively, into its equivalent network of 1D elements.



Figure 23: Plenum sub-model

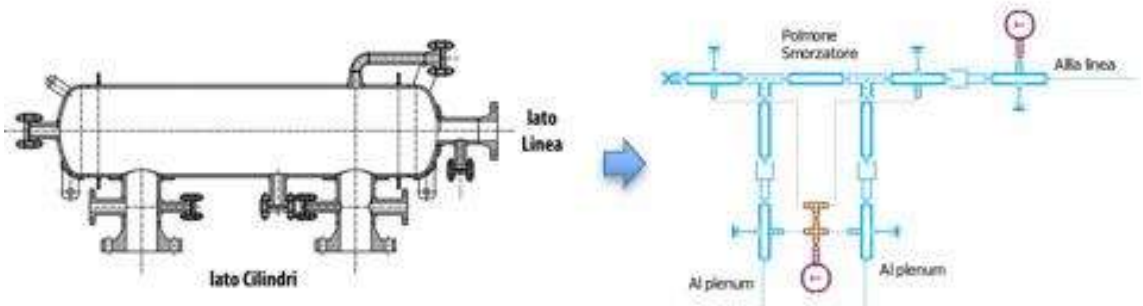


Figure 24: Pulsation damper sub-model

The modelling of tube-bundle heat exchangers utilizes a specific approach to capture both pressure drops and gas density variations. The component is schematized as a flow division into parallel channels (flow-parallel split). The exchanger headers introduce concentrated pressure losses due to sudden cross-sectional changes. Inside the tubes, the fluid undergoes distributed losses due to skin friction and density changes due to heat transfer. The schematic of the heat exchanger model is shown in Figure 25.

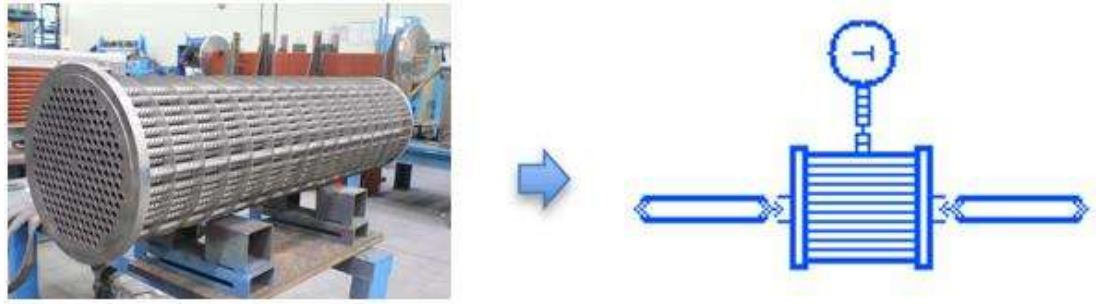


Figure 25: Cooler sub-model

The piping network constitutes the transmission medium for pressure waves. The model topology is constructed by faithfully importing coordinates and dimensions from the plant's available isometric drawings. Each pipe section is subdivided into finite control volumes to solve the 1D Navier-Stokes equations. The model includes frequency-dependent friction terms to correctly simulate the damping of the pulsating wave over long distances. Elbows, T-junctions, and reducers are computed with the CFD1D method elements, and they provide the application surfaces for the calculation of unbalanced shaking forces.

A distinctive feature of reciprocating compressors, which substantially differentiates them from turbomachinery and internal combustion engines, is the acoustic separation of the plant sections. During the compression and expansion phases, the suction and discharge valves remain closed, physically isolating the suction environment from the discharge one. Consequently, pulsation phenomena developing on the suction side do not directly influence those on the discharge side, and vice versa. This characteristic allows for the analysis and optimization of the two plant sections in a partially decoupled manner, simplifying the identification of critical issues.

Before proceeding to the operational simulation with the compressor forcing function, a fundamental preliminary analysis is performed to characterize the "passive" frequency response of the plant. In this phase, the system is excited by introducing a white noise signal at the cylinder flanges. This signal, characterized by uniform energy content across the entire frequency spectrum, simultaneously excites all acoustic modes of the piping. Analyzing the response spectrum allows for the identification of the plant's acoustic natural frequencies, highlighting which frequencies are amplified by the geometry and which are dampened (Figure 26).

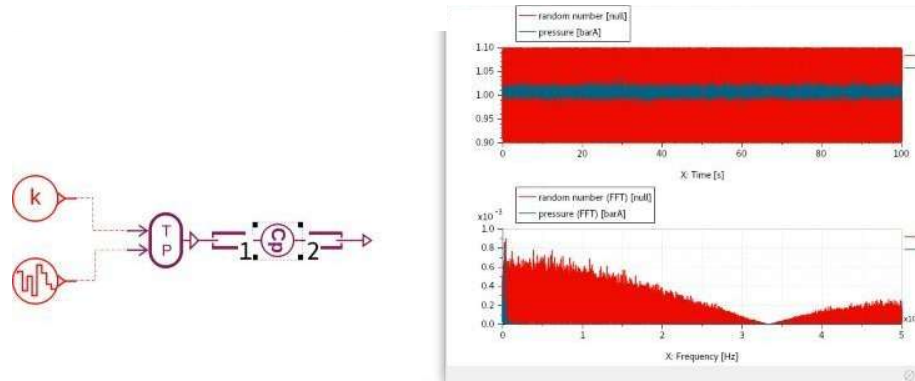


Figure 26: White noise analysis example

Comparing these natural frequencies with the harmonics of the driver rotational speed is the critical first step to identify potential resonances even before simulating the machine's operation.

FEM: mechanical assessment

The mechanical assessment of the piping system was conducted using the SolidWorks ecosystem. This choice was driven by the strategic advantage of operating within a unified environment that integrates both Computer-Aided Design (CAD) and Finite Element Analysis (FEA) capabilities (SolidWorks Simulation). This seamless integration ensures that any geometric modification required for optimization, such as shortening a span, modifying a constraint, or adding a stiffener, is immediately reflected in the simulation model without the data loss or translation errors typical of exporting files to external solvers.

Unlike long transmission pipelines, the piping of industrial reciprocating compressors' skids is characterized by complex, compact geometries with low slenderness ratios. Under these conditions, simplified 1D beam approximations are insufficient to capture the true dynamic behaviour and local stress concentrations. Therefore, a full 3D finite element approach was adopted.

To balance geometric fidelity with computational efficiency, the piping is discretized using shell elements. A specific methodological optimization was developed and implemented regarding the generation of the simulation geometry. Conventionally, the piping was modelled as a full solid body. The analyst must then manually select the faces to convert them into shells or extract mid-surfaces, a process that is time-consuming and prone to selection errors. To streamline this workflow, a dedicated parametric component library was developed. This tool generates the piping geometry directly as external surface bodies based on 2D routing sketches and 3D CAD inputs. An example of the two different libraries is displayed in Figure 27. In the meshing phase, the finite element mesh is generated directly on these surfaces. The physical wall thickness (derived

from the pipe schedule) is assigned as a shell parameter, and the software computes the element stiffness by projecting the thickness inwards from the outer surface. This procedure eliminates the redundant solid modelling step, significantly reducing file size and pre-processing time while ensuring a high-quality mesh that accurately represents the mass and stiffness distribution.

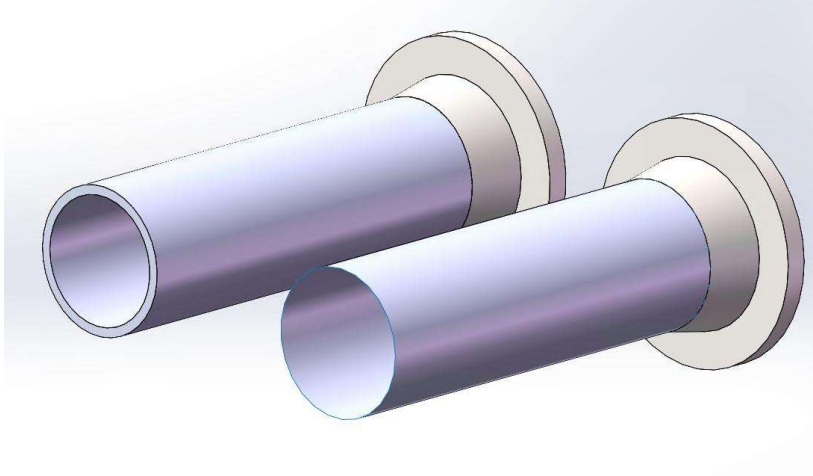


Figure 27: Comparison between solid body and shell pipe libraries

The finite element model is completed by assembling all the components and defining boundary conditions. Standardized heavy components such as flanges, valves, and specific fittings are inserted using pre-existing 3D libraries. Depending on the analysis requirements, these are meshed as solid elements or treated as rigid masses connected to the piping shell nodes to correctly simulate their inertial contribution.

The support system is modelled based on the available data. Where detailed 3D geometry of the carpentry is available, it is included in the analysis to capture the real structural compliance. In cases where the support geometry is not yet finalized, constraints are applied directly to the pipe surfaces (or specific split-lines created on the surface), restricting the relevant degrees of freedom to simulate the interaction with guides, hold-downs, or resting supports.

The mesh strategy is governed by a rigorous dynamic length-scale criterion, and a element distortion check to ensure solution independence. The discretization strategy follows two fundamental rules. To accurately capture the mode shapes, the global element size is dimensioned to ensure a minimum of 6 elements per acoustic wavelength at the maximum frequency of interest (typically the 10th harmonic of the compressor speed). At stress concentration points (e.g., nozzle intersections, tee junctions), the mesh is locally refined. The validity of the refinement is confirmed when a further reduction in element size results in a variation of the peak stress result of less than 5%. Post-generation, the

mesh topology undergoes a quality compliance check to prevent numerical errors associated with element distortion. Key shape metrics are monitored against solver thresholds to ensure the proper conditioning of the stiffness matrix. Aspect ratio values are controlled to remain below a limit of 5:1 in the global domain and 3:1 in high-stress regions to avoid shear-locking phenomena. Then, skewness and Jacobian ratio are verified to ensure that the mapping between the element's local coordinate system and the global system remains valid, particularly for elements mapped onto high-curvature surfaces (elbows). Lastly, warpage is monitored to ensure that the four nodes of the shell elements remain coplanar within a tolerance of 5-10 degrees, preventing artificial stiffening during bending deformation.

Before applying any dynamic load, a modal analysis is performed on the constrained geometry. This calculation solves the eigenvalue problem to identify the mechanical natural frequencies and the associated mode shapes of the piping system. The objective of this phase is to verify the separation margin. According to API 618 guidelines, the mechanical natural frequencies must be separated from the significant excitation harmonics of the compressor by a minimum safety percentage. If a resonance is identified (i.e., a natural frequency falls within the exclusion band) the support layout is modified before proceeding to the forced response analysis.

Once the modal behaviour is verified, the Harmonic Response Analysis is executed. The unbalanced shaking forces calculated in the 1D-CFD analysis are applied as dynamic loads at the piping discontinuities (elbows, tees, closed ends). The primary metric for validating the system design is the vibration velocity, measured in mm/s. In accordance with API 618 and ISO standards, vibration velocity is the most reliable indicator of fatigue risk for piping systems operating in the frequency range typical of reciprocating compressors. The simulation results are compared against the allowable screening curves.

While dynamic stress and displacement are calculated, they are considered secondary indicators. Stress evaluation becomes critical only if velocity limits are exceeded or in the presence of stress concentration factors not fully captured by the velocity criterion. Displacement is monitored primarily to ensure sufficient clearance between vibrating pipes and surrounding structures.

Results and Discussion

The validation of the proposed workflow focuses on a first highly complex industrial application: a reciprocating compressor unit designed for pipeline evacuation services. This case study was selected due to the extreme variability of its operational

envelope, which poses a unique challenge to standard acoustic design methodologies. Unlike stationary process units operating at fixed nominal conditions, this machine must manage a transient suction pressure ranging from 0 to 55 bar and discharge pressures from 0 to 115 bar. Furthermore, to optimize efficiency across this wide thermodynamic range, the unit features a variable staging architecture: the cylinders can be switched between parallel and series configurations, effectively modifying the machine layout from a 1-stage to a 3-stage inter-refrigerated compressor. Additionally, the driver operates via a variable speed driver, allowing for continuous speed modulation. In this context, the multi-case analysis described in the methodology becomes mandatory: the acoustic response must be verified not just for a single design point, but across the entire permutation of staging configurations, pressures, and speeds.

A critical constraint for this application is the physical layout. Being a mobile unit mounted on a transportable trailer, the available footprint is strictly limited. This spatial constraint precludes the use of traditional reactive dampening strategies (i.e., increasing pulsation bottle volume), forcing the adoption of optimized dissipative strategies (acoustic resistance) to control resonances.

Upon completion of the 1D CFD simulations for all operational cases, the system's response was benchmarked against the acceptance criteria of API 618 (5th Edition). The standard imposes two distinct limits to ensure system integrity. To prevent interaction with valve dynamics, peak-to-peak pulsation levels ($\Delta p/p$ %) at the cylinder flange (Figure 28) must not exceed the lesser of 7% of average line pressure or the value derived from the pressure ratio (R):

$$\frac{\Delta p}{p} \% < 7\% \text{ or } 3R\% \quad (15)$$

To minimize shaking forces in the downstream piping, the pulsation level beyond the suppression device (bottle) must remain below the threshold defined by Equation (16):

$$\frac{\Delta p}{p} \% = \sqrt{\frac{a}{350}} \times \left(\frac{400}{(p_L \times D_I \times f)^{0.5}} \right) \quad (16)$$

Where:

- a is the speed of sound [m/s]
- p_L is the line pressure [bar]
- D_I is the pipe internal diameter [mm]
- f is the frequency [Hz]

The multi-case 1D-CFD analysis revealed a critical exceedance of the API 618 line limits in the suction inter-stage piping. Specifically, a distinct resonance was observed at the 2nd harmonic of the compressor running speed.

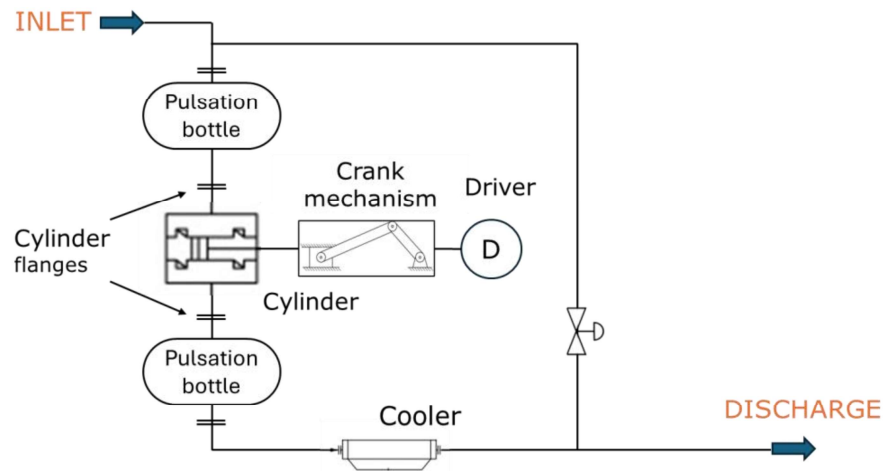


Figure 28: Reciprocating compressor PFD example

The calculations of the harmonic spectrum was performed for all the main operating cases considered, and the worst case was identified as the only one exceeding the prescribed limits (Figure 29). The coincidence between the excitation frequency and the piping acoustic natural frequency resulted in an amplification factor exceeding allowable limits. As previously noted, the simplified approach of damping the pulsation by increasing the bottle volume was not viable due to the mobile unit's dimensional constraints.

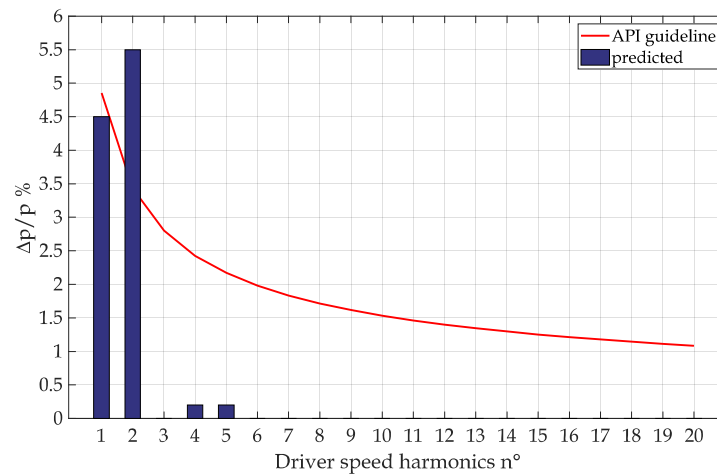


Figure 29: Pulsation harmonic spectrum before orifice introduction

To mitigate the resonance within the existing geometric envelope, a dissipative approach was adopted by introducing a restrictive orifice plate. The effectiveness of a resistive acoustic element is strictly dependent on its location within the standing wave

field. As highlighted by Eijk [128], acoustic damping is proportional to the dynamic pressure drop (Δp), which is a function of the square of the acoustic particle velocity (v_{ac}):

$$\Delta p = k \frac{1}{2} \rho v_{ac}^2 \quad (17)$$

Where:

- ρ is the gas density
- k is a loss coefficient

Therefore, to maximize damping efficiency, the orifice must be located at the resonance wave anti-nodes (a point of maximum particle velocity and minimum acoustic pressure fluctuation), as shown in the standing wave diagram in Figure 30. Placing the orifice at a pressure maximum would render it acoustically transparent and ineffective.

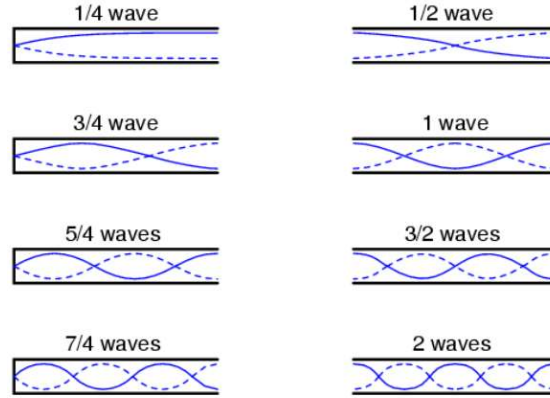


Figure 30: Pipe's resonance wave shape

Using the 1D-CFD results, the optimal orifice location in the piping system was identified. The analysis was repeated with the correctly sized orifice installed at the determined anti-node. The results confirmed that the introduced acoustic resistance successfully dampened the resonant peak, bringing the residual pulsation levels below the API 618 limits (Figure 31) for all operating cases without incurring excessive static pressure loss.

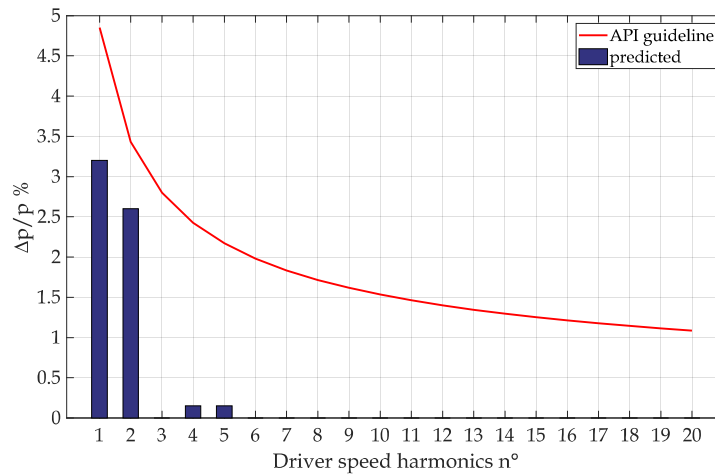


Figure 31: Pulsation harmonic spectrum after orifice introduction

The second application selected for validation concerns a carbon dioxide (CO₂) compressor. This specific service presents distinct modelling challenges due to the complexity of the piping system as the layout involves a complex network of inter-stage coolers and auxiliary lines. Additionally, detailed data regarding the structural supports geometry were available, allowing for higher fidelity.

Consequently, this project served as the pilot application for the development and validation of the parametric shell library described in the methods section. The adoption of the automated shell-generation workflow allowed for a reduction in pre-processing time by around 30% compared to standard solid modelling, ensuring mesh convergence at critical stress locations while maintaining a manageable computational cost.

Once the unbalanced shaking forces were computed via the multi-case 1D-CFD analysis, the forced harmonic response of the piping system was evaluated. Unlike the previous case which focused on API 618 pulsation limits, this study focused on the resulting structural vibration velocity. The assessment utilized the ISO 20816-8 standard, which defines specific acceptance zones based on RMS velocity values for each direction (X,Y,Z), as detailed in Table 6.

