



The University of
Nottingham

UNITED KINGDOM • CHINA • MALAYSIA

Fluids and Thermal Engineering Research Group, Engine and Combustion Division

Department of Mechanical, Materials and Manufacturing Engineering

Theoretical and Experimental Study of a Hydraulic-Assisted Turbocharger (HAT)

Annalisa Ferrante

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Abstract

Efficiency improvements for internal combustion engines are essential to counteract the harmful emissions that negatively contribute to the Climate Change issue. Hydraulic-Assisted Turbochargers (HAT) fall within these measures; according to APC roadmaps and retrofit considerations, heavy-duty vehicles are more likely to benefit from HAT as thermo-propulsion systems and hybrids are predicted to have a strong role in this transportation category. The research work carried out in this thesis aims to theoretically and experimentally investigate a HAT prototype and understand the feasibility of this solution as a turbo-lag mitigation strategy and efficiency enhancement technique. A bespoke experimental rig was designed and manufactured to mimic the HAT system in a hydraulic hybrid vehicle configuration. High-pressurised oil was stored in a 20L accumulator up to 280 bar and released through two automated solenoid valves and two nozzles into the rotating assembly. A HAT prototype inherited from a previous research project was tested; the unoptimised bearing casing caused hydraulic lock, high friction and low speeds. A free running Pelton was then used to inform the new design which included in its final configuration shrouding, open draining and ceramic ball bearings to minimise windage losses, hydraulic lock and bearing friction respectively. This final design was embedded into a standard Garrett GTX-2867r and tested in a bespoke engine-like setup. At the maximum available pressure of 250 bar and using 2x1.5mm nozzles, the maximum speed performance was 63k rpm in 0.64s producing a boost of 1.31 bar. The work at the compressor does not influence the maximum speed however it increases the time response by 10%. If compared to an Electric-Assisted Turbocharger, the HAT shows a lower time response by 42-57% on average, but its absolute maximum speed is lower by 23% on average. A 0D computational model was developed to mimic the HAT dynamic performance. The model can predict, adapting the windage losses equation with empirically deduced parameters, the maximum speeds of the assembly, however, it underestimates the time response of 12%.

Publications

Conferences

A. Ferrante, A. La Rocca, A. Cairns. *Modelling a Hydraulic Assisted Turbocharger (HAT) and comparison with experimental results*. SAE Powertrains, Fuels & Lubricants Conference and Exhibition, Poland, September 2022.

Journals

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Nomenclature

Acronyms

ACBB: Angular Contact ball bearings
APC: Advanced Propulsion Centre
BEV: Battery Electric Vehicles
BEHDT: Battery Electric Heavy-duty Trucks
CI: Compression Ignition Engine
CNG: Compressed Natural gas
CO: Carbon Monoxide
DI: Direct Injection
DP: Discharge Pressure
DTI: Dial Test Indicator
EAT: Electrically-assisted turbocharger
EGR: Exhaust Gas Recirculation
EHV: Electric Hybrid Vehicles
EPA: Environmental Protection Agency
EU: European Union
EV: Electric Vehicles
FC: Fuel Cell
FCEV: Fuel Cell Electric Vehicles
FI: Forced Induction
FRP: Free Running Pelton
GHG: Greenhouse Gases
HAT: hydraulically-assisted turbocharger
HC: Hydrocarbons
HDV: Heavy-duty Vehicles
HHV: Hydraulic Hybrid Vehicles
ICE: Internal Combustion Engine
IVP: Initial Value Problem
KERB: Kinetic Energy Recovery Boosting System
KERS: kinetic energy recovery systems
LDV: Light-Duty Vehicles