Table 6: ISO 20816-8 vibration limits

Vibration			
Zone	Velocity [mm/s RMS]	Criteria	Description
A	≤ 12.7	Acceptable	Compressor systems with vibration within these zones are normally considered acceptable for long-term operation
B	$12.7 < x \leq 19.0$		
C	$19.0 < x \leq 28.5$	Marginal	Analysis and possible correction to be undertaken. Clarify between the vendor and the purchaser that the compressor is suitable for long-term safe operation.
D	> 28.5	Unacceptable	Urgent correction to be performed or shutdown to be considered.

The FEM simulation results confirmed the robustness of the design optimization.

As illustrated in the directional contour plots (Figure 32Figure 34), the maximum vibration velocity recorded on the piping system remained below 12.7 mm/s RMS in all three principal axes (X, Y, Z). The same compliance is observed for the resultant velocity vector (Figure 35). Consequently, the system falls entirely within zone A, ensuring long-term operation without fatigue risk.

Finally, the study provided a crucial cross-check between the modal analysis (mode shapes) and the dynamic response. A comparison between the response peak in Figure 35 and the mode shapes in Figures 35 and 36 demonstrates the correlation between the mechanical natural frequencies of the piping and the peak dynamic response. The alignment of the shell-element model's prediction with the expected stiffness characteristics confirms that the formulation correctly captures the mass-stiffness distribution of the system, validating the library for future compressor applications.

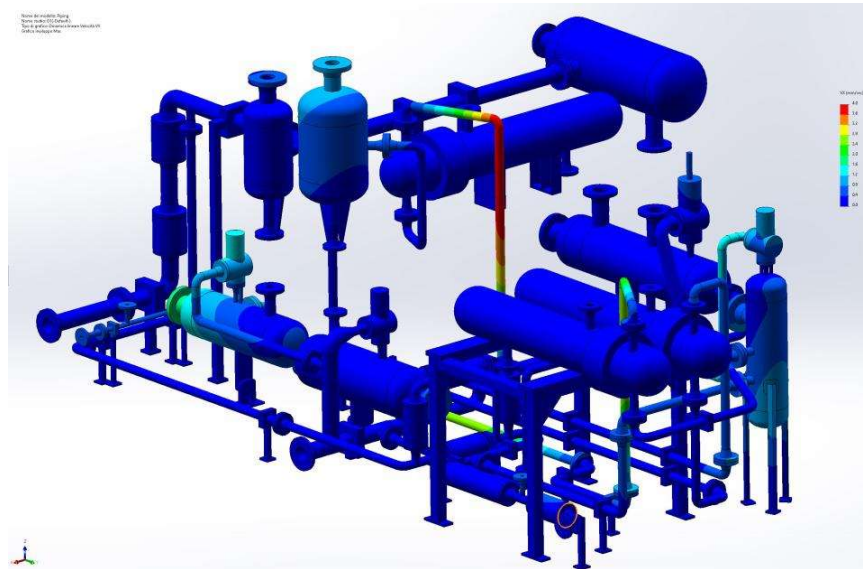


Figure 32: Contour plot of X velocity

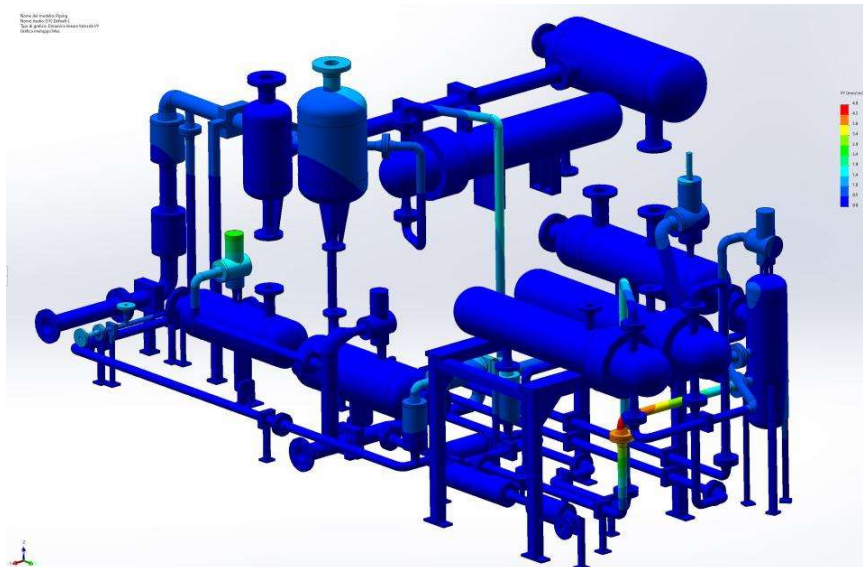


Figure 33: Contour plot of Y velocity

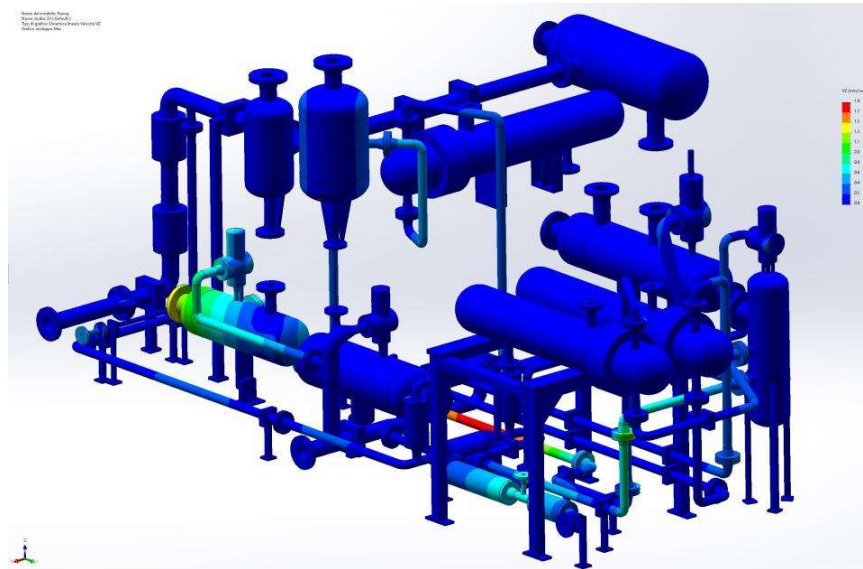


Figure 34: Contour plot of Z velocity

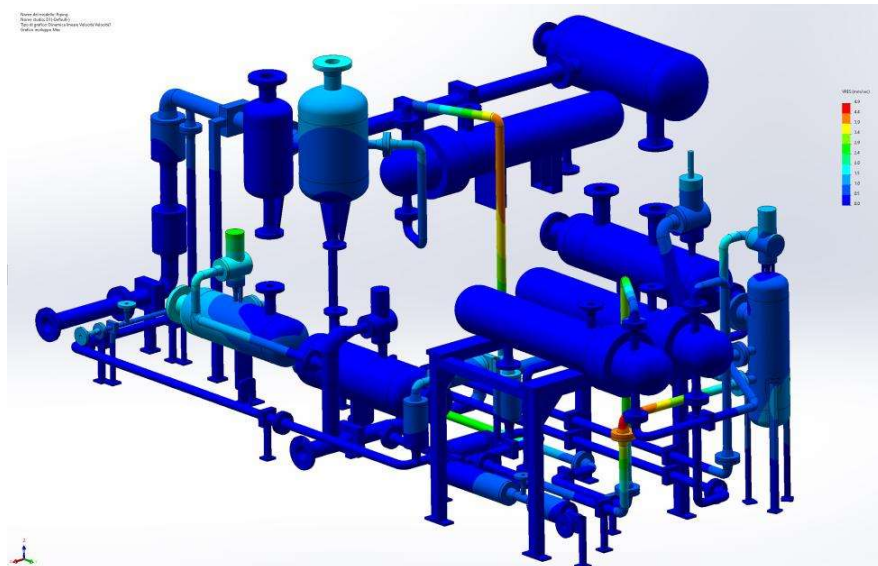


Figure 35: Contour plot of resultant velocity

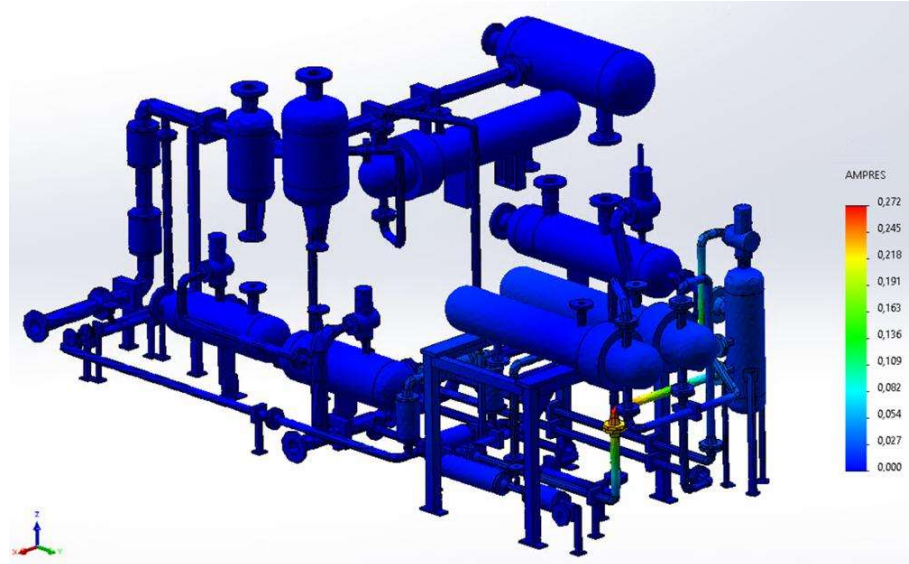


Figure 36: System's third mode shape

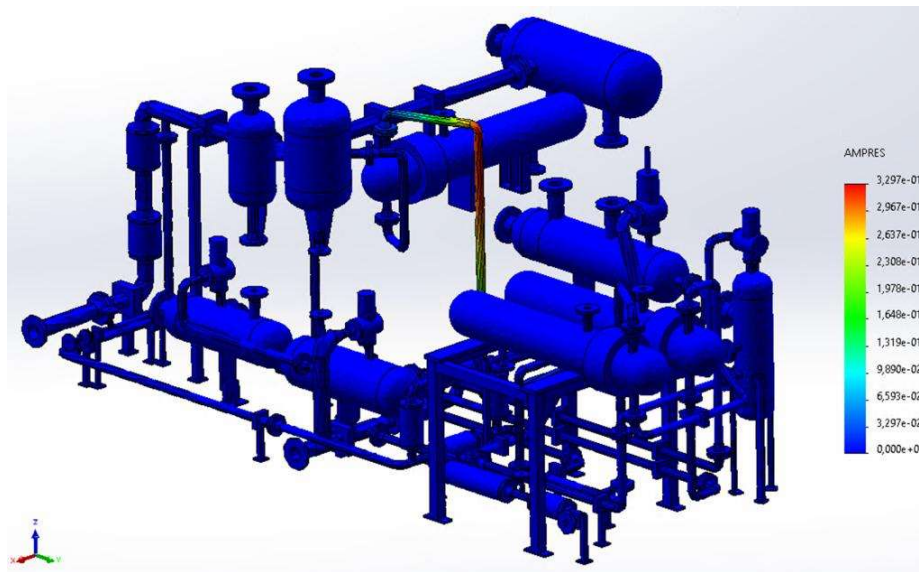


Figure 37: System's fifth mode shape

Conclusions

The research presented in this section focused on the critical analysis and optimization of the design workflow for reciprocating compressor systems, addressing the inherent risks of flow-induced vibrations and fatigue failure. By integrating 1D Computational Fluid Dynamics (Acoustic) with 3D Finite Element Analysis (Mechanical), the study established a robust framework for meeting the stringent requirements of API 618 and 688 standards (Design Approach 3).

Analysis The investigation into the mobile pipeline evacuation unit demonstrated that the traditional single-point design verification is inadequate for modern industrial applications. The complexity of the operational envelope, characterized by wide pressure ranges (0-115 bar), variable speed, and dynamic staging, revealed that acoustic resonances are often latent phenomena that manifest only under specific thermodynamic combinations. The identification of a critical resonance, which appeared only in specific load configurations, validated the adoption of the multi-case envelope strategy. Furthermore, the study proved that when physical constraints preclude the use of reactive damping (volume), dissipative impedance control becomes the primary optimization lever. The successful application of a resistive orifice plate, positioned at the velocity anti-node identified via 1D-CFD, confirmed that pulsation control is strictly a matter of spectral impedance matching rather than pulsation damper volume sizing.

From a structural perspective, the research addressed the trade-off between geometric fidelity and computational efficiency in finite element analysis. The development and validation of the parametric shell element library represents a significant methodological advancement for the analysis of compact compressor skids. The application to the CO₂ compressor demonstrated that the shell formulation provides a reduction in pre-processing time of approximately 30% without compromising solution accuracy. The strong correlation observed between the predicted mechanical natural frequencies (mode shapes) and the forced dynamic response confirms that the optimized shell discretization correctly captures the mass-stiffness distribution of complex piping networks.

Ultimately, the comparison of results against API 618 pulsation limits and ISO 20816-8 vibration criteria underscores that acoustic and mechanical analyses cannot be treated in isolation. A system may be compliant with pulsation limits yet fail structurally due to local mechanical resonances, or conversely, exhibit high pulsations that do not translate into damaging vibrations due to high structural stiffness. The integrated workflow presented herein provides a deterministic path to decouple these interactions. By rigorously defining mesh quality criteria and validating the solver against field-representative case studies, this thesis demonstrates that the design approach 3 is not merely a regulatory requirement, but an essential engineering tool to guarantee the long-term reliability of reciprocating machinery under the increasingly demanding conditions of the energy sector.

Part III: Progressive dividers

Introduction

Reciprocating compressors require accurate lubricant delivery, depending on the facility's process and the design strategy. In most applications, it is important to deliver the correct amount of lubricant fluid to different points. On one side, insufficient lubrication causes increased wear on the components. On the other side, if too much lubricant is used, it can contaminate the processed gas. According to the current technologies, two systems are used to achieve the correct delivery of lubricant fluid: the former is the Pump to Point (PTP) system, the latter is the Progressive Divider (PD) system. The first one uses independent lines based on a volumetric pump for each lubrication point, allowing for flow rate adjustments on each delivery line. The second one is based on a volumetric flow divider, that splits the fluid delivered from a centralized pump to the different points, depending on its configuration.

Progressive dividers allow to centralize the feeding unit, keeping independent the delivery lines. As described further on, the distribution of the flow rate is carried out thanks to the volumetric layout of the modules that form the whole system. Currently, the literature reports very few contributions about progressive dividers. Maffei et al. [97] describe the development of a hybrid lubrication system used for a hydrogen reciprocating compressor. This system uses the pre-existing PTP system together with a new PD one. Sun et al. [98] study an oil mist PD system and develop a simulation model within Amesim environment. Veprek [99] uses genetic algorithms to assist the design process of a PD, focusing the best configuration for the modules of a distributor to fulfil the lubrication requirements. Other material is available at PD producers websites, such as [95, 96]. The current article is aimed at analyzing a progressive divider operating under the proper conditions of reciprocating compressors (6). Based on lumped-parameter approach (7), a mechanical-hydraulic model of the PD is built in Amesim environment and validated thanks to specific experimental activities. The model allows to simulate the PD system, providing primary information on its operation.

Materials and Methods

The PD is based on a modular layout that is typically tailored on the system requirements in terms of volume delivery per cycle. Figure 1 illustrates how a three-module PD operates. In the first phase, the lubricant flows from the input section, it goes through the end section and moves the first piston (or spool) to the right. During its stroke, the spool displaces a specific volume of fluid towards the delivery point number 5. In the

second phase, the fluid moves the second spool to the right through the input section, feeding point number 2. In the third phase, the fluid moves the third spool to the right, through the medium section, feeding point number 4. In the fourth phase, the fluid moves the first spool to the left, which feed the point number 6. In the fifth phase, the fluid moves the second spool to the left, feeding point number 1. Finally, in the sixth phase, the lubricant moves the third spool to the left, feeding point number 3. After that, a new cycle starts. As the PD is a volumetric device, the cycle frequency is determined by the flow rate at inlet port. On each delivery line the flow rate is discontinuous. Since the delivered amount on each line depends on the swept volume of each spool, at industrial level the fluid flow rate is indirectly checked by counting the number of cycles in a time window.

The current activities are referred to a commercial PD that is configured to serve a reciprocating compressor. The considered PD is set-up on three sub-systems. The first one is made of an inlet-module, that is provided with the inlet hydraulic connection. In the current configuration, the two delivery lines are connected to serve a single line (single-ended). This module is equipped with a Namur proximity sensor that allows the detection of the spool state, enabling the time count of the PD cycles. The last sub-section is the module that closes the circuit. The module is here single-ended, and it is provided with a visual indicator that is magnetically driven by the spool, to quickly check the operation of the PD. Two single-ended and one double-ended identical modules form the intermediate sub-system, providing four delivery lines. Two module sizes are used to build the PD under investigation. The biggest is 0.60 cc/stroke and it is used as the inlet sub-systems. All the other modules are smaller, 0.04 cc/stroke. The internal layout of the PD has been carefully inspected. Spool diameters have been measured, whereas internal passages and cavities have been indirectly measured through silicone molds. The measured geometry has been compared to the 3D drawings available on the website of the PD producer.

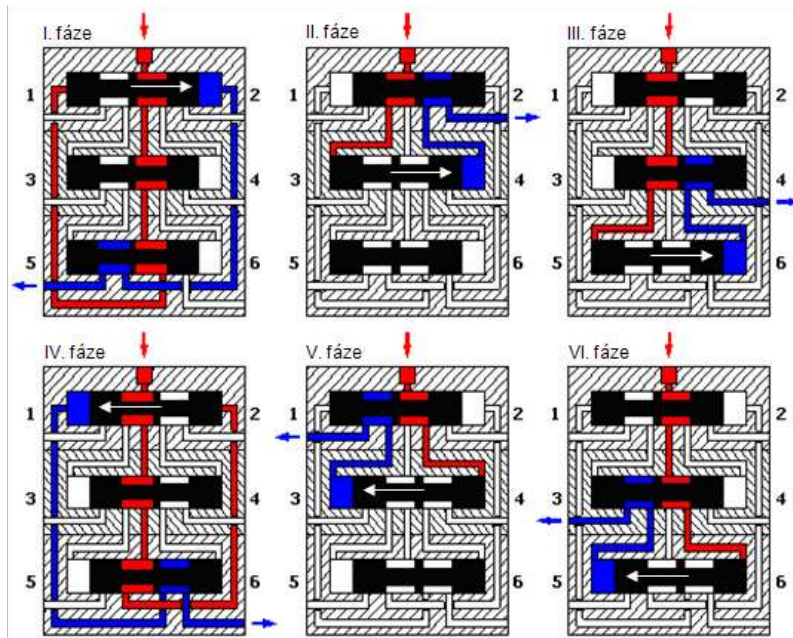


Figure 38: Pressure divider operating scheme

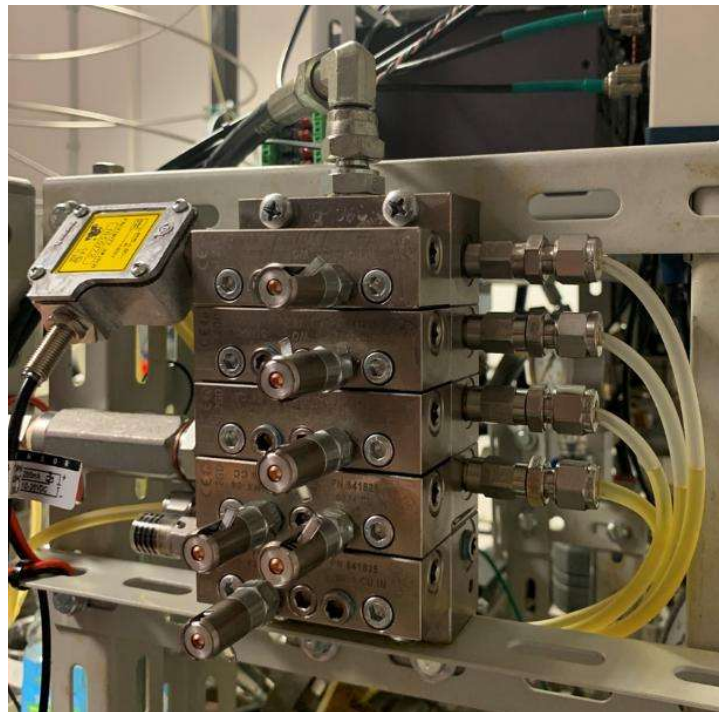


Figure 39: View of the progressive divider under investigation

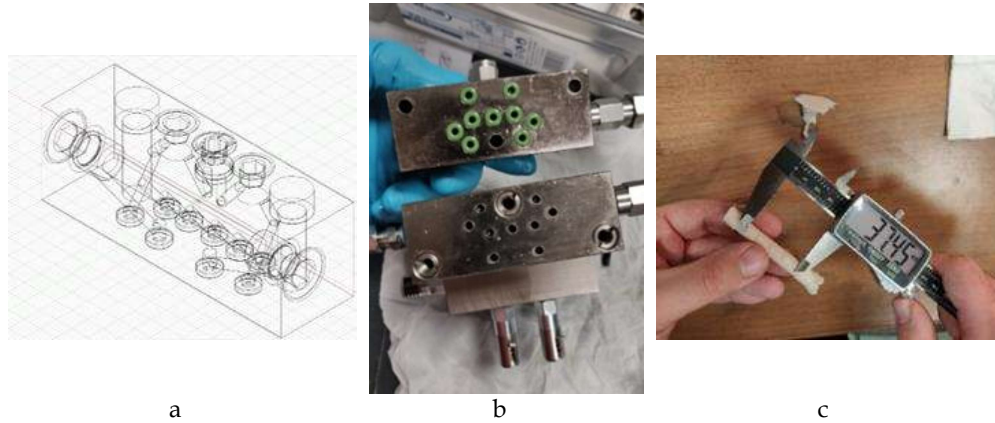


Figure 40: CAD drawings of the PD module (a), two separated modules (b), silicone molds inspection (c)

Experimental setup

The experimental setup is realized at Fluid Power Laboratory of DIIEM – Roma TRE University in Rome. The hydraulic system is based on a volumetric pump (A), a safety relief valve (B) and a flow-control valve (C). Fluid is delivered to the divider (D), which then splits it to the different delivery points. As reported above the divider is set with five modules and six outlets. Each outlet has a spring-loaded check valve that prevents backflow. One of the outlets is equipped with a relief valve (E) that allows to unbalance the PD delivery lines setting a certain load-pressure. Four sensors measure the upstream volumetric flow rate (F), the upstream pressure (G), the pressure at the unbalanced outlet (H), the divider cycle frequency (I).