Nomenclature

MGU: motor/generator unit
NA: Naturally aspirated
NEDC: New European Driving Cycle
NPSH: Net Positive Suction Head
NYCT: New York City Truck
ODE: Ordinary Differential Equation
OTP: Original Turbocharger Prototype
PEM: Polymer Electrolyte Membrane
PM: Particulate matter
RB: Regenerative braking
REAT: Regenerative Electric-Assisted Turbocharger
RHAT: Regenerative Hydraulic-Assisted Turbocharger
SDP: Static Discharge Pressure
SI: Spark Ignition Engines
TPP: Turbo-Pelton Prototype
TPS: Thermal Propulsion Systems
TPSR: Turbo-Pelton Stand Rig
UKRI: UK Research and Innovation
UPS: United Parcel Service
VDE: Variable Displacement Engine
VGT: Variable Geometry Turbocharger
VMT: Vehicles miles travelled
VVT: Variable Valve Timing
WHO: World Health Organization
WLTC: Worldwide Harmonised Light Vehicles Test cycle

List of symbols

A : exit area of the nozzle [m^2]
 A_p : pipes area [m^2]
 B : Bucket width [m]
 C_0 : oil jet at nozzle exit [m/s]
 C_1 : oil jet speed impacting at the centre of the bucket [m/s]
 C_2 : oil jet exit absolute velocity [m/s]

Nomenclature

C: polytropic expansion constant
 C_{1p} : constant in hydraulic accumulator theoretical approach
 C_{2p} : constant in hydraulic accumulator theoretical approach
 C_3 : initial condition constant in hydraulic accumulator theoretical approach
 C_C : contraction coefficient
 C_D : nozzle discharge coefficient
 C_m : torque coefficient
 C_p : pressure coefficient in cavitation theoretical approach
 C_{pm} : differential equation constant in Pelton dynamic theoretical approach
 C_{pt} : differential equation constant in hydraulic accumulator theoretical approach
 C_s : basic static load [N]
 C_{vv} : valve flow coefficient
 c : radial clearance in plain bearings [m]
 c_p : heat capacity at constant pressure [J/molK]
 c_v : heat capacity at constant volume [J/molK]
 D : diameter of the pipe [m]
 D_n : diameter of the nozzle [m]
 $D_{c/t}$: diameter compressor/turbine wheel [m]
 d_m : bearing medium diameter [m]
 E_{BR} : Braking Energy [J]
 E_{TR} : Tractive Energy [J]
 F : force [N]
 F_a : pressure loss at the accumulator
 F_{ax} : axial force [N]
 F_b : friction force in bearings [N]
 F_r : radial force [N]
 F_p : Force impacting on Pelton bucket [N]
 F_s : static equivalent load [N]
 F_{TR} : Tractive Force [N]
 f : friction coefficient
 f_l : hydrodynamic coefficient of friction
 f_s : metal-to-metal coefficient friction
 g : gravity acceleration [m/s²]

Nomenclature

H_a : head on the surface of the liquid in the tank
 H_f : total friction loss in the suction pipe
 H_s : vertical distance of the free surface of the liquid above the centre-line of the pump impeller
 H_{vp} : vapour pressure of the liquid at the pumping temperature
 h : bucket height [m]
 I : inertia [kgm^2]
 K : coefficient for pressure losses
 K_a : Pressure loss coefficient of the accumulator
 K_{cor} : pressure loss correction coefficient
 K_{GC} : coefficient for pressure losses in gradual contraction
 K_{sc} : coefficient for pressure losses in sudden contraction
 K_v : coefficient for valve pressure losses
 k : Vogel coefficient [mPas]
 k_t : empirical coefficient for Tondell windage losses equation
 L : length of the pipeline [m]
 L_b : bearing length [m]
 l' : length of the half bearing [m]
 $L_{c/t}$: length of compressor/turbine blades [m]
 M_{hv} : hybrid vehicle mass [kg]
 m : mass [kg]
 $\dot{m}$: mass flow rate [kg/s]
 m_{egr} : exhaust gas recirculation mass [kg]
 m_r : rotational mass [kg]
 $m_{f,hv}$: fuel consumed for tractive purposes [kg]
 $m_{f,TR}$: fuel consumed for tractive purposes [kg]
 M_t : total friction torque bearings [Nm]
 n_r : number of crank revolutions for each power stroke
 n_{teeth} : number of teeth
 N : rotational speed [rpm]
 N_e : engine rotational speed [rpm]
 N_n : number of nozzles
 N_t : turbocharger rotational speed [rpm]

Nomenclature

P_b : brake power [W]
 P_f : friction power loss in journal bearings [W]
 P_w : windage losses power [W or HP]
 P_{TR} : Tractive Power [W]
 p_{1p} : reference static pressure of the pump [Pa]
 p : pressure [Pa]
 p_s : bearing fed pressure [Pa]
 p_{st} : static pressure [Pa]
 p_1^T : inlet total pressure pump [Pa]
 p_v : vapor pressure [Pa]
 Q_{HV} : fuel lower heating value [J/kg]
 Q_s : bearings flow rate [m³/s]
 R : specific gas constant [J/K mol]
 R_a : Pelton wheel radius [m]
 $R_{c/t}$: radius of compressor/turbine wheel [m]
 Re : Reynolds number or rotational Reynolds number
 r : radius of disc or gear or pitch circle Pelton [m]
 r_b : Journal bearings radius [m]
 $r_{(F/A)}$: fuel/air ratio
 r_{tr} : tire rolling resistance coefficient in tractive force
 r_w : effective rolling radius of a wheel assembly in braking energy [m]
 s : axial clearance between disc/wheel and casing [m]
 T : temperature [K]
 T_b : roller bearings friction torque [Nm]
 T_f : final torque [Nm]
 T_{ft} : total ball bearing friction torque [Nm]
 T_l : resistant torque due to bearing load [Nm]
 T_p : Pelton Ideal Torque [Nm]
 T_v : resistant torque due to the fluid friction of bearing element in contact with the lubricant
 T_w : windage loss torque [Nm]
 T_{wd} : windage loss torque disc [Nm]
 T_{wb} : windage loss torque bucket [Nm]

Nomenclature

t : time [s]
 t_{sim} : simulation time [s]
 U : constant speed of translating bucket of the Pelton [m/s]
 u : velocity in the pipes [m/s]
 u_{1acc} : oil velocity at the accumulator [m/s]
 $\dot{v}$: volumetric oil flow rate [m³/s]
 v : linear velocity [m/s]
 v_{jet} : oil jet velocity at the nozzle [m/s]
 v_s : reference velocity in the expansion/contraction [m/s]
 V : volume [m³]
 V_{sw} : Swept volume [m³]
 V_t : Volume of oil ejected through the nozzle in the selected timestep
 W : relative velocity [m/s]
 x : bucket active length [m]
 X_s : coefficient for bearing friction
 Y_s : coefficient for bearing friction
 z : elevation [m]

Greek letters

α : pressure-viscosity coefficient of lubricant [m²/N]
 α : angle in contraction/expansion
 α_b : metal-to-metal contact constant
 α_p : angular acceleration [rad/s²]
 β : volumetric expansion coefficient
 β_2 : bucket exit angle [radian]
 γ : polytropic coefficient
 γ_w : specific weight [m³/kg]
 Δp_f : friction pressure losses [Pa]
 Δp_{gc} : gradual contraction pressure losses [Pa]
 Δp_l : various losses [Pa]
 Δp_{se} : sudden expansion pressure losses [Pa]
 Δp_{sc} : sudden contraction pressure losses [Pa]

Nomenclature

Δp_v :	valve's pressure losses [bar]
ε :	Axial clearance in plain bearings [m]
ε_B :	electromotive force
ε_b :	eccentricity in bearings [m]
η_{BR} :	braking efficiency
η_{dt} :	drivetrain efficiency
η_f :	fuel conversion efficiency
η_{pc} :	power conversion efficiency
η_v :	volumetric efficiency
θ_1 :	Vogel coefficient
θ