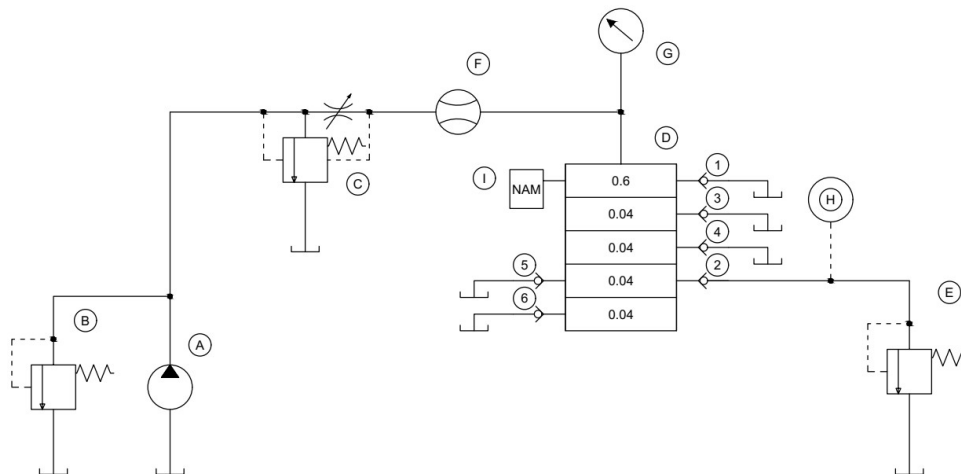


Figure 41: Sketch of the experimental set-up

The progressive divider is put into operation at fixed flow rate through the (C) valve. Alternatively, the feeding system can be easily configured to provide a constant pressure level at PD inlet. During the experiments, the inlet flow rate, the cycle frequency, and the pressure upstream of the divider are measured and acquired by a DAQ system. Fluid is thermoregulated at $30\pm 1^\circ\text{C}$. After a certain operating time, the fluid delivered at each line is gravimetrically measured, ensuring 0.5% accuracy in the worst case. The tests are repeated for different operating frequencies of the distributor, with all the lines at the same pressure and loading one of the lines through the valve (E).

Lumped parameter modelling

Based on the PD layout, a mechanical-hydraulic lumped parameter model has been built in Amesim environment. Figure 5 shows the model sketch developed for this study. The brown elements represent the hydraulic sub-models representing the PD spools, whereas the elements are the hydraulic connections, pressure and flow sources, valves (check valves and pressure relief valves), and leakages. The model considers the same operating logic as described in Figure 1, plus the leakages related to the clearances of the system. Table 1 reports the main parameters used in the model and the considered fluids. These cover a relatively wide viscosity range, depending on the specific tasks.

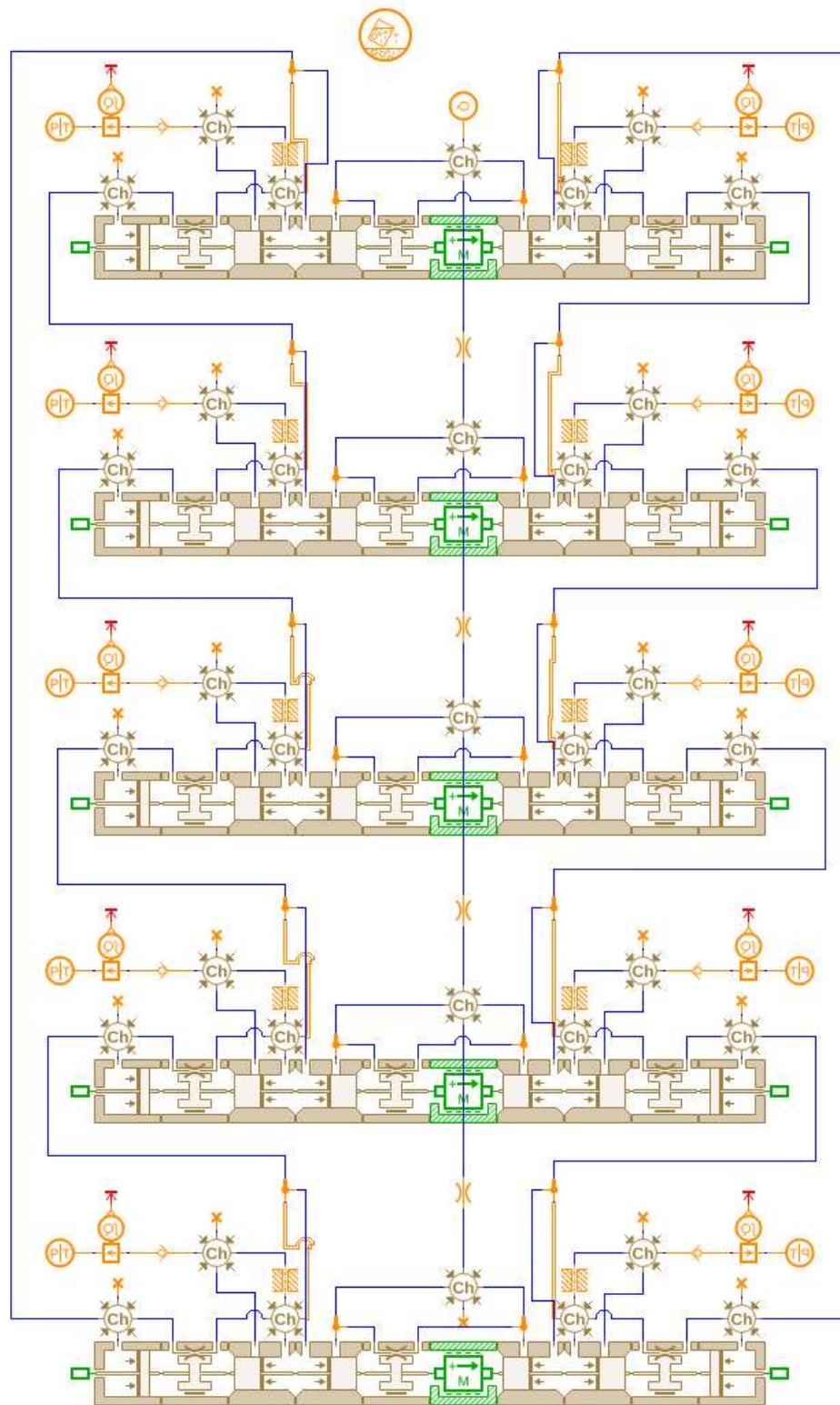


Figure 42: Sketch of the PD model in Amesim environment.

Table 7: Model parameters

	0.6 cc module	0.04 cc module
Parameter		
Spool mass	28 g	2 g
Highest displacement limit	4.25 mm	2.83 mm
Lowest displacement limit	-4.25 mm	-2.83 mm
Spool diameter	9.5 mm	3 mm
Spool rod diameter	7 mm	2 mm
Rod diameter of proximity chamber	2 mm (sensor)	0
Chamber length at zero displacement	7.8 mm	7.8 mm
Clearance on diameter	0.006 mm	0.005 mm
Fluids	Denomination	Viscosity @40°C
Reference fluid	ISO VG 46	38 cP
Low Viscosity Fluid	LVF	2.3 cP

Results

Pressure-Flow rate chart

The progressive distributor was subjected to an experimental investigation aimed at characterizing its hydraulic behaviour. The PD is a volumetric device, capable of processing a well-defined volume of fluid for each operating cycle. In the tested configuration, the processed volume is equal to 1.52 cm³/cycle. Therefore, the PD works at the operating frequency that is necessary to process the input flow rate. The pressure level upstream the PD depends on the pressurization level of the delivery lines and on the mechanical-hydraulic behaviour of the divider components. Keeping all the delivery lines at the same pressure (atmospheric), the pressure upstream of the divider depends only on the behaviour of the spools and on the internal flow features. Figure 6 shows the trend of the flow rate measured through the PD as a function of the supply pressure. Feeding the PD with an adjustable pressure source, it is observed that the system starts operating when a threshold pressure level is overcome. Once started, the system requires a pressure increase that varies linearly with the flow rate. The linearity of the pressure-flow characteristic suggests that the operation of the PD spools is mainly governed by viscous friction phenomena. This is in agreement with what is reported in [98], in which a similar system operates with air-oil mist. As regards the minimum pressure level, it has been addressed to the spring-loaded check valves at the outlet of the delivery lines, which

require the establishment of a certain force to overcome the preload of the spring-loaded poppets. The mechanical-hydraulic model built in Amesim proves to be able to correctly reproduce the behaviour of the real system. In accordance with what has been highlighted above, the only parameter that has been tuned in the light of the experiments is the viscous friction coefficient affecting the spools. Figure 6 shows the good agreement between simulated and experimental pressure-flow rate trends.

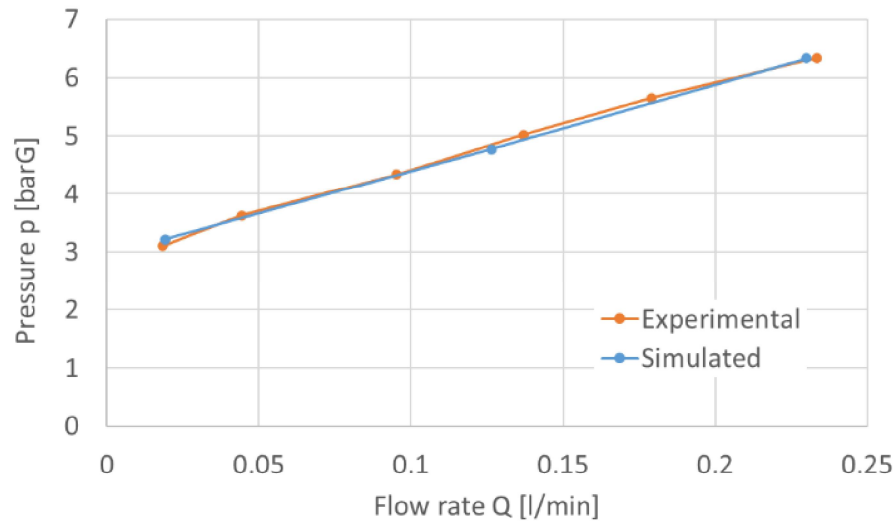


Figure 43: Pressure-flow rate chart – PD model validation

Flow rate divider

Once the hydraulic behaviour of the PD is obtained, the experimental activities are carried out to investigate the flow distribution among the delivery lines. Three cycle frequencies are considered, in two operating modes (balanced and unbalanced delivery lines). Experiments point out that the progressive distributor can operate in two different ways, depending on whether the delivery lines are balanced or not. In the case of balanced lines, the pressure upstream of the PD is essentially given by the combination of the load on delivery lines and of the viscous friction on the spools. Close agreement is found between experimental and simulated results (deviations below 0.5%).

In the case of unbalanced lines, the supply pressure changes within the operating cycle, according to the load level of each line. In the field of reciprocating compressors, significant pressure differences are expected among the various delivery lines, depending on the lubrication points, which include atmospheric (or low-pressure) zones (e.g., crank mechanism) and other zones that encounter the process fluid. Thus, the most pressurized delivery line raises the pressure throughout the pump delivery volume. In this volume (or capacity), pressurized fluid accumulates. Therefore, the other less loaded lines may

continue running at high upstream pressure. When high pressure drops arise between supply and delivery, the spool displacement happens at high speed, in a small time transient that is governed by the fluid discharge, from pump delivery ambient to delivery line. In real applications, the volume of the pump delivery ambient easily exceeds the fluid volume that is handled in a single operating cycle of the PD. Therefore, by simply unbalancing one of the delivery lines, the flow distributor is very prone to operate in a "characteristic" way: the cycle frequency is always imposed by the average pump flow rate, but the periods of the sub-phases are significantly altered. The most loaded module switches "slowly", whereas the other unloaded modules switch quickly in sequence since the spool viscous friction is far too weak to control the speed of the unloaded line spools. In these operating conditions, PD become noisy, and the high-speed switching of the spools promotes the onset of significant pressure pulsations downstream the delivery lines.

Figure 43 shows the experimental fluid distribution among the six delivery lines, for three inlet flow rate levels (or cycle frequency). It can be observed that the distribution of the flow rate over the various branches is not significantly affected by the frequency of operation. On the other hand, the results referred to the unbalanced condition (10 bar pressure at "2" line in Figure 4), show that the fluid distribution is altered. All the lines are influenced by the pressure unbalance. Line "2" and line "5", that are derived from the same hydraulic element, receive less fluid (up to 10% loss). Line "1" line receives more fluid (up to 2% gain). The other lines are affected by the unbalance, but there is low sensitivity to inlet flow rate (cycle frequency). At the highest flow rate (3 Hz), the lowest distribution alteration is found.



Figure 44: Measured fluid distribution

The mechanical-hydraulic model of the system captures the response to pressure unbalance of the progressive divider. Line “2” is set at 10 bar pressure. Figure 8 reports the flow distribution among the lines in terms of percentage difference, referred to the base case (all the lines at the same pressure). Good agreement is found between experiments and simulations. The model can qualitatively predict how the fluid is distributed among the lines and to quantitatively predict the extent of the unbalances. The model also gives the chance to explore the PD behaviour when LVFs (Low Viscosity Fluids) (recommended for specific uses) are required. Unbalances are increased and the way of the alteration among the six lines is unvaried, with lines “2” and “5” receiving less fluid.

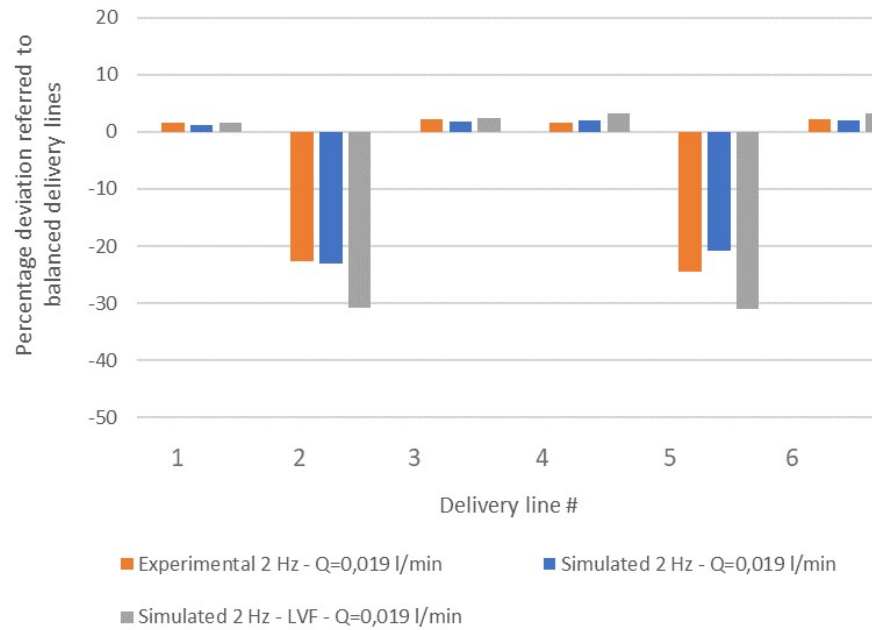


Figure 45: Measured and simulated flow distribution among delivery lines in terms of percentage deviation referred to the base case (balanced lines) – 10 bar pressure unbalances at “2” line

Conclusions

The experimental activities have outlined the behaviour of the progressive divider and have led to the set-up of a mechanical-hydraulic model under lumped parameter approach. The investigations have been devoted to exploring the PD behaviour, under the proper operating conditions in the field of reciprocating compressors. The linear relationship between supply pressure and flow rate has allowed to find the extent of viscous friction between the spools and the housings. When pressure imbalance affects the delivery lines, the peculiar character of the progressive distributor has emerged. On one side, the switching cycle of the spools is affected by the pressure transients that arise in the pump delivery environment, altering the characteristic periods of the sub-phases. On the other side, the distribution of the fluid is affected by the line pressure imbalance, with alterations in the order of a few tens of percentage points. The mechanical-hydraulic model, built based on the geometrical features of the PD, has proved to be a suitable tool for representing the behaviour of the system without tuning, both as regards the steady hydraulic characteristic and as regards its ability to model the distribution of the fluid. The effect of pressure unbalance is properly modelled. Once validated, the model proved to be useful for exploring the sensitivity of PD to the type of fluid, characterized by very low viscosity. The results, in this case, indicate that the fluid viscosity affects the fluid distribution in a limited extent.

Part IV: Turbomachinery material selection

1. Introduction

The selection of materials for turbomachinery constitutes a fundamental pillar of the design process, carrying direct implications not only for thermodynamic performance but also for component durability, production costs, and the feasibility of repairs.

To ensure a comprehensive and structured analysis, this study is organized into two distinct sections. While the primary focus of this thesis remains on centrifugal compressors, the analysis conducted in this section has been deliberately extended to include industrial steam and gas turbines. The initial part focuses exclusively on centrifugal compressors, addressing the specific challenges imposed by corrosive environments (such as sour gas applications).

Subsequently, the analysis extends to industrial steam and gas turbines. This expansion is motivated by the necessity to explore the spectrum of high-temperature applications, thereby evaluating advanced material classes, such as superalloys, which are standard in turbine engineering but offer superior solutions for special or extreme applications within the compression field as well.

This specific research activity stems from a critical operational requirement identified within the context of Service and Reverse Engineering: addressing the intrinsic limitations of rapid diagnostic tools commonly available in the workshop. The instrument for on-site material verification is Positive Material Identification (PMI) via X-ray fluorescence. However, this technology presents significant blind spots that can induce critical errors during the repair or component replacement phases. A major limitation lies in the superficial nature of the analysis; when components are treated with anti-wear coatings or thermal barriers, the instrument detects the composition of the coating rather than the base substrate. Furthermore, PMI provides elemental chemical composition but remains completely blind to the microstructural state of the alloy. Consequently, it is incapable of distinguishing between a raw material and one that has undergone essential heat treatments, such as quenching and tempering, solubilization, or precipitation hardening, which are decisive for ensuring the mechanical properties required for safe operation.

The primary objective of this work is to bridge the gap between these instrumental limitations and rigorous engineering requirements. The development of advanced competencies in metallurgy and heat treatment strategies serves to provide robust scientific support to the practical experience consolidated within the workshop, significantly reducing the risks associated with material procurement. This approach

allows the engineering team to move beyond a passive replacement strategy, where obsolete materials are simply replicated. Instead, it enables the proposal of modern solutions and material upgrades that enhance the performance and longevity of the repaired components. Finally, the codification of this knowledge into a dedicated technical manual, specifically developed for Blackstone Industrial, aims to create a standardized technical language. This initiative is designed to streamline the flow of information between the engineering department and the manufacturing workshop, ensuring that complex metallurgical specifications are correctly interpreted and implemented, thereby optimizing the overall production efficiency.

2. Centrifugal compressors

Key Factors in Centrifugal Compressors Material Selection

The selection of the appropriate material for centrifugal compressor components is the synthesis of a multi-variable optimization process [100]. According to industry literature and standards, this process is governed by the interdependence of four primary factors, whose combined evaluation determines the suitability of a metal alloy for a specific application. These factors are:

- Operative fluid and environment
- Temperatures
- Manufacturing
- Mechanical stresses

The analysis of these factors allows for the definition of a set of "target" physical and mechanical properties that guide the selection of the alloy and heat treatment. The main properties considered during design are summarized in Table 8.

Table 8: Material properties for turbomachinery design [100]

Category	Property	Relevance in Design
Mechanical	Tensile / yield strength	Resistance to centrifugal and pressure loads.
	Module of elasticity	Defines stiffness, deformation under load, and natural frequencies.
	Fracture toughness	Resistance to crack propagation; fundamental to preventing brittle fracture.
	Fatigue strength	Ability to withstand cyclic loads (pulsations, vibrations).
	Damping Capacity	Dissipation of vibratory energy (critical for open impellers).
Technological	Hardenability	Ability to achieve the required hardness throughout the component thickness.
	Weldability	Suitability for joining without defects or embrittlement of the Heat Affected Zone (HAZ).

	Machinability	Ease of material removal to achieve tight dimensional tolerances.
Physical & Chemical	Thermal Expansion Coeff.	Influences operating clearances and interference fits at different temperatures.
	Corrosion Resistance	Resistance to chemical attack (general, pitting, SSC) in aggressive environments.
	Erosion Resistance	Resistance to mechanical wear due to solid particles or liquid droplets.

Operative fluid and environment

The characterization of the operative fluid and the specific environmental conditions constitute the primary filter in the material selection matrix. In industrial applications, centrifugal compressors rarely operate with inert media; conversely, they frequently handle gas mixtures containing a variety of corrosive agents. The chemical interaction between the metal surface and specific molecules, such as hydrogen (H_2), chlorine (Cl_2), hydrogen chloride (HCl), carbon dioxide (CO_2), and hydrogen sulfide (H_2S), determines the prevalent degradation mechanisms and, consequently, imposes stringent constraints on the alloy composition.

While agents such as carbon dioxide and chlorine primarily trigger general corrosion or pitting phenomena [102] – typically mitigated through the selection of stainless steels with an adequate Pitting Resistance Equivalent Number (PREN), the presence of hydrogen introduces a more insidious threat known as hydrogen embrittlement. This mechanism particularly affects high-strength steels and specific alloys, causing a drastic reduction in ductility that can lead to sudden, catastrophic brittle failures even at loads below the yield limit.

Within the broad phenomenology of hydrogen embrittlement, this analysis specifically deepens the understanding of Sulfide Stress Cracking (SSC), as it represents the most complex and severe manifestation of the phenomenon. SSC emerges from the synergistic interaction between Hydrogen Sulfide (H_2S) and liquid water. It is imperative to underscore that for SSC to occur, the presence of an aqueous phase is a necessary condition to facilitate the electrochemical reactions that promote atomic hydrogen uptake into the metal lattice; if water is present but remains exclusively in the vapor phase under all operating and standstill conditions, the phenomenon does not initiate.

The severity of SSC is heavily influenced by environmental variables. The risk is maximized under acidic conditions ($pH < 4.5$) and is particularly aggressive between room temperature and approximately $60^\circ C$. Above this thermal threshold, hydrogen diffusivity increases, allowing the gas to escape the lattice, which generally reduces the material's susceptibility to cracking.

To manage the risks associated with such aggressive environments, the material selection process must adhere to specific international regulations established by NACE (National Association of Corrosion Engineers), specifically NACE MR0175 / ISO 15156 and NACE MR0103. Although these standards were historically developed for the hydrocarbon industry, they are currently adopted as the technical reference also for "green" applications, such as geothermal energy generation, where hydrogen sulfide is naturally present in the process fluids.

While a comprehensive analysis of these standards is presented in a dedicated chapter in this document, it is essential at this stage to highlight their primary implication for mechanical design: the rigorous restriction of material hardness. Extensive empirical data has demonstrated that harder materials are significantly more susceptible to brittle fracture in sour environments. Consequently, compliance dictates that the base material and the Heat Affected Zones (HAZ) must generally maintain a hardness below 22 HRC (Rockwell C) or 250 HBW (Brinell). This requirement acts as a design constraint, often forcing the engineer to limit the maximum yield strength of the selected alloys to ensure resistance to environmental cracking.

A notable engineering exception to these strict limits is provided by API 617. The standard acknowledges that for between-bearing compressor shafts, the areas subjected to the highest mechanical stresses, such as impeller interference fits, are often isolated from direct exposure to the corrosive gas or protected by buffer gas systems. Therefore, the standard permits the shaft material to exceed NACE hardness limits in these specific zones, allowing priority to be given to mechanical strength and rotordynamic stability where environmental exposure is controlled.

Temperature

While the operating conditions of standard centrifugal compressors typically fall within a thermal range that does not pose critical metallurgical challenges for low-alloy steels, applications involving extreme temperatures, whether cryogenic or high-temperature service, impose severe constraints that significantly narrow the field of eligible materials. The thermal factor must be evaluated not only for its effect on mechanical strength but also for its influence on physical stability and fracture mechanisms.

In high-temperature scenarios, the primary degradation mechanisms shift from immediate mechanical overload to time-dependent phenomena. Materials operating above approximately 40% of their absolute melting temperature become susceptible to creep, a slow, continuous plastic deformation under constant stress. Furthermore, elevated temperatures accelerate surface oxidation and hot corrosion rates.

Under these conditions, standard carbon steels lose structural integrity. To maintain dimensional stability and mechanical strength, the selection process must shift toward turbine-grade materials. This often necessitates the adoption of Nickel-based superalloys or high-temperature austenitic stainless steels. These advanced alloys, which will be analysed in depth in the subsequent chapter dedicated to turbines, are engineered to resist thermal expansion issues that could compromise rotor-stator clearances and to withstand the long-term effects of high-temperature exposure.

Conversely, at cryogenic or sub-zero temperatures, the critical concern is the ductile-to-brittle transition. Most ferritic and martensitic steels exhibit a sharp decrease in ductility and toughness as the temperature drops below a specific threshold. This transition significantly increases the risk of sudden brittle fracture, where the material fails catastrophically without prior plastic deformation warnings. To mitigate this risk, material selection is governed by the impact strength (toughness) measured at the Minimum Design Metal Temperature (MDMT). Applications requiring reliable operation at cryogenic temperatures typically demand the exclusion of ferritic structures in favour of materials with a Face-Centered Cubic (FCC) crystal lattice, such as austenitic stainless steels (e.g., AISI 300 series) or specific nickel-based alloys. These materials retain high ductility and toughness even at temperatures approaching absolute zero, ensuring safety against brittle failure.

Manufacturing

The selection of the manufacturing technology is a critical decision in the design process, possessing a direct and reciprocal influence on material selection. The manufacturing method is not merely a means to achieve a geometric shape; it fundamentally alters the material's microstructure, defect distribution, and mechanical properties. Consequently, selecting the appropriate technique ensures that the final component satisfies the performance, durability, and economic requirements defined in the design phase. For critical rotating components such as compressor impellers, the industry primarily relies on three distinct manufacturing routes: casting, forging combined with machining, and fabrication via joining.

Casting techniques, particularly investment casting and the use of monobloc cores, are favored for their ability to produce complex aerodynamic shapes, including shrouded impellers with intricate internal channels that would be impossible to machine from a solid block. While cost-effective for series production, casting introduces specific challenges regarding porosity and the homogeneity of the grain structure, necessitating rigorous non-destructive testing (NDT) to rule out inclusions or voids.

Conversely, forging followed by machining represents the standard for high-performance applications. Forging ensures a consolidated, refined grain structure free from porosity, offering superior mechanical properties and fatigue resistance compared to cast equivalents. This method provides the highest dimensional accuracy and surface finish but is often limited by the geometric accessibility of the milling tools (e.g., for closed impellers).

Finally, Fabrication may involve joining separate hub and blade components. This more traditional approach typically utilizes welding, brazing, or riveting, and is often employed for impellers that are too large for single-piece forging or have geometries that preclude integral machining.

The fabrication of impellers through joining techniques requires rigorous metallurgical control. Processes such as welding introduce a localized thermal cycle that creates a Heat-Affected Zone (HAZ) adjacent to the weld bead. In this zone, the material's microstructure is altered, often leading to high residual stresses and locally increased hardness. These microstructural changes can severely compromise mechanical integrity and, as discussed above, increase susceptibility to environmental cracking (SSC) in sour gas applications. To mitigate these deleterious effects, post-weld heat treatments are mandated by API standards. These thermal cycles are essential to relieve residual tensile stresses generated during cooling and to temper the martensitic structures in the HAZ, thereby restoring the material's toughness and enhancing its fatigue resistance to safe operational levels.

The ultimate choice of manufacturing technology is governed by a multi-variable trade-off analysis. Geometric complexity is often the primary driver; highly twisted blades or shrouded designs may necessitate casting or brazing. Mechanical requirements also play a pivotal role; high-stress applications typically demand the superior material integrity of forgings. Furthermore, the risk of defects must be weighed against cost; while casting lowers production expenses for high volumes, the potential for defects requires extensive quality control. Finally, dimensional accuracy and surface finish requirements often favour machining from solid blocks, which eliminates the variability inherent in casting or welding distortions.

Mechanical stresses

Within the compressor assembly, the impeller represents the component subjected to the most severe and complex stress regime. Its structural integrity is determined by the superposition of three distinct loading categories: steady-state mechanical loads, thermally induced stresses, and dynamic vibratory stresses.

The primary stress baseline is defined by steady-state mechanical loads. Foremost among these are the centrifugal forces generated by rotation, which induce high tensile stresses, particularly at the impeller bore and blade roots. Simultaneously, the aerodynamic function of the component generates mean gas forces resulting from the pressure differential established across the blades and the shroud. Additionally, the mechanical assembly itself contributes to the static stress state; the shrink fit (or interference fit) required to secure the impeller to the shaft generates significant hoop stresses in the hub. The material must possess sufficient yield strength to accommodate this preload without undergoing plastic deformation, while maintaining enough toughness to prevent brittle fracture under the combined tensile load.

Although often secondary during standard stable operations, thermally induced stresses can become critical under specific process conditions. If a compressor operates in surge or prolonged (hot-gas) recycle mode, the resulting heat buildup leads to rapid thermal expansion of the components. This phenomenon can generate severe thermal gradients within the impeller body. A critical failure mode in this scenario is the loss of shrink fit: if the impeller hub expands significantly more than the shaft due to thermal accumulation, the interference pressure drops, potentially compromising the mechanical integrity of the rotor assembly and leading to instability.

Superimposed on these static loads are dynamic vibratory stresses, which are the primary drivers of high-cycle fatigue failure. This issue is particularly acute in open impeller designs, where the lack of a shroud reduces structural stiffness and damping. The critical failure mechanism involves the coincidence of the blade's natural frequency with a harmonic multiple of the operating speed (typically up to $8\times$ rotational speed). This risk is exacerbated in applications utilizing Variable Speed Drives (VSD), where the wide operating speed range increases the likelihood of crossing a resonance point. To mitigate these risks, the design process relies on advanced Finite Element Method (FEM) analysis to map the resonance modes. When potential resonance risks are identified, blade frequency derating, a specific geometric modification to shift the natural frequency, is applied to detune the system, ensuring that the vibratory stresses remain below the material's fatigue limit.

Mechanical Properties

The selection of the optimal alloy is ultimately a process of matching the operational constraints analysed in previous sections with the intrinsic physical and mechanical properties of the material. Each property acts as a specific design variable that influences the component's performance, safety margins, and manufacturability.

The foundational requirement for any rotating component is adequate Tensile and Yield Strength. These parameters define the material's ability to withstand the steady-state forces previously discussed, such as the immense centrifugal pull at the impeller tip and the hoop stresses generated by interference fits. However, static strength cannot be considered in isolation from Fatigue Strength, which defines the relation between cyclic stress and number of cycles before failure. Given the pulsating nature of aerodynamic forces and rotor vibrations, high-cycle fatigue resistance is essential to prevent failure over the machine's operational life.

Closely related to structural dynamics is the Modulus of Elasticity (Young's Modulus). This property defines the material's stiffness, governing the elastic deformation of the impeller under load. It plays a dual role: mechanically, it dictates the radial expansion of the impeller at speed, directly affecting the running clearances and shrink fit pressure, and dynamically, it is a primary variable in the equation determining the natural frequencies of the blades. For open impellers or variable-speed applications, Damping Capacity becomes equally critical. A material with high internal damping can absorb vibratory energy, significantly reducing the amplitude of resonant stresses and mitigating the risk of high-cycle fatigue failure.

Among all mechanical properties, Fracture Toughness is often regarded as the single most critical parameter for impeller safety. It represents the material's resistance to crack propagation under load. Unlike yield strength, which prevents plastic deformation, fracture toughness prevents catastrophic, brittle failure in the presence of inevitable microscopic defects or flaws. Consequently, the material selection and heat treatment strategy prioritize maximizing toughness to ensure a "leak-before-break" failure mode, providing a safety margin against sudden rupture even under impact or cyclic loading.

The thermal response of the machine is governed by the Coefficient of Thermal Expansion and Thermal Conductivity. A mismatch in expansion coefficients between mating components (e.g., impeller and shaft, or rotor and casing) can lead to severe issues, ranging from the loss of interference fits to the seizure of running clearances. High thermal conductivity aids in dissipating heat during transient operations, preventing localized hotspots that could degrade material properties. Additionally, Specific Heat influences the transient thermal inertia, determining how rapidly a component heats up during surge or recycle conditions.

From a manufacturing perspective, Hardenability and Weldability are decisive. It is crucial to distinguish hardenability, the ability to achieve a martensitic transformation to a significant depth, from simple surface hardness. This property dictates alloy selection based on component size; for instance, while AISI 41xx series steels are sufficient for

smaller diameters, larger shafts require the superior hardenability of AISI 43xx series to ensure consistent mechanical properties at the core. Similarly, weldability is a prerequisite for fabricated components, ensuring that hubs and blades can be joined without inducing cracking or porosity in the heat-affected zones.

Finally, the material's interaction with the environment is defined by its resistance to degradation. Corrosion Resistance is paramount in sour or wet environments to prevent oxidation and chemical attack. Erosion Resistance is essential for maintaining aerodynamic profiles in the presence of particulate-laden flows or liquid droplets. Coefficient of Friction influences the wear behaviour at bearing interfaces and seals. In specific applications utilizing non-metallic components (composites or polymers), resistance to UV exposure and chemical degradation must also be evaluated to prevent premature aging.

Quality assurance

Given the central role of the impeller in the turbomachinery train, a structural failure represents a catastrophic event that invariably leads to prolonged unscheduled shutdowns and significant financial losses. Consequently, ensuring reliability requires the implementation of a comprehensive Quality Assurance (QA) framework that spans the entire lifecycle, from the initial design phase through to the final manufacturing steps.

To verify integrity and compliance, a rigorous sequence of destructive and Non-Destructive Tests (NDT) is mandated. Recognized industry standards (such as API, ASTM and ASME) provide the foundational protocols for these inspections. However, for high-criticality applications, these baselines are often supplemented by customized, more stringent requirements defined by the Original Equipment Manufacturer (OEM) or the end-user. Figure 46 provides a schematic overview of the standard inspection hierarchy adopted for centrifugal compressor impellers, categorizing tests by their objective and execution phase.

Test Procedure	Standard	
	ASTM, ASME, API	DIN/ISO, SEP, SEL, AD
chemical analysis	ASTM E 30	Handbuch für das Eisenhüttenlaboratorium
tension test	ASTM A 370	DIN 50125/DIN 50145
notched bar impact test	ASTM E 23/A 370	DIN 50115
hardness test – Vickers – Brinell – Rockwell	ASTM E 92/E 384 ASTM E 370/E 10 ASTM A 370/E 18	DIN 50133 DIN 50351 DIN 50103 T.1
ultrasonic examination – procedure – of forgings – of ferritic cast steel parts	ASTM E 114 ASTM A 388 ASTM A 609	DIN 54126 T.2 SEP 1921 SEP 1922/DIN 1690 T.2
magnetic particle exam. – procedure – forgings – of steel castings	ASTM E 709 ASTM A 275 ASTM E 709/E 125	DIN 54130 SEP 1935
liquid penetrant exam. – procedure – of steel castings	ASTM E 165 ASTM E 433	DIN 54152 T.1 SEP 1936/DIN 1690 T.2
overspeed test	API 617	

Figure 46: Standards for inspections and tests of impellers [104]

A critical challenge in metallurgical validation is determining the mechanical properties of the finished component without compromising its integrity. To address this, the industry utilizes sacrificial test specimens (often referred to as "coupons" or "prolongations"). These specimens are manufactured from the same material heat batch as the impeller and accompany the component through every stage of the heat treatment cycle. By subjecting these coupons to the exact same thermal history, they develop a microstructure and mechanical response representative of the actual impeller. Destructive tests, such as tensile testing to determine yield strength and Charpy V-Notch impact testing to verify toughness, are then performed on these sacrificial elements. This methodology ensures that the certified properties accurately reflect the physical state of the operating component, validating the heat treatment process without damaging the final product.

Rotor shaft

The rotor shaft serves as the backbone of the compressor train, tasked with transmitting torque and supporting the rotating assembly under high dynamic loads. Unlike impellers, where fracture toughness and corrosion resistance often dominate, the primary metallurgical driver for shaft selection is hardenability.

The industry standard relies heavily on low-alloy steels from the AISI 41XX (Chromium-Molybdenum) and AISI 43XX (Nickel-Chromium-Molybdenum) series. While these families share similar mechanical properties in small cross-sections, they differ significantly in their depth of hardening. For smaller shafts, AISI 4140 provides sufficient

performance. However, as the shaft diameter increases, the thermal inertia prevents the core from cooling rapidly enough during quenching to transform fully into martensite. In these cases, AISI 4340 is the preferred choice; the addition of Nickel significantly increases hardenability, ensuring consistent mechanical properties from the surface to the core even in large sections. Furthermore, AISI 43XX alloys are mandatory for operating temperatures below -45°C due to their superior low-temperature toughness.

Regarding environmental compliance, shaft materials may need to adhere to NACE standards depending on the application. However, API 617, 9th Edition (Paragraph 6.2.1.8) provides an exemption for between-bearing compressors. It allows the shaft material to exceed standard yield strength and hardness limits in highly stressed areas (such as impeller seats), provided these areas are protected from the sour gas environment. This allows the designer to prioritize mechanical torque transmission capability over environmental cracking resistance in these specific zones.

Turbomachinery shafts are manufactured using either Rolled Bars or Forgings. Both processes are viable, but their application is governed by size constraints and supply chain considerations. Rolled Bars are characterized by tighter dimensional tolerances and superior surface finish, which reduces the extent of final machining required. They also offer a slight advantage in thermal stability due to a highly uniform grain structure. However, API 617 (Section 6.6.2.1.1) limits the use of rolled bars to a maximum diameter of 8 inches (200 mm). From a procurement perspective, using rolled bars may require purchasing an entire heat lot to ensure material consistency, which can be uneconomical for small production runs.

Forgings, conversely, are the standard solution for shafts exceeding the 200 mm limit or featuring complex, stepped geometries. Being typically made-to-order, forgings offer greater flexibility in sizing and reduce material waste for custom designs. While they may require more extensive machining to reach final tolerances, they allow for the optimization of grain flow lines to match the component's stress distribution. A schematic comparison of these two manufacturing routes is provided in Table 9. Rolled bars vs forging comparison.

Table 9. Rolled bars vs forging comparison [100]

Aspect	Rolled Bars	Forgings
Tolerances	Tighter tolerances, reducing machining	May require additional machining
Thermal Stability	Slightly better due to uniform grain structure	Slightly lower due to grain flow
Material Availability	May require purchasing entire heat batch	Typically made-to-order, reducing excess material

API 617 Compliance	Allowed up to 8 inches (200 mm) (API 617 9th ed., 6.6.2.1.1)	No size limitation, commonly used for larger shafts
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A specific failure mode associated with compressor shafts is wire wooling, a catastrophic wear phenomenon that typically occurs during start-up sequences. This mechanism is initiated by a small foreign particle embedding itself in the softer bearing material (usually the thrust bearing pads). Friction between the shaft and this particle generates intense localized heat, causing material transfer from the shaft to the particle. This transferred material work-hardens instantly, transforming the particle into a hard cutting tool that machines deep, wool-like grooves into the rotating shaft surface, destroying it in seconds.

To mitigate this risk, metallurgical literature recommends limiting the steel's Chromium content to below 1.8%; consequently, grades like AISI 4140 and 4340 are favored over higher-alloy variants. If the application demands steels with higher Chromium content (e.g., for corrosion resistance), protective countermeasures such as chrome plating, hardened sleeves, or advanced surface coatings must be implemented to prevent the initial adhesion and scoring mechanism.

Casings

The compressor casing serves as the primary pressure containment boundary and the structural backbone of the stationary assembly. For the majority of multistage centrifugal compressors, the material selection favors carbon or low-alloy steels. These grades provide an optimal balance between mechanical strength, machinability, and cost-effectiveness, making them suitable for standard industrial applications. The manufacturing strategy for casings is highly versatile; depending on the specific design pressure, size, and geometric complexity, they may be produced as single-piece castings, massive forgings, or fabricated structures welded from heavy plates.

When the application involves aggressive operating conditions, the material selection criteria shift significantly. For environments containing corrosive agents (such as sour gas) or for operations at cryogenic temperatures, standard carbon steels are often insufficient due to their susceptibility to chemical attack or brittle fracture. In these scenarios, high-alloy steels or stainless steels are employed to ensure adequate corrosion resistance and retained toughness. Furthermore, as with internal components, the selection process is frequently governed by NACE standards depending on the end-user's specific requirements.

From a structural safety perspective, the casing is treated as a pressure vessel. Consequently, API 617 mandates compliance with the ASME Boiler & Pressure Vessel Code (BPVC). A critical provision of this code is the strict regulation of material toughness

to prevent catastrophic failure. The code defines the required impact strength (measured via Charpy V-Notch energy absorption) as a function of the component's thickness; as the wall thickness increases, the probability of undetected defects and the constraint state at the crack tip increase, necessitating higher intrinsic toughness to ensure safety.

Finally, for fabricated casings constructed from plates or requiring structural welds, the selection of welding consumables is as critical as the base metal itself. To guarantee structural continuity, the filler material must be carefully matched to the base alloy to ensure that the weld joint exhibits mechanical properties (strength and toughness) and corrosion resistance equal to or exceeding those of the parent material, thereby eliminating weak points in the pressure boundary.

Diaphragms

The material selection for diaphragms, the stationary components that define the diffuser and return channel geometry, has undergone a significant evolution driven by the dual requirements of mechanical strength and maintainability.

Historically, diaphragms were predominantly manufactured from grey cast iron. While this material offered excellent castability and vibration damping properties, its low tensile strength and brittleness limited its application in high-pressure or high-load scenarios. When design conditions demanded higher structural competence, the industry initially transitioned toward ductile iron (nodular cast iron) or early forms of fabricated mild steel. regardless of the specific alloy employed, standard manufacturing practice invariably included a stress-relief heat treatment to mitigate the internal tensions generated during the casting or cooling process, ensuring dimensional stability.

A decisive shift in industry standards occurred during the 1970s and 1980s, establishing fabricated mild steel as the preferred material for modern compressors. This configuration typically features a bolted, split-line construction with blades that are either integrally milled or welded into place. The primary driver for this transition was weldability. Grey cast iron is notoriously difficult to repair; cracks or erosion damage often necessitate the scrapping of the entire component due to the material's inability to accept structural welds without cracking. In contrast, mild steel offers superior weldability, allowing for extensive repairs and refurbishment during the machine's service life. While ductile iron remains a viable alternative, offering a better compromise between castability and weldability compared to grey iron, fabricated mild steel remains the standard choice for optimizing the balance between manufacturing cost, structural integrity, and long-term maintainability.

Turbines

The transition from centrifugal compressors to industrial turbines marks a fundamental shift in the boundary conditions governing material selection. While compressors are primarily limited by mechanical stress and environmental corrosivity, the design of turbines is dominated by a single, critical parameter: Temperature. Turbines operate at thermal levels significantly exceeding those of typical compression processes, necessitating a complete re-evaluation of alloy capabilities and degradation mechanisms.

Material selection in this context not only ensures thermal resistance; it is a direct driver of plant performance. Thermodynamics dictates that the efficiency of power generation cycles (whether Rankine for steam or Brayton for gas) is a function of the peak cycle temperature. Consequently, the development of materials capable of withstanding higher turbine inlet temperatures allows for immediate gains in thermal efficiency and power output. However, this pursuit of performance introduces a complex economic trade-off as advanced high-temperature alloys significantly impact life-cycle costs, manufacturing complexity, and maintenance requirements.

Elevated temperatures introduce time-dependent failure modes that are negligible in standard compressor applications. Creep and thermal fatigue become the primary life-limiting factors, acting together with accelerated mechanisms of hot corrosion and oxidation. Within the turbine assembly, the blading system represents the most critical metallurgical challenge. While stator vanes are exposed to the maximum gas temperatures, the rotor blades (or buckets) operate under the most severe combined stress regime: they must withstand extreme thermal loads while simultaneously enduring the immense centrifugal tensile stresses generated by rotation. Consequently, the rotor blades are identified as the most stressed elements in the machine and will be the primary focus of the following analysis.

The following sections will analyse the material requirements for the two primary categories of turbomachinery utilized in industrial power generation and mechanical drive applications: Steam Turbines (ST) and Industrial Gas Turbines (IGT). While the specific working fluids differ, the structural components share similar functional requirements. The analysis will cover the four main sub-assemblies that define the machine's integrity: the Rotor, the Blades, the Casing, and the high-temperature Bolting.

Material selection factors

While the material selection for compressors is primarily driven by mechanical strength and aqueous corrosion resistance, the design of turbine components necessitates the evaluation of a distinct set of physical properties capable of withstanding extreme thermal environments. In addition to standard tensile and fatigue strength, turbine

materials are qualified based on four additional critical performance parameters: Creep strength, Stress rupture resistance, Hot corrosion resistance, and High-temperature hardness.

Creep and stress rupture: Larson-Miller parameter

At elevated temperatures, materials subjected to constant stress undergo progressive plastic deformation known as creep, which can lead to rupture even at loads significantly below the yield strength. To predict this time-dependent behaviour and compare the performance of different superalloys, the industry universally adopts the Larson-Miller Parameter (P_{LM}). This empirical parameter correlates the three governing variables of high-temperature endurance: temperature, stress, and time.² Mathematically, the Larson-Miller Parameter is expressed by the following equation:

$$P_{LM} = T(20 + \log t) \times 10^{-3}$$

Where:

- T = Operating temperature [K]
- t = Time to rupture [h]

The Larson-Miller relation allows engineers to extrapolate long-term performance from shorter laboratory tests performed at higher temperatures. By plotting stress against the calculated P_{LM} , a master curve is generated for each alloy, providing a fundamental tool for selecting materials that ensure dimensional stability over the machine's design life.

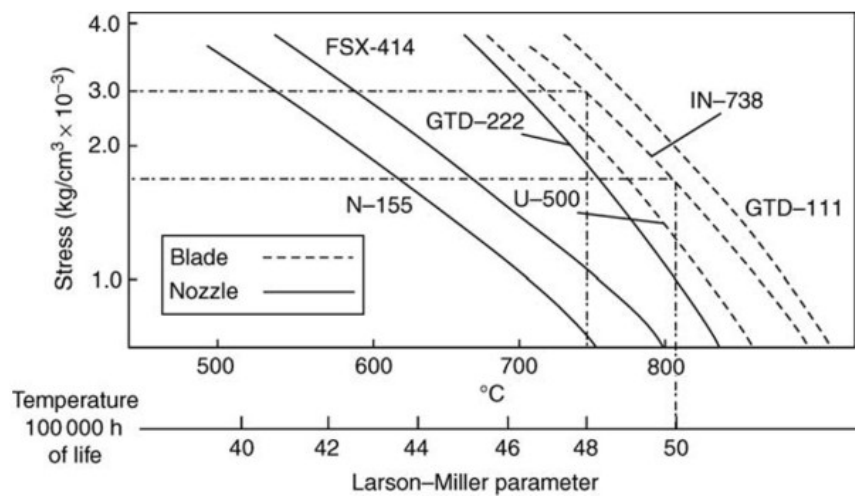


Figure 47: Larson Miller diagram [111]

FATT: toughness vs temperature

While high-temperature performance is paramount during steady-state operation, the mechanical integrity of the turbine is most vulnerable during cold startups. In these transient phases, the rotor is subjected to high centrifugal stresses while the material is still

at a relatively low temperature, potentially operating in a brittle regime. The critical parameter defining this risk is the Fracture Appearance Transition Temperature (FATT).⁴ The FATT represents the thermal threshold at which the material's failure mode shifts from ductile (high energy absorption) to brittle (low energy absorption). For safe operation, it is important that the rotor material possesses a FATT significantly lower than the lowest expected operating temperature during startup. Consequently, ASTM standards define the FATT values (Figure 48) for turbine rotor steels to assess the risk of catastrophic brittle fracture during cold starts. The variation of a material's fracture toughness (K_{IC}) as a function of temperature is frequently estimated using empirical correlations derived from the FATT. A specific formulation used to model this behaviour is:

$$K_{IC} = \frac{A}{B - (T - FATT)}$$

Where:

- A and B are material-specific constants
- T is the operating temperature

This relationship highlights that as the operating temperature (T) drops approaching the FATT, the denominator increases, mathematically reflecting the critical reduction in available toughness and the increased risk of brittle failure.

TABLE 2 Tensile and Notch Toughness Requirements								
	Grade A	Grade B		Grade C		Grade D		Grade E
	Class 2	Class 3	Class 4	Class 5	Class 6	Class 7	Class 8	Class 9
Tensile strength, min, ksi [MPa]	80 [550]	90 [620]	105 [725]	90 to 110 [620–760]	105 to 125 [725–860]	120 to 135 [825–930]	105 to 125 [725–860]	95 [655]
Yield strength, min, ksi [MPa]	55 [380]	70 [485]	85 [585]	70 [485]	85 [585]	95 [655]	85 [585]	70 [485]
0.2 % offset	60 [415]	75 [520]	90 [620]	75 [520]	90 [620]	100 [690]	90 [620]	75 [520]
0.02 % offset	55 [380]	70 [485]	85 [585]	70 [485]	85 [585]	95 [655]	85 [585]	70 [485]
Elongation in 2 in. or 50 mm, min, %:								
Longitudinal	22	20	17	20	18	18	17	20
Radial body	20	17	16	18	17	17	14	15
Reduction of area, min, %:								
Longitudinal	50	48	45	52	52	52	43	40
Radial body	50	45	40	50	50	50	38	35
Transition temperature FATT ₅₀ max, °F [°C]	100 [38]	110 [43]	140 [60]	10 [–12]	20 [–7]	30 [–1]	250 [121]	175 [80]
Room temperature impact, min, ft-lbf [J]	28 [38]	25 [34]	20 [27]	50 [68]	45 [61]	40 [54]	6 [8]	12 [16]

Figure 48: FATT values from ASTM standards

Hot corrosion resistance

Unlike aqueous corrosion encountered in compressors, hot corrosion is an accelerated degradation mechanism driven by the presence of molten salts deposited on component surfaces at high temperatures. This phenomenon is particularly severe in gas turbines in presence of contaminants.

Resistance to hot corrosion is primarily a function of the alloy's chemical composition. The introduction of higher percentages of Chromium is the standard

metallurgical solution; Chromium promotes the formation of a stable oxide scale on the surface. This passivation layer acts as a barrier, preventing the underlying metal from reacting with the corrosive contaminants and preserving the component's aerodynamic profile and structural integrity.

Steam turbines

Steam turbines operate within a distinct thermal and chemical envelope compared to their gas turbine counterparts. With maximum steam temperatures typically capped at approximately 590°C, the operating environment is generally less thermally severe than the combustion path of a gas turbine. However, compared to centrifugal compressors, steam turbines face significantly harsher conditions driven by the combination of high thermal gradients, sustained creep loading, and the potential for erosion from wet steam.

Consequently, the material selection strategy for steam turbines is bifurcated. The massive structural components, such as casings and rotors, are predominantly manufactured from low-alloy steels (Cr-Mo-V grades) capable of sustaining high pressures and temperatures. Conversely, the high-temperature flow path components, specifically the blading, require advanced 12% Chromium alloys to resist oxidation and erosion while maintaining creep strength.

The selection of blade materials is governed by ASTM standards, which ensure quality and consistency. Metallurgically, steam turbine blades are generally classified into three primary categories based on their position in the expansion line and the resulting stress-temperature state. The most diffused materials are 12-13% Chromium martensitic stainless steels, often alloyed with Molybdenum (Mo), Tungsten (W), Niobium (Cb), and Vanadium (V). These form stable carbides that pin grain boundaries, significantly enhancing creep resistance at high temperatures. For applications requiring superior strength-to-weight ratios or corrosion resistance, High-chromium precipitation-hardened steels are employed. Finally, titanium alloys are employed for specialized cases; they are frequently selected for the extremely long blades of the final low-pressure stages. In these locations, the centrifugal load is the dominant stress factor, and titanium's high specific strength allows for longer aerodynamic profiles without exceeding the mechanical limits of the rotor attachment.

Rotor

The steam turbine rotor serves as the central power transmission element and is typically manufactured from high-purity low-alloy steel forgings. The structural architecture of the rotor can be realized through three distinct manufacturing configurations, each presenting specific advantages and limitations.

The Monoblock configuration, where the shaft and disks are machined from a single, massive integral forging, represents the most widely used design in modern high-performance turbines. This monolithic approach offers superior mechanical integrity by eliminating interface surfaces and potential fretting zones. Alternatively, the Shrunk-fit design involves manufacturing disks individually and thermally shrinking them onto a central shaft. While this modular approach facilitates the replacement of damaged stages, it introduces stress concentrations at the keyways or interference zones and is generally limited to lower-temperature or lower-speed applications to prevent disk loosening. Finally, Welded rotors, constructed by welding separate forged disks together, are utilized by specific manufacturers. However, this method is less common due to the metallurgical challenges associated with maintaining consistent material properties in the heat-affected zones and ensuring dimensional stability across the welded joints.

Gas turbines

Gas turbines operate under extremely demanding conditions, with turbine inlet temperatures frequently exceeding 1300°C . In addition to these extreme thermal loads, the hot gas path is often subjected to corrosive contaminants and high-velocity erosive flows. Consequently, the material selection strategy is no longer just about strength; it has to mitigate thermodynamic degradation, necessitating advanced materials and specialized manufacturing techniques.

To cope with these extreme conditions, traditional blade manufacturing methods have evolved into specialized investment casting techniques designed to optimize the material's microstructure against creep and thermal fatigue. As illustrated in Figure 49, three primary crystallographic grain structures are currently utilized in gas turbine blade design: equiaxed grain (EG), directionally solidified (DS), and single crystal (SC).

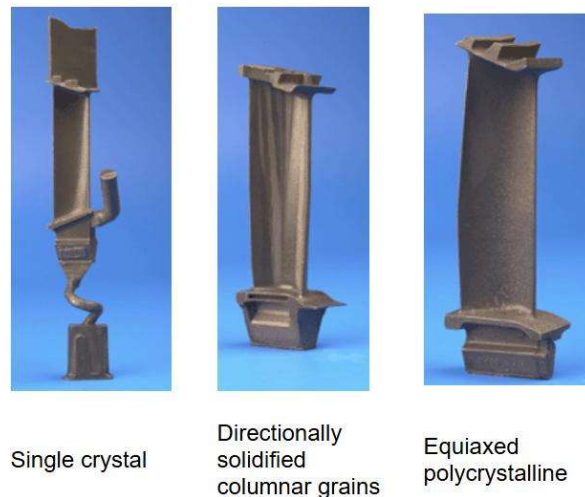


Figure 49: Blade crystallographic structures [112]

Equiaxed grain blades are produced through conventional vacuum investment casting, these blades consist of grains with random orientation. While this remains the most cost-effective method, it offers the lowest high-temperature performance among the three types. The presence of grain boundaries aligned perpendicular to the primary stress axis creates weak points susceptible to creep cavitation and sliding.

In directionally solidified process, the casting cooling rate is strictly controlled to align the grains parallel to the primary stress direction (the radial axis). This alignment eliminates transverse grain boundaries, thereby significantly reducing grain boundary sliding and improving resistance to both creep and thermal fatigue compared to EG blades.

The single crystal structure represents the most advanced and expensive option, SC blades are cast as a continuous single grain structure with no internal grain boundaries. By eliminating boundary-related weaknesses entirely, SC blades offer superior mechanical strength, exceptional creep resistance, and maximum thermal fatigue endurance, allowing for the highest operating temperatures.

The extreme operating environment of the gas turbine has driven the development of superalloys, a class of specialized metallic materials engineered to maintain high mechanical integrity under prolonged exposure to elevated temperatures. While definitions in metallurgical literature vary, a superalloy is generally defined as:

"A metallic alloy which can be used at high temperatures, often in excess of 0.7 of the absolute melting temperature" [112].

"Alloys generally capable of operating at temperatures above 540°C" [111].

Figure 50 illustrates the distribution of these materials within the gas turbine engine, highlighting how superalloys dominate the combustion and turbine sections.

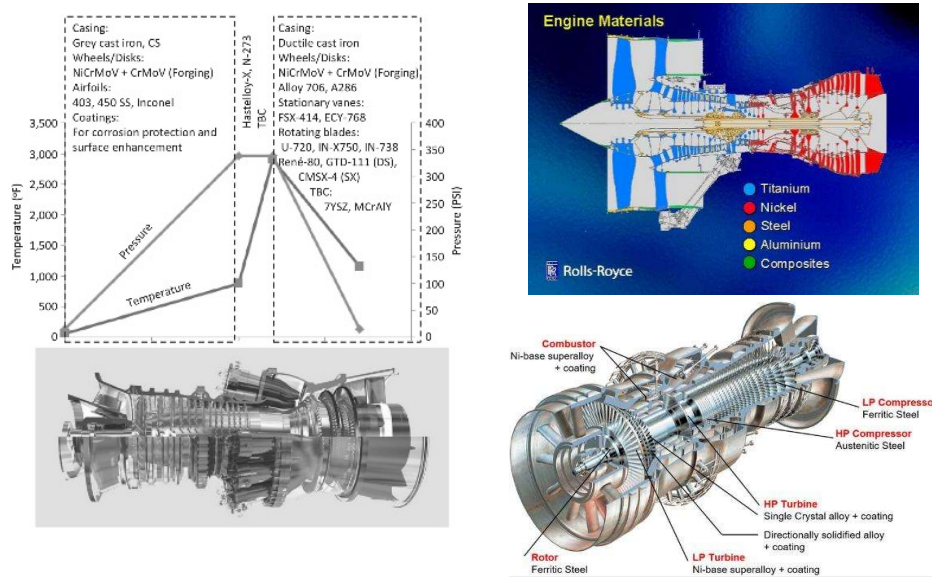


Figure 50: Material distribution inside gas turbine engines [112]

For Industrial Gas Turbines (IGTs), superalloys are primarily classified based on their principal element:

- Iron-Nickel-Based Superalloys
- Nickel-Based Superalloys
- Cobalt-Based Superalloys

It is important to note that despite their high strength-to-weight ratio and high melting point, Titanium alloys are strictly avoided in the hot sections of the turbine. This exclusion is due to the risk of "titanium fire" (rapid oxidation and ignition) at temperatures exceeding 400°C [112].

At the microstructural level, all three superalloy classes share a common Austenitic Face-Centered Cubic (FCC) lattice structure, referred to as the gamma (γ) matrix. This ductile matrix provides the necessary solubility for alloying elements. The critical difference lies in the strengthening mechanisms utilized.

Iron-nickel and nickel-based superalloys utilize the precipitation of an intermetallic phase known as gamma prime (γ') combined with carbides to enhance strength and creep resistance.

Cobalt-based superalloys rely almost exclusively on carbide strengthening (solid solution hardening and carbide precipitation), as the formation of the γ' phase is thermodynamically unstable above 750°C in cobalt matrices [113].

Figure 51 highlights the fundamental crystallographic distinctions between the matrix and the strengthening phase. The gamma phase (left) functions as a disordered substitutional solid solution; within its Face-Centered Cubic (FCC) lattice, solute atoms

randomly substitute parent atoms without a fixed pattern. In contrast, the gamma prime phase (right) is an intermetallic compound characterized by an ordered $L1_2$ structure. Unlike the random distribution in the gamma matrix, the γ' lattice exhibits a strict geometric arrangement where specific solute atoms (typically Aluminum or Titanium) always occupy the cube corners, while the solvent atoms (Nickel) occupy the face centers. This ordered atomic locking mechanism is the primary driver of the alloy's high-temperature strength and creep resistance.

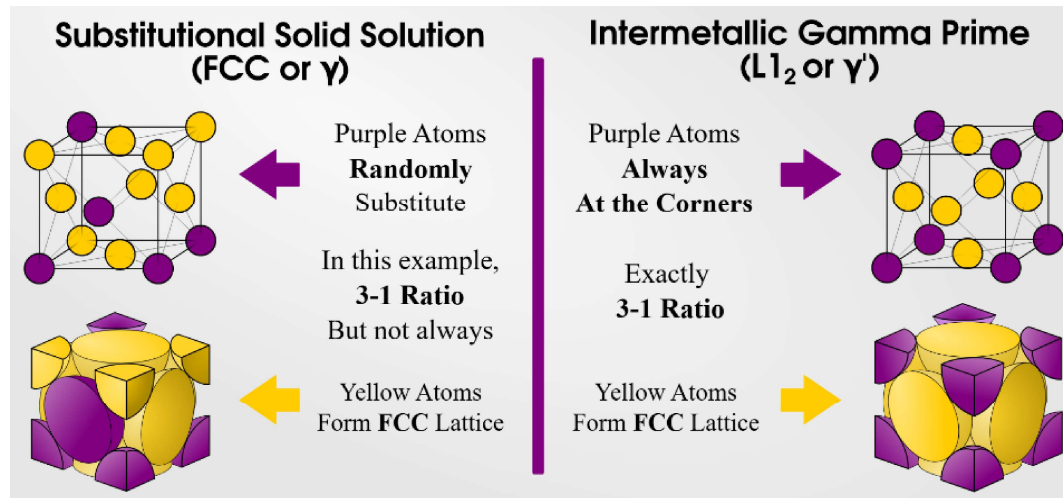


Figure 51: Gamma vs gamma prime lattice comparison

Iron-Nickel-based superalloys

Among the three high-temperature superalloys, Iron-Nickel-based alloys represent the most economically viable option. Their cost advantage, derived from the abundance of iron relative to nickel and cobalt, combined with satisfactory high-temperature mechanical performance, makes them the preferred choice for moderate-temperature turbine applications where the extreme capabilities of pure nickel-based alloys are not strictly required.

The primary constituent of these alloys is Iron, which forms the bulk of the matrix. However, to ensure stability at elevated temperatures, approximately 25% Nickel is typically added. The primary function of nickel in this context is to stabilize the Face-Centered Cubic (FCC) austenitic gamma (gamma) matrix, preventing phase transformations that would degrade properties at operating temperatures.

Beyond the base elements, the formulation includes specific alloying additions such as Chromium, Molybdenum, and Manganese. These elements are introduced primarily to provide solid solution strengthening, distorting the lattice to impede dislocation motion—and to enhance oxidation and corrosion resistance in the turbine environment.

The mechanical behaviour of Iron-Nickel superalloys is defined by how they impede deformation. According to the framework established by Akca and Gursel [115][15], these alloys can be grouped into three distinct categories based on their primary strengthening mechanism: martensitic transformation strengthened alloys, work-hardened austenitic alloys, and precipitation-hardened austenitic alloys. Martensitic transformation strengthened Alloys rely on a martensitic structure for strength. However, their thermal stability is limited; generally, this typology is restricted to operating temperatures below 540°C, as higher temperatures induce over-tempering and a loss of strength. Work-hardened austenitic alloys achieve their strength through cold working which increases dislocation density. They maintain the stable austenitic structure necessary for higher-temperature service. Precipitation-hardened austenitic alloys represent the highest performance tier within this group; these alloys utilize the precipitation of intermetallic phases (such as γ') to lock the crystal lattice. They are suitable for temperatures significantly exceeding the 540°C limit of the martensitic grades. Within the industrial landscape, standardization is critical for material selection. Akca and Gursel [115] further report a detailed subclassification of the AISI 600 series of superalloys.

Nickel-based superalloys

Nickel-based superalloys represent the most advanced high-temperature materials and are the most widely used class of alloys for the hottest rotating parts of industrial and aerospace gas turbines [111]. These materials are defined by their exceptional balance of high-temperature tensile strength, creep rupture resistance, and oxidation/corrosion stability. Consequently, they are the mandatory choice for turbine blades and disks operating in environments exceeding 1000°C.

The superior performance of these alloys is derived from the two-phase microstructure. The gamma phase matrix is strengthened by the gamma prime one. The latter, nominally $\text{Ni}_3(\text{Al,Ti})$, exhibits an ordered L1_2 crystal structure. It acts as a barrier to dislocation movement within the γ matrix, effectively locking the microstructure against deformation. The volume fraction of γ' is tailored to the application temperature [112].

For moderate service ($\sim 750^\circ\text{C}$), a γ' fraction of approximately 20% is typically sufficient. In severe services ($>1000^\circ\text{C}$), for high-pressure turbine blades, the γ' fraction is increased to as high as 60% to maximize creep resistance.

To visualize the complex phase relationships between the primary constituents (Nickel, Aluminium, and Titanium), ternary phase diagrams are utilized (Figure 52). In these diagrams, each vertex represents a pure element, and the grid lines parallel to the opposite edge indicate the iso-percentage of that specific element, allowing metallurgists to predict phase stability zones.

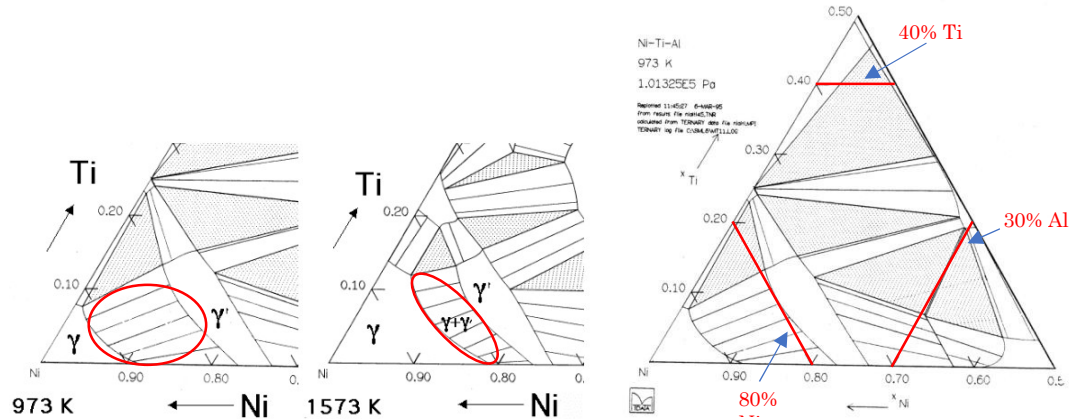
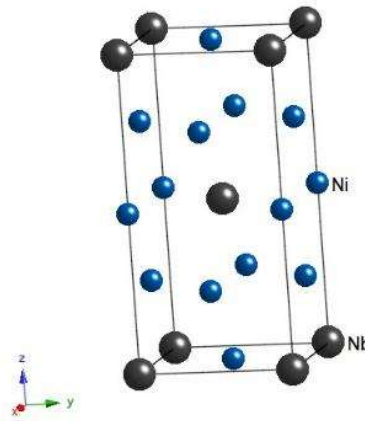


Figure 52: Nickel based superalloys ternary diagrams [112]

While blades prioritize extreme temperature resistance, turbine disks require exceptional tensile strength at slightly lower intermediate temperatures. For these applications, a third strengthening phase, gamma second (γ''), is often introduced.

Typical of alloys like Inconel 718, the γ'' phase (typically Ni_3Nb or Ni_3V) possesses a body-centered tetragonal (BCT) lattice structure (Figure 53). This precipitate provides superior mechanical strength up to approximately 700–800°C, making it ideal for rotor disks, though it becomes unstable at the higher temperatures experienced by airfoils.

Figure 53: γ' phase lattice [112]

The chemical composition of nickel-based superalloys is tailored based on the manufacturing method (EG, DS, and SC) and the specific environment (Figure 54).

Alloy	Cr	Co	Mo	W	Ta	Re	Nb	Al	Ti	Hf	C	B	Y	Zr	Other
<i>Conventionally Cast Alloys</i>															
Mar-M246	8.3	10.0	0.7	10.0	3.0	—	—	5.5	1.0	1.50	0.14	0.02	—	0.05	—
Rene' 80	14.0	9.5	4.0	4.0	—	—	—	3.0	5.0	—	0.17	0.02	—	0.03	—
IN-713LC	12.0	—	4.5	—	—	—	2.0	5.9	0.6	—	0.05	0.01	—	0.10	—
C1023	15.5	10.0	8.5	—	—	—	—	4.2	3.6	—	0.16	0.01	—	—	—
<i>Directionally Solidified Alloys</i>															
IN792	12.6	9.0	1.9	4.3	4.3	—	—	3.4	4.0	1.00	0.09	0.02	—	0.06	—
GTD111	14.0	9.5	1.5	3.8	2.8	—	—	3.0	4.9	—	0.10	0.01	—	—	—
<i>First-Generation Single-Crystal Alloys</i>															
PWA 1480	10.0	5.0	—	4.0	12.0	—	—	5.0	1.5	—	—	—	—	—	—
Rene' N4	9.8	7.5	1.5	6.0	4.8	—	0.5	4.2	3.5	0.15	0.05	0.00	—	—	—
CMSX-3	8.0	5.0	0.6	8.0	6.0	—	—	5.6	1.0	0.10	—	—	—	—	—
<i>Second-Generation Single-Crystal Alloys</i>															
PWA 1484	5.0	10.0	2.0	6.0	9.0	3.0	—	5.6	—	0.10	—	—	—	—	—
Rene' N5	7.0	7.5	1.5	5.0	6.5	3.0	—	6.2	—	0.15	0.05	0.00	0.01	—	—
CMSX-4	6.5	9.0	0.6	6.0	6.5	3.0	—	5.6	1.0	0.10	—	—	—	—	—
<i>Third-Generation Single-Crystal Alloys</i>															
Rene' N6	4.2	12.5	1.4	6.0	7.2	5.4	—	5.8	—	0.15	0.05	0.00	0.01	—	—
CMSX-10	2.0	3.0	0.4	5.0	8.0	6.0	0.1	5.7	0.2	0.03	—	—	—	—	—
<i>Wrought Superalloys</i>															
IN 718	19.0	—	3.0	—	—	—	5.1	0.5	0.9	—	—	0.02	—	—	18.5Fe
Rene' 41	19.0	11.0	10.0	—	—	—	—	1.5	3.1	—	0.09	0.005	—	—	—
Nimonic 80A	19.5	—	—	—	—	—	—	1.4	2.4	—	0.06	0.003	—	0.06	—
Waspaloy	19.5	13.5	4.3	—	—	—	—	1.3	3.0	—	0.08	0.006	—	—	—
Udimet 720	17.9	14.7	3.0	1.3	—	—	—	2.5	5.0	—	0.03	0.03	—	0.03	—
<i>Powder-Processed Superalloys</i>															
Rene' 95	13.0	8.0	3.5	3.5	—	—	3.5	3.5	2.5	—	0.065	0.013	—	0.05	—
Rene' 88 DT	16.0	13.0	4.0	4.0	—	—	0.7	2.1	3.7	—	0.03	0.015	—	—	—
N18	11.2	15.6	6.5	—	—	—	—	4.4	4.4	0.5	0.02	0.015	—	0.03	—
IN100	12.4	18.4	3.2	—	—	—	—	4.9	4.3	—	0.07	0.02	—	—	—

Figure 54: Compositions of commercial Ni-based superalloys (wt.% bal Ni) [115]

Key alloying elements serve distinct functions. Chromium (Cr) is essential for forming a protective Cr_2O_3 oxide layer to resist hot corrosion and oxidation. Cobalt (Co), Molybdenum (Mo), Chromium (Cr) provide solid solution strengthening to the matrix. Grain Boundary Strengtheners (C, B, Zr, Hf) are critical for EG and DS blades, as they segregate to grain boundaries to prevent sliding and cracking. However, in SC alloys, grain boundaries are non-existent; therefore, these elements are removed to raise the incipient melting point of the alloy.

The development of single crystal superalloys is categorized by "generations," defined primarily by the addition of refractory metals like Rhenium (Re) and Ruthenium (Ru) to improve high-temperature creep resistance and thermal stability (Table 10). These generational advances represent a continuous effort to extend component life and enable higher turbine inlet temperatures, directly driving the efficiency of modern aerospace and industrial gas turbines.

Table 10: SC Ni-based superalloys generations	
Generation	Key additions
1 st	none
2 nd	2-3% Re
3 rd	5-6% Re
4 th	Re + Ru
5 th	Increased Ru content

Cobalt-based superalloys

While Nickel-based superalloys dominate the rotating sections of the turbine due to their superior intermediate-temperature strength, Cobalt-based superalloys occupy a

critical niche in the highest temperature zones. They are renowned for their exceptional thermal stability and ability to retain mechanical strength at temperatures approaching their melting point, making them the preferred material for stationary hot-section components, such as combustion chambers and first-stage stator vanes (nozzles) [114].

The operational boundary between Nickel and Cobalt alloys is distinct. Nickel-based superalloys generally offer superior yield strength and creep resistance up to approximately 1100°C; however, beyond this threshold, the γ' precipitate coarsens or dissolves, leading to a rapid drop in properties. In contrast, Cobalt-based alloys exhibit a flatter strength-vs-temperature curve, maintaining structural integrity significantly closer to their absolute melting temperature. Traditionally, the high-temperature strength of Cobalt alloys does not rely on the γ' precipitate found in Nickel systems. Instead, it is derived primarily from refractory metal carbides (involving Tungsten or Chromium) that precipitate at the grain boundaries. This robust carbide network effectively "pins" the grain boundaries, enhancing the alloy's creep resistance and preventing grain boundary sliding at extreme thermal loads. Note on Advanced Research: Although the $L1_2$ structured γ' phase is historically absent in commercial Cobalt alloys, recent metallurgical research has identified the formation of stable γ' precipitates in novel Co–Al–W ternary systems [113]. This discovery, illustrated in the phase diagrams of Figure 55, suggests the potential for a new generation of precipitation-hardened Cobalt superalloys.

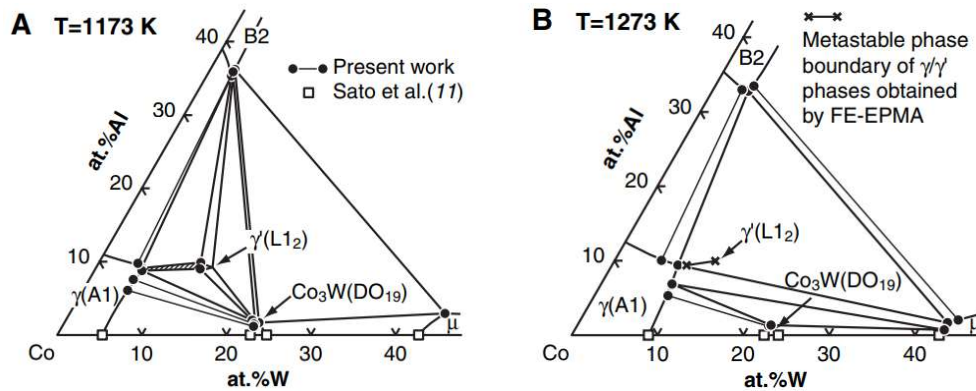


Figure 55: Co-Al-W superalloys ternary diagram [113]

Cobalt superalloys exhibit an allotropic transformation, possessing two distinct crystal structures depending on the temperature:

- Face-Centered Cubic (FCC): The dominant stable phase at high operating temperatures.
- Hexagonal Close-Packed (HCP): The stable phase at lower temperatures.

The chemical composition is meticulously balanced to optimize this phase stability and environmental resistance (Figure 56).

Compositions of cast and wrought cobalt-based alloys

Alloy	Amount (wt.%) of the following elements								
	C	Cr	Ni	W	B	Ta	Zr	Others	Co
X-40	0.5	26.5	10.5	7.5	0.01	—	—	—	Balance
X-45	0.75	26.5	10.5	7.5	0.01	—	—	—	Balance
FSX-414	0.35	29.5	10.5	7	0.01	—	—	—	Balance
FSX-418	0.35	29.5	10.5	7	0.01	—	—	0.15 Y	Balance
MAR-M 302	0.85	21.5	—	10	0.005	9	0.2	—	Balance
MAR-M 509	0.60	21.5	10	7	—	3.5	0.5	0.2 Ti	Balance
WI-52	0.45	21	—	11	—	—	—	2 Nb	Balance
AiResist 13	0.45	21	1	11	—	—	—	3.5 Al	—
HA-188	0.08	25.5	22	15	—	—	—	0.20 Si, 0.08 La	Balance
HA-25	0.1	20	10	15	—	—	—	1.5 Mn	—

Figure 56: Co-based superalloys compositions [114]

The principal alloying elements, nickel, chromium, and tungsten, act as solid solution strengtheners but influence the lattice differently:

- Nickel is added to stabilize the FCC lattice, ensuring ductility and toughness.
- Chromium and Tungsten tend to promote the HCP structure.

To survive the harsh combustion environment, Chromium content is typically maintained above 20% [114]. This high concentration allows the continuous formation of a protective Cr_2O_3 scale, providing essential resistance to oxidation and hot corrosion.

Carbon is a decisive alloying element, typically present in concentrations between 0.25% and 1%. It drives the formation of the carbide networks essential for creep strength.

However, reliance on carbide strengthening introduces specific manufacturing constraints. Hot working processes, such as forging or rolling, can mechanically disrupt these continuous carbide networks. Since the alloy's high-temperature creep strength relies heavily on the integrity of these grain boundary precipitates, breaking them can severely diminish performance. Consequently, Cobalt-based components are frequently manufactured via investment casting to preserve the carbide morphology, and careful control of post-processing conditions is required to avoid degrading the microstructure.

Part V: Overhaul and repair

Introduction

The research activities described in this section were conducted during an international placement at the headquarters of Blackstone Industrial in Canada (Edmonton, Alberta). While the previous sections of this thesis focused on the design and predictive analysis (CFD/FEM) of compression systems, this part addresses the operational reality of the machinery lifecycle: the Overhaul and Repair (O&R) phase.

In the heavy machinery sector, a historical disconnect often exists between the Engineering Department, responsible for design specifications and acceptance criteria, and the workshop/field operations, responsible for the physical execution of maintenance and repairs. This information asymmetry can lead to operational inefficiencies, such as delays in decision-making when non-conformities are detected, loss of critical field data that could improve future designs, and ambiguity in repair execution standards.

The primary objective of the activities performed at Blackstone Industrial was to bridge this gap. The work focused on analysing the existing workflow and implementing new protocols to streamline the bidirectional communication between the shop floor and the engineering office, transforming the O&R process from a purely execution-based task into an integrated engineering feedback loop.

The complexity of this specific industrial context is significantly amplified by the geographical separation between the two key stakeholders: the Engineering Department (CST), located in Florence, Italy, and the Repair Workshop (Blackstone Industrial), located in Western Canada. This trans-continental structure introduces profound logistical and communicative challenges that define the boundary conditions of the project.

The 8-hour time difference between Italy and Alberta creates a condition of operational disjointedness. The overlap window for synchronous communication is compressed. A technical query raised by the Canadian workshop during their morning shift typically reaches the Italian engineering team after business hours. Consequently, the technical settlement cannot be issued until the following European working day, introducing a systematic 12-to-24 hour latency in the decision-making chain. This "stop-and-go" workflow is incompatible with the urgency often required.

Unlike a local arrangement where the engineer can physically inspect the damaged component, the CST engineering team relies entirely on digital proxies (photographs, inspection reports, dimensional sheets) transmitted by the workshop. This lack of direct sensory contact increases the risk of misinterpretation. A nuanced defect, such as a specific

fretting pattern or a subtle crack indication, is difficult to evaluate remotely without standardized, high-fidelity data acquisition protocols.

Therefore, the optimization activity described in the following chapters was not merely procedural but structural, aiming to build a communication infrastructure capable of mitigating the friction caused by this geographical and temporal distance.

Direct Field Observation

To bridge the gap between theoretical design and operational reality, this research necessitated direct integration into the Quality Control (QC) workflow on the shop floor. Active participation in inspection protocols facilitated the empirical correlation of theoretical failure modes with physical evidence of wear and damage.

The study integrated several inspection methodologies into the digital reporting workflow, including dimensional metrology, advanced digitization (3D scanning), material verification, contact analysis, and non-destructive testing (NDT).

Precision dimensional metrology serves as the foundational assessment for determining component reusability and characterizing wear conditions. Direct measurements were conducted using calibrated outside micrometers and internal bore gauges, focusing on critical interfaces such as the crosshead pin-to-bushing assembly or shaft interference fits. Multi-point circumferential measurements were performed to identify ovalization and account for non-uniform wear patterns along cylindrical surfaces.

Crucially, these are not merely geometric verifications but diagnostics of the machine's tribological health. For instance, verifying diametrical clearances is essential to ensure correct fits between parts. Deviations beyond OEM repairability ranges were established as immediate rejection criteria.

To mitigate the limitations of remote engineering and capture complex component geometries, 3D laser scanning was implemented. Beyond inspection, this technology facilitates specialized reverse engineering, geometric upgrades, and high-fidelity data archiving. Two primary scanning modalities were utilized: the absolute (articulated) arm and the handheld scanner.

The absolute arm scanner features a laser probe mounted on the end of an articulated kinematic structure. The first tool is the absolute arm, which features a scanner placed at the tip of an articulated kinematic structure (Figure 57). The primary advantage of this instrument lies in its rapid calibration procedure, as the arm utilizes its fixed base as the main geometric reference. Additionally, an integrated contact probe allows for the precise definition of datum geometries, such as planes and cylindrical axes, making the obtained digital model easier to handle during post-processing. However, this tool is

constrained by a limited operational range defined by the arm's geometry, which restricts the size of suitable scanning objects. Furthermore, accuracy may degrade near the boundaries of this range, where the arm is forced into awkward configurations that complicate the handling of the scanner.



Figure 57: Absolute arm 3D scanner

Conversely, the handheld scanner (Figure 58) provides a potentially unlimited range, making it suitable for larger components. However, this increased volume is often related to lower accuracy, as error accumulation depends on the distances involved between points. The main disadvantage of this scanner type is the object preparation required. Since the scanner is free-floating without a fixed physical reference, it relies on photogrammetry and tracking methods. This necessitates the application of reference stickers or magnets (fiducial markers) on the object to establish a coordinate system. This is a time-consuming procedure, and data integrity can be compromised if any markers are displaced during the measurement process.



Figure 58: Handheld 3D scanner

While these tools are incredibly useful for overhaul, repair, and engineering activities, it is important to remark on a significant limitation affecting both technologies. It remains difficult to capture the internal geometries typical of centrifugal compressors, specifically labyrinth seals and shrouded impeller vanes. While the profile of labyrinth seals can be complemented by micrometer measurements, the complete digitization of internal impeller flow paths remains a significant technical challenge due to line-of-sight constraints.

In the context of repair strategies involving component substitution or thermal joining processes such as soldering and welding, material verification represents a critical risk factor. Components arriving for overhaul frequently lack adequate traceability tags, a challenge that is particularly prevalent in legacy equipment. To definitively determine the metallurgical composition of these components, Positive Material Identification (PMI) was systematically performed using portable X-Ray Fluorescence (XRF) analysers (Figure 59).

This non-destructive testing technique operates by irradiating the sample with X-rays, subsequently measuring the fluorescent energy emitted by specific elements within the alloy to determine its elemental composition. It is imperative to highlight that this procedure demands the employment of specialized and certified personnel, given the safety risks associated with potential radiation exposure if the device is operated incorrectly. The tool facilitates the rapid identification of metal composition on the component's surface. While it is capable of correlating elemental data with standard AISI codes for common alloys, the operator must be aware that certain alloy compositions may exhibit spectral overlap. Consequently, rigorous interpretation is necessary to avoid

misidentification, a limitation that, along with specific material selection criteria, has been extensively detailed in Part IV.

From a structural integrity perspective, verifying the constituent material of a component is crucial for preventing the inadvertent use of lower-grade substitutes, which could severely compromise fatigue life. By validating the material, PMI ensures that the material properties utilized in the theoretical engineering assessment are fully aligned with the physical reality of the component.



Figure 59: PMI tool example

While digital metrology provides essential quantitative data, analogue methods remain indispensable for the verification of physical contact quality between mating surfaces. The standard procedure employed involves the application of a thin, uniform layer of non-drying engineer's blue (Prussian blue) to the interface of the component. The part is subsequently assembled with its counterpart, and the prescribed clamping force is applied to simulate assembly operational conditions. Upon disassembly, the transfer or displacement of the blue pigment reveals the effective contact pattern (Figure 60).

This empirical test serves to validate the geometric conformity and load transfer capability of the connection. Generally, a contact area coverage of less than 75-80% is deemed unacceptable, as it induces localized stress concentrations and, in the case of sealing surfaces, increases the susceptibility to leakage paths. Direct observation of this verification process provided the research team with immediate, tangible evidence regarding the integrity of the assembly, complementing the digital datasets collected during earlier phases.

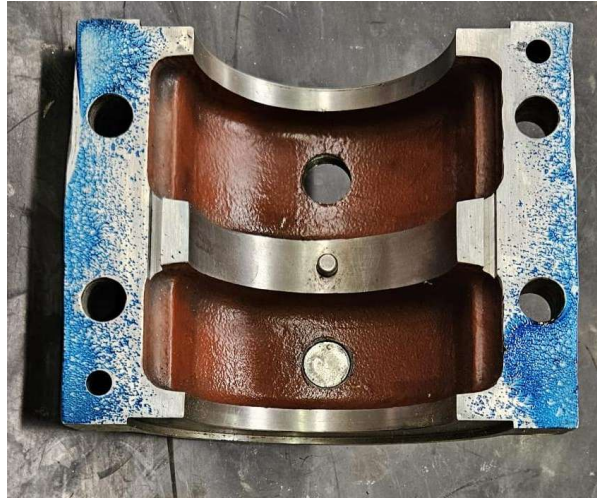


Figure 60: Prussian blue test result

Complementing the dimensional and material verifications, the inspection protocol incorporates advanced Non-Destructive Testing (NDT) methods to identify surface and sub-surface discontinuities: Liquid Penetrant Inspection (LPI) and Magnetic Particle Inspection (MPI).

To detect incipient failures invisible to the naked eye, liquid penetrant inspections were systematically monitored (Figure 61). The process begins with the application of a low-surface-tension dye (penetrant) to the pre-cleaned surface of components such as connecting rods or cylinder liners. Following a specific dwell time to allow for intrusion, the excess penetrant is removed, and a developer is applied. Through capillary action, the developer draws the penetrant out of surface-breaking defects, creating a high-contrast indication. This method is critical for identifying fatigue crack initiation sites; the detection of even a single micro-crack typically constitutes immediate rejection criteria, necessitating the scrapping of the component to prevent catastrophic failure in operation.

This is critical for identifying fatigue crack initiation sites. Finding a micro-crack in any component means it will be scrapped.

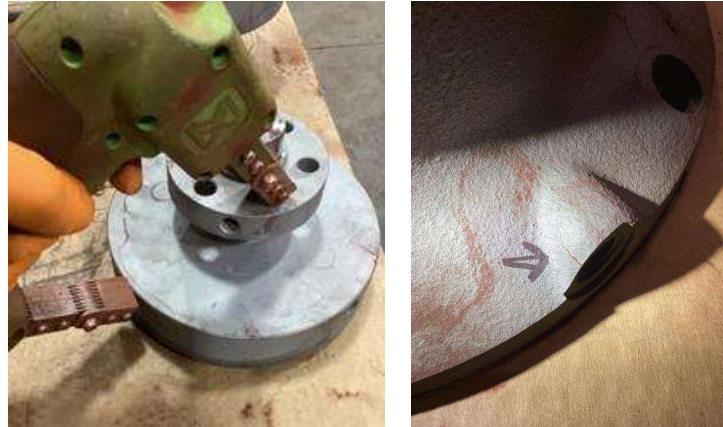


Figure 61: LPI examination results example

Where applicable, particularly on ferromagnetic materials, magnetic particle inspection offers superior sensitivity for detecting surface and near-surface discontinuities (Figure 61). This technique relies on the principle of magnetic flux leakage. To perform the inspection, the component is positioned between the poles of an electromagnetic yoke or within a coil to induce a magnetic field throughout the part. While the magnet is energized, a suspension of ferromagnetic particles is applied to the surface.

In a sound material, the magnetic flux lines flow undisturbed. However, in the presence of a discontinuity, the magnetic field is interrupted and forced to exit the surface, creating a local leakage field. This leakage field acts as a magnet itself, attracting the suspended particles to cluster directly over the flaw, thereby rendering the defect's size and shape visible under appropriate lighting conditions.

It is crucial to note that the sensitivity of this method is strictly directional. The leakage field is strongest, and therefore the defect indication is clearest, when the discontinuity is oriented perpendicular to the magnetic flux lines. Defects parallel to the magnetic flow may not cause sufficient leakage to trap the particles. Consequently, to ensure comprehensive coverage and the detection of randomly oriented flaws, the inspection is repeated by rotating the magnetic field generation relative to the component, typically in orthogonal directions.



Suspension of ferromagnetic particles

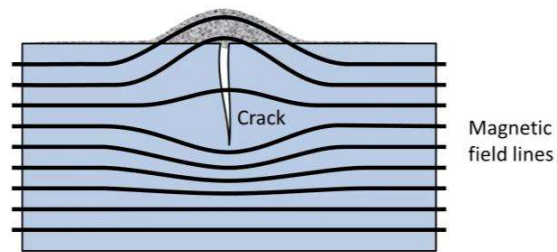


Figure 62: MPI examination results example

All the overhaul methodologies and diagnostic techniques previously described are systematically consolidated within the Inspection and Test Plan (ITP). Far from being a mere procedural checklist, the ITP serves as the overall governance document that codifies, categorizes, and sequences every operational intervention. It establishes a rigid framework that maps specific operational steps to their corresponding industrial standards and internal specifications, while assigning precise responsibilities between the technical operations team, quality control, and engineering management.

The ITP functions as a central hub that dictates the collection of specific technical data through linked data sheets. Rather than allowing for generic qualitative assessments, the protocol mandates the recording of quantitative values. For instance, during the geometric inspection of compressor cylindrical components, the system requires orthogonal measurements across multiple planes to quantify ovality and taper against strict tolerance bands. Similarly, run-out analyses and surface finish verifications are recorded to ensure the integrity of critical tribological interfaces. Furthermore, essential assembly dimensions, such as the stand-off distance in piston assemblies, are documented both pre-disassembly and post-reassembly to guarantee that the compression volume remains consistent with the original design parameters.

Beyond data collection, the ITP enforces a risk management strategy through designated "hold points." These are critical junctures where the workflow is deliberately arrested, typically following cleaning and NDT phases, to allow for a formal review of the aggregated evidence. At this stage, engineering management must validate the diagnostic data to definitively define the repair scope before any irreversible machining or coating processes commence. This approach shifts the maintenance paradigm from a standard execution model to a condition-based decision-making process.

By integrating technical metrology with project management milestones, the ITP ensures complete traceability and accountability. Its scope is holistic, encompassing the entire project lifecycle, from the preliminary equipment receipt and crate inspection through to the final "green tag" quality assurance release. This rigorous documentation creates a permanent audit trail, ensuring that every engineering decision is supported by empirical data collected from the shop floor.

Training manuals

A significant output of the research activity was the development of a comprehensive suite of training manuals designed to bridge the gap between abstract Original Equipment Manufacturer (OEM) specifications and practical shop-floor application. While OEMs typically provide dimensional tolerances and assembly diagrams, they often lack the operational context and diagnostic logic required for effective troubleshooting. To address this, the research synthesized data from Repair Engineering Instructions (REI), available technical literature [129, 130], and direct empirical observation into a structured pedagogical framework.

The primary objective of these manuals is to standardize practical expertise often held by senior technicians, converting it into an explicit, reproducible educational asset. The development methodology followed a consistent structure applicable across various critical machine components.

The structure prioritizes operator safety by identifying mechanical risks associated with disassembly before any technical instruction begins. In the context of complex reciprocating machinery, specific emphasis was placed on stored energy hazards. Many components utilize high-tension mechanisms or pressurized chambers; improper disassembly techniques can result in the rapid, uncontrolled release of kinetic energy. The manuals explicitly map these hazardous zones, integrating safety protocols directly into the technical workflow rather than treating them as a supplementary addendum.

A critical deficiency identified in standard maintenance documentation is the lack of functional context; technicians are often instructed on what to measure, but not why the

component functions as it does. The developed manuals address this by incorporating a theoretical overview of the component's role within the broader mechanical or thermodynamic system. The text clarifies the distinction between different operational methods and provides a clear nomenclature of component sub-types. This theoretical grounding ensures that operators understand the downstream consequences of a single component failure, such as the inability of the system to regulate load or maintain pressure, thereby fostering a holistic understanding of the machinery.

Moving beyond simple dimensional checks, the manuals integrate a diagnostic approach to inspection. The text guides the operator to look for specific evidence of wear that correlates with known failure modes. Inspection criteria are expanded to include material degradation, fatigue indications, and surface irregularities on critical interfaces. This section focuses on tribological indicators, such as fretting, galling, or seal degradation, that could indicate misalignment, poor lubrication, or chemical attack. This approach transforms the inspection process from a passive measurement task into an active root-cause analysis.

Finally, the manuals close the loop between repair and operation by defining rigorous testing protocols. The research emphasized that dimensional restoration alone is insufficient; functional verification is required. The protocols outline methods to simulate operational conditions on the workbench, validating actuation strokes, sealing integrity, or load-bearing capabilities. By standardizing these bench tests, the manuals ensure that the component is not only geometrically correct but functionally valid before it is reinstalled in the critical path of the machinery.

Conclusions

The research activities conducted at Blackstone Industrial represent a crucial phase in the broader study of compression system lifecycle management. By embedding the research directly within the operational environment of the repair shop, it was possible to deconstruct the traditional barriers that separate the theoretical design office in Italy from the physical execution of maintenance in Canada and Spain.

To address the operational disjointedness, a newly developed procedure for remote damage assessment and governance was implemented. Unlike standard inspection methods, this novel framework integrated advanced 3D scanning and NDT data directly into a newly structured Inspection Test Plan (ITP) architecture. The application of this specific protocol yielded tangible results: it reduced decision-making latency by around 20% and allowed the successful remote evaluation of turbomachinery components.

The implementation of the diagnostic and procedural frameworks described in this section, ranging from advanced 3D scanning and NDT protocols to the rigorous governance of Inspection Test Plans (ITP), mitigated the challenges imposed by geographical distance and operational disjointedness. The transition from qualitative, operator-dependent assessments to quantitative, standardized data acquisition has effectively created a digital detailed record of the damage state. This allows the engineering team to make informed, condition-based decisions without the necessity of physical presence, thereby mitigating the latency risks associated with the trans-continental time difference.

Furthermore, the standardization of the overhaul process through ITPs has transformed the repair workshop from a passive execution centre into an active data generation hub. The systematic collection of dimensional shifts, tribological wear patterns, and metallurgical data provides the design department with empirical feedback. This data stream is essential for validating theoretical safety factors and refining future component generations, ensuring they are robust against the actual failure modes observed in the field.

While these activities provide a basic methodic system, they paved the way to a more structured and effective system of operation for the company. Finally, to ensure the sustainability of these newly implemented procedures, tacit knowledge was codified into structured training manuals. Initial deployment of these manuals resulted in improved operator compliance and reduction in procedural errors], addressing the human factor in maintenance engineering. By formalizing inspection logic and safety protocols, the research ensures that the high-fidelity standards established by the ITPs are sustainable and independent of individual workforce turnover. Ultimately, this operational integration demonstrates that high-quality maintenance is not merely a restorative activity but a critical extension of the engineering design process, essential for maximizing the reliability and safety of heavy machinery.

Part VI: Dry gas seals

Introduction

In the domain of high-performance centrifugal compressors, the shaft sealing system represents one of the most critical subsystems governing overall machine availability and reliability. Historically, the containment of process gas was achieved through wet oil seal systems. While effective, these systems required complex auxiliary lubrication units and posed inherent risks of process gas contamination and significant lubricant consumption. The operational landscape shifted dramatically in the late 20th century with the introduction and subsequent standardization of the Dry Gas Seal (DGS).

Today, the DGS is the industry standard for centrifugal compressors in the industrial sector, codified by standards such as API 617 and API 692. Unlike their hydraulic predecessors, Dry Gas Seals operate without liquid lubrication, utilizing the process gas itself (or a conditioned buffer gas) as the working fluid. This technological transition has not only reduced parasitic power losses due to shear forces but has also drastically minimized fugitive emissions, aligning mechanical design with increasingly stringent environmental regulations.

The fundamental operating principle of a Dry Gas Seal relies on the generation of a stable fluid film that separates the stationary and rotating components during operation. The seal consists of two primary mating faces: a rotating ring (mating ring), typically attached to the shaft sleeve, and a stationary ring (primary ring), which is held within the seal cartridge but allowed axial movement.

In a static condition, the faces are held in contact by spring force and the hydrostatic pressure of the system, ensuring a leak-tight seal when the compressor is stopped. However, the defining characteristic of the DGS is its non-contacting dynamic operation. As the compressor shaft rotates, the seal faces separate, creating a microscopic running gap, typically in the range of 3 to 5 micrometers. This separation is achieved through the hydrodynamic lifting forces generated by the specific geometry of the mating ring.

The separation mechanism is driven by a pattern of shallow grooves machined onto the face of the rotating ring (Figure 63). These grooves typically follow a logarithmic spiral or T-slot profile and do not extend across the entire face width, terminating at a specific radius known as "dam."

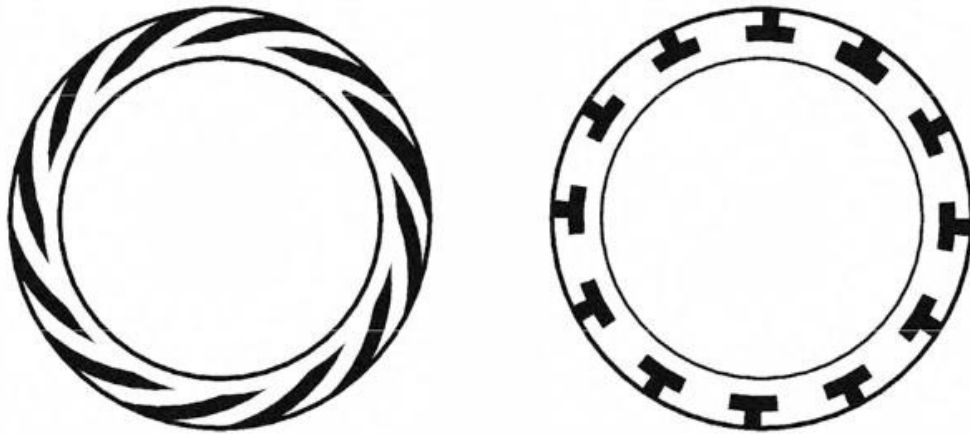


Figure 63: Spiral and T grooves [131]

As the ring rotates, gas is drawn into the grooves at the outer diameter and pumped inward towards the dam. Since the dam acts as a restriction to flow, the gas is compressed within the converging geometry of the groove. This compression results in a localized pressure rise that exceeds the closing forces (spring load plus hydrostatic pressure). When this opening force becomes sufficient, the stationary ring is pushed away, establishing the running gap.

This equilibrium is self-regulating. If the gap decreases due to axial displacement or vibration, the gas compression within the grooves increases, generating a stronger restoring force that pushes the faces apart. Conversely, if the gap opens too wide, the pressure generation drops, and the closing forces restore the optimal gap. This stiffness of the gas film is the critical parameter that prevents face contact – and catastrophic failure, during high-speed operation.

The structural integrity of a DGS depends heavily on advanced materials science. The mating rings are constructed from high-performance ceramics, typically silicon carbide or tungsten carbide, chosen for their extreme hardness, thermal stability, and high modulus of elasticity. The secondary sealing elements, which must seal the stationary ring against the cartridge while allowing for axial compensation, utilize advanced polymers or O-rings compatible with the aggressive chemistry of process gases.

DGS systems are rarely installed as single seals in critical process gas applications. They are most commonly arranged in a tandem configuration (Figure 64).

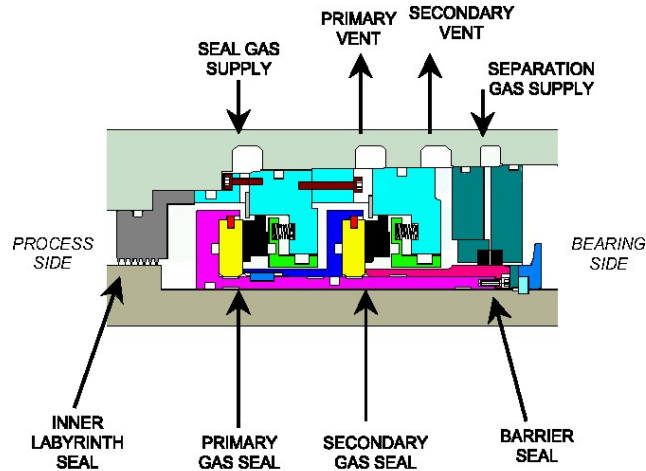


Figure 64: Dry gas seal tandem configuration

In this arrangement, a primary seal handles the process pressure, while a secondary seal allows the injection of a buffer gas, operating at a lower pressure. Lastly, a barrier seal is installed as a backup safety device. This redundancy ensures that in the event of a primary seal failure, the secondary seal can temporarily contain the full process pressure, allowing for a controlled shutdown of the machinery.

This study focuses on the critical integration between the compressor and its auxiliary support systems. Specifically, the research assesses the hydraulic capacity of the venting piping system applied to a high-pressure compressor. The safety of the installation is contingent upon the system's ability to manage catastrophic scenarios; therefore, the analysis simulates the hydraulic conditions resulting from a simultaneous failure of both the primary and secondary sealing stages.

The fundamental design requirement mandates that, under these extreme failure conditions, the venting infrastructure must be sized to efficiently evacuate the resulting high-velocity gas flow without generating excessive back-pressure. The primary objective of this assessment is to guarantee that the pressure acting upon the tertiary barrier seal (separation seal) does not exceed the maximum allowable limits (5 barG) prescribed by the Original Equipment Manufacturer (OEM). Ensuring this condition is vital to prevent the hazardous migration of process gas into the bearing housing and the lubrication reservoir, which could lead to an explosive atmosphere within the oil console.

Materials and methods

The geometric and functional characterization of the venting piping system was performed through a rigorous review of the "As-Built" documentation. The primary data sources included the Piping and Instrumentation Diagrams (P&IDs), isometric piping drawings, and the technical datasheets for all in-line instrumentation and valves.

Specifically, the flow coefficients (C_v or K_v) and opening characteristics for the pressure safety valves (PSV) and check valves were derived directly from the manufacturer catalogues. The working fluid for the simulation is Hydrogen, representing the actual process gas; its thermophysical properties were calculated at the reference operating conditions to ensure accurate estimations.

To assess the pressure buildup on the barrier seal, a numerical model was developed based on a lumped parameter approach (Figure 65). This methodology discretizes the continuous auxiliary piping system into a network of nodes (capacities) and branches (resistive elements), allowing for the calculation of pressure distribution and flow rates through the solution of mass and momentum conservation equations in a steady-state regime.

Specifically, the lumped parameter model (Figure 65) is constructed to represent the DGS system (Figure 66) under a critical condition. In this scenario both primary and secondary seal are broken, creating two chambers (A and B) represented by the C_p symbols in the model. At the same time, both seal gas and intermediate injection ports are closed due to the failure. Consequently, the process gas flows through the process labyrinth seal to chamber A. From this volume the gas splits into two directions: to the primary vent and to chamber B through the intermediate labyrinth. The primary vent is represented by the left branch in Figure 65, which contains all the concentrated and distributed losses elements. Here, the flare is represented by a back pressure that is required by the device to correctly operate and burn the hydrogen. In this branch an additional flare with lower resistance is added and controlled by the parameter k to evaluate its influence over the chamber depressurization. From chamber B the only direction available for the gas is the secondary vent port. Here, the gas is vented to a safe location and passes through a flame arrestor that provide a back pressure for this branch.

The simulations were conducted exclusively under steady-state regime conditions; consequently, the transient dynamics of system pressurization were not characterized, focusing instead on the final equilibrium pressures.

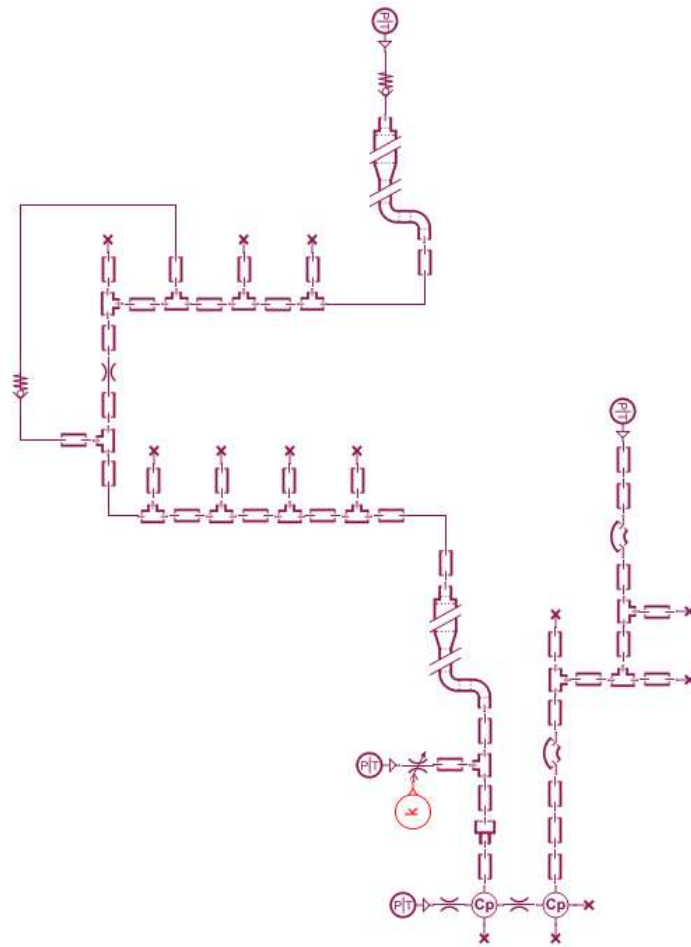


Figure 65: Lumped parameter model

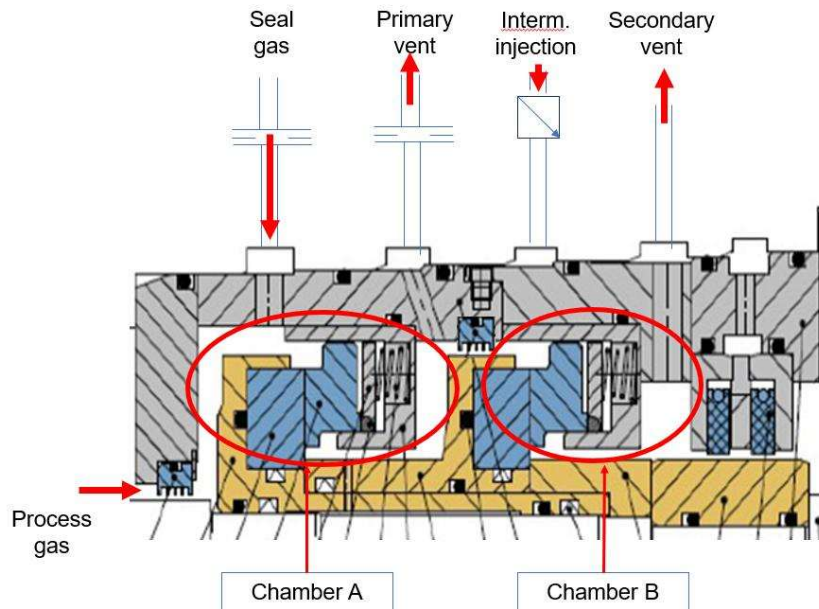


Figure 66: DGS system under analysis

The discretization of the system involved defining specific mathematical representations for each physical component. The boundary conditions were established as infinite capacity nodes representing the process Side, the flare System, and the atmosphere. Labyrinth seals were modelled as equivalent orifices, where the leakage flow is calculated using standard discharge equations adapted for compressible flow. The cavities between seals were schematized as capacitive nodes with a reference volume of one litre. However, given the steady-state nature of the analysis, the magnitude of these volumes serves primarily as topological connection points and does not influence the final equilibrium pressure results. Concurrently, piping lines were treated as resistive elements accounting for both distributed pressure losses, based on pipe length and roughness, and concentrated losses derived from bends and fittings.

The simulation imposes worst-case boundary conditions to evaluate the robustness of the sealing system. The high-pressure source is defined by the Settle Out Pressure (SOP) of 188 barg at a temperature of 120°C, representing the equilibrium condition of the compressor immediately following a trip and isolation. The discharge sinks are defined by the flare system, set at a constant back-pressure of 3.5 barG and 100°C, and the external environment at standard atmospheric conditions of 1 bar absolute and 20°C.

To account for the aging of the machinery and provide a comprehensive safety assessment, the study incorporated a sensitivity analysis based on a non-dimensional wear factor, designated as u . This parameter was varied within the range of 0.5 to 1.0 to simulate progressive degradation levels. In this definitions framework, the lower bound of $u = 0.5$

represents the nominal condition, corresponding to new labyrinth seals with full geometric integrity. The intermediate value of $u = 0.75$ simulates a degraded state, corresponding to a fifty percent wear level on the labyrinth teeth. Finally, the upper bound of $u = 1.0$ corresponds to the catastrophic failure condition (complete destruction) of the labyrinth sealing elements. In this critical condition, the model assumes that the specific resistance of the labyrinth geometry is nullified, and the residual fluid dynamic resistance is generated solely by the annular clearance between the rotating shaft and the stator bore. In addition to evaluating geometric degradation, the model was utilized to explore the efficacy of potential plant design modifications. Specifically, a virtual vent valve was integrated into the hydraulic network to assess its capacity to mitigate pressure buildup. The operational state of this auxiliary component is governed by a binary control parameter, k . In this framework, a value of $k=0$ represents the valve in a fully closed position (reflecting the current "As-Built" plant configuration), while a value of $k=1$ simulates the valve in a fully open position. This parametric approach allowed for the quantification of the hydraulic benefit provided by an additional relief path under the varying wear conditions defined by factor u .

Results and discussion

The primary criterion for validating the venting system is the mechanical resistance of the tertiary barrier seal. Based on the specific datasheet of the installed component, the maximum allowable back-pressure is limited to 6 barA. Exceeding this threshold compromises the seal's integrity, creating a direct pathway for process gas migration into the bearing housing, thereby violating the safety requirements for the lubrication system.

The steady-state simulation was firstly conducted on the existing piping configuration (corresponding to the virtual valve parameter $k=0$) to evaluate the system's response under the two defined wear scenarios.

Scenario 1: Worn Intermediate Labyrinth

In this operational case, representing a degraded but plausible condition (50% labyrinth wear), the model yielded the following equilibrium pressures (p^*):

- PSV Closed: $p^* = 5.1$ barA.
- PSV Open: $p^* = 2.9$ barA.

Under these conditions, the system remains compliant. Even in the event that the Pressure Safety Valve (PSV) fails to open ("PSV Closed"), the resulting pressure of 5.1 barA remains below the safety limit of 6 barA. The system is therefore deemed robust against standard operational wear.

Scenario 2: Destroyed Intermediate Labyrinth

This case simulates a failure where the intermediate labyrinth provides no geometric resistance.

- PSV Closed: $p^* = 10.2 \text{ barA}$.
- PSV Open: $p^* = 8.6 \text{ barA}$.

The values obtained (10.2 and 8.6 barA) represent an extremely catastrophic theoretical limit. In this simulation run, the model assumed a total loss of resistance, effectively neglecting the residual fluid dynamic friction generated by the annular clearance (the physical gap) between the rotating shaft and the stator bore.

Consequently, these pressure values should be interpreted as a conservative upper bound. In a real-world physical failure, the fluid dynamic resistance of the shaft-to-bore clearance would restrict the mass flow rate more than predicted by this simplified model, likely resulting in equilibrium pressures lower than those simulated. However, even accounting for this overestimation, the magnitude of the calculated pressure suggests that the safety margin is critically compromised in the "As-Built" configuration.

Parametric Sensitivity Analysis

To address the vulnerabilities identified in the catastrophic scenario, the analysis was extended to evaluate the efficacy of an additional dedicated vent line. Figure 65 illustrates the evolution of the barrier seal pressure as a function of the wear factor u , comparing the baseline configuration ($k=0$) with the proposed modification ($k=1$).

The trend analysis identifies a critical intersection point at a wear factor of 0.85. For the baseline case where k equals zero, the pressure curve crosses the safety threshold of 6 barA as the wear factor exceeds 0.85. This confirms that, despite the conservative nature of the Scenario 2 results, the system enters a danger zone well before total destruction. In contrast, the activation of the virtual vent valve drastically alters the system response. The simulation indicates that with the auxiliary vent open ($k=1$), the equilibrium pressure remains consistently below 5 barA across the entire range of degradation. This ensures compliance even under the theoretical worst-case assumptions of total seal loss.

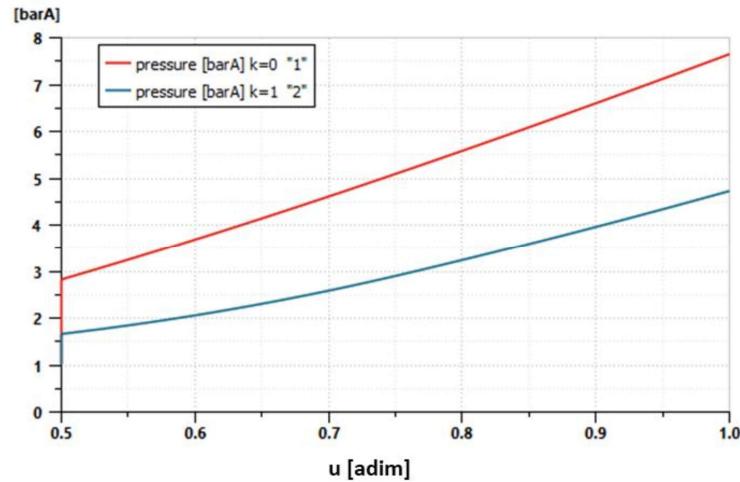


Figure 67: DGS pressure vs wear factor

The results demonstrate a clear dichotomy in the system's reliability. While the current architecture is adequate for handling standard wear, the analysis reveals potential limitations under extreme failure conditions. Although the calculated pressures for the catastrophic failure are likely overestimated due to the neglect of the shaft-bore clearance resistance, the parametric trend confirms that the system lacks the venting capacity to manage severe seal degradation. The study concludes that the installation of an additional vent valve acts as an essential engineered safeguard. Its implementation ensures that the pressure acting on the barrier seal remains decoupled from the mechanical state of the labyrinth seals, guaranteeing the safety of the oil console under all foreseeable operating conditions.

Conclusions

The research presented in Part VI addressed the critical hydraulic interaction between high-pressure centrifugal compressors and their auxiliary venting infrastructures. The primary objective was to verify whether the existing plant design could guarantee the structural integrity of the tertiary barrier seal in the event of a simultaneous failure of the primary and secondary Dry Gas Seal stages. This assessment is crucial for process safety, as the barrier seal represents the final line of defence preventing the migration of flammable process gas into the lubrication oil reservoir and the subsequent formation of explosive atmospheres within the oil chamber.

Through the application of a lumped parameter numerical model, the study simulated the steady-state response of the venting system under varying degrees of labyrinth seal degradation. The quantitative analysis revealed a distinct operational dichotomy. Under standard degradation conditions, defined by a wear factor of 0.75, the existing "As-Built" configuration demonstrates sufficient venting capacity to maintain the

intermediate chamber pressure below the safety threshold of 6 barA. This indicates that for the majority of the operational lifecycle, the standard pressure safety valves provide adequate protection. However, the projection of a catastrophic failure scenario, where the labyrinth seal loses all geometric resistance, exposed a fundamental design vulnerability. Although the simulation relied on conservative assumptions by neglecting the residual resistance of the shaft-to-bore clearance, the predicted pressures significantly exceeded the allowable mechanical limits of the barrier seal. This finding highlights that the current passive safety systems are insufficient to cope with the peak mass flow rates generated during a total seal collapse, leaving the machine exposed to a high-risk failure mode. The parametric sensitivity analysis identified the integration of an auxiliary vent valve as a decisive mitigation strategy. The results conclusively demonstrate that the activation of this secondary relief path decouples the intermediate chamber pressure from the physical condition of the upstream labyrinth seals. With the auxiliary valve in the open position ($k=1$), the system maintains the barrier seal pressure within the safe operating envelope (below 5 barA) even under the theoretical worst-case conditions. In conclusion, this study advocates for a shift in safety system design criteria for high-pressure applications. Relying solely on the anticipated service life of labyrinth seals to limit leakage rates is a strategy fraught with residual risk. The implementation of redundant, high-capacity relief paths provides an engineered safeguard that renders the system inherently safer. By ensuring that the barrier seal integrity is preserved regardless of the upstream damage state, the proposed modification significantly enhances the overall reliability and safety profile of the compression train.

Conclusions

The research presented in this doctoral thesis addresses the multifaceted challenge of enhancing reliability and operational safety in high-performance compression systems. By traversing the entire machinery lifecycle, from the preliminary phases of dynamic simulation and material selection to the operational realities of overhaul and repair, the study demonstrates that true asset integrity can only be achieved through a holistic integration of theoretical engineering and practical field feedback.

The initial investigation into the dynamic behaviour of compression trains highlighted the critical role of predictive analysis. The application of advanced computational methods, ranging from fluid-dynamic modelling of pulsation networks to the mechanical assessment of reciprocating components, proved indispensable for identifying potential failure modes before they manifest in the physical hardware. The study confirmed that acoustic resonance and valve dynamics are not merely localized phenomena but systemic issues that define the boundary conditions for the entire installation. This theoretical rigor was further underpinned by the analysis of auxiliary subsystems, specifically the flow distribution study of progressive dividers and the material selection criteria for turbomachinery. These sections established that mechanical robustness is foundational, relying heavily on the correct matching of metallurgical properties with the aggressive chemical and physical demands of the process environment.

However, a purely theoretical approach remains inherently incomplete without validation from the operational field. The research activities conducted during the international placement at Blackstone Industrial (Canada and Spain) served as the pivotal bridge between design intent and service reality. The analysis of the overhaul and repair process revealed that the historical disconnect between remote engineering offices and workshop operations acts as a significant barrier to reliability. By implementing standardized governance protocols through Inspection and Test Plans (ITP) and integrating digital diagnostic tools such as 3D scanning, the research successfully transformed the repair workshop from a passive execution centre into an active hub for high-fidelity data generation. This transition allowed for the codification of tacit technical knowledge into structured training assets, ensuring that maintenance standards remain consistent and independent of workforce turnover.

Furthermore, the implications of this research extend beyond immediate mechanical reliability to encompass broader sustainability goals. By optimizing the overhaul process and validating repairability through advanced metrology, the lifecycle of critical components is significantly extended, reducing the carbon footprint associated with

manufacturing new spare parts. The rigorous material selection criteria analysed in Part IV, combined with the overhaul techniques detailed in Part V, promote a circular economy model within heavy industry. Instead of the traditional "replace-on-failure" approach, the validated repair protocols allow for the recovery of high-value components, minimizing material waste while maintaining compliance with stringent API and NACE standards.

Looking forward, the frameworks established in this thesis lay the groundwork for the next generation of "Smart Maintenance." The structured digitization of workshop data, currently managed through the new ITPs, creates a fertile training set for future machine learning algorithms. Future research developments should focus on integrating this offline repair data with real-time telemetry from the operating machine. By automating the correlation between the "as-found" wear states recorded during the overhaul and the operational trends logged by the plant's distributed control system, it will be possible to transition from preventive schedules to truly predictive, condition-based maintenance strategies.

The operational criticality of the integrated approach was further exemplified by the safety analysis of Dry Gas Seal venting systems. The fluid dynamic modelling of catastrophic failure scenarios demonstrated that compliance with standard steady-state requirements is insufficient to guarantee total plant safety. The parametric optimization of the venting infrastructure provided a concrete example of how simulation tools, when informed by realistic operational assumptions, can drive design modifications that significantly mitigate environmental and safety risks.

Ultimately, this thesis concludes that the advancement of compression system engineering lies in the dissolution of the structural barriers separating design, manufacturing, and maintenance. The "digital twin" generated by predictive simulation must be continuously validated by the "digital thread" of maintenance data. Only by closing this information loop is it possible to evolve from a reactive repair model to a condition-based engineering strategy, ensuring that these critical industrial assets operate at the peak of their safety and efficiency potential.

Future Perspectives

The methodologies and results presented in this thesis open several pathways for future investigation, particularly in the context of the energy transition. A logical extension of this work involves applying the developed transient models to compressors operating with hydrogen or hydrogen-natural gas blends in line with the energy transition. Given the distinct physical properties of hydrogen, specifically its low molecular weight and high

speed of sound, the dynamic behaviour of compression systems will require re-evaluation, making the predictive capabilities established here even more relevant.

Additionally, future research should aim to deepen the integration of Fluid-Structure Interaction (FSI) into the simulation environment. While this study successfully linked fluid dynamics to mechanical dynamic response, a fully coupled real-time FSI analysis could provide instantaneous and direct feedback on component deformation during transient events. Finally, the validation of active surge control strategies on full-scale industrial machinery represents a necessary step to translate these theoretical and numerical advancements into standard industrial practices, ultimately driving the sector towards higher standards of safety, efficiency, and reliability.

Scientific and industry production

1. Benedetti, I., Chiavola, O., Palmieri, F., & Tundo, L. (2023, December). Analysis of Flow Rate Distribution in Progressive Divider for Reciprocating Compressor Lubrication. In *Journal of Physics: Conference Series*.
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4. Report sull'analisi dinamica di un impianto di compressione con compressore centrifugo a due stadi (CST report), 28/08/2023
5. Configurazioni di test per dosatore progressivo (CST-Dropsa presentation), 28/03/2024
6. Analisi e modellazione di una condizione critica di dry gas seals (CST presentation), 05/04/2024
7. Plant depressurization analysis (CST presentation), 17/06/2024
8. Dynamic System Analysis (CST presentation), 23/09/2024
9. Compressor Pulsation and Mechanical Analysis Report (CST report), 21/11/2024
10. Acustica dei compressori centrifughi (CST presentation), 16/12/2024
11. Reverse engineering analysis of reciprocating pump's valves (Blackstone Canada report) 13/03/2025
12. Dynamic system analysis of start-up procedure (CST presentation) 11/04/2025
13. Dynamic System Analysis (Blackstone Canada presentation), 05/05/2025
14. CFD and FEM analyses (Blackstone Canada presentation) 26/05/2025
15. Technologies for High-performance Reliability and Ultimate Strength in Turbomachinery (Blackstone Canada report) 30/05/2025
16. Literature research methodology (Blackstone Canada presentation) 30/05/2025
17. Training manuals for reciprocating compressor overhaul, maintenance and repair (Blackstone Canada) of the following components:
 1. Connecting rod
 2. Crosshead
 3. Cylinder
 4. Packing bundles
 5. Piston rods

6. Plate style valves
7. Unloaders
8. Hypercompressors radial valves
9. Hypercompressor Axial valves
18. Lista parti pompa (Blackstone Spagna document) 29/09/2025
19. Centrifugal pump overhaul and repair ITP (Blackstone Spagna document)
26/09/2025
20. Centrifugal pump technical Procedure template (Blackstone Spagna document)
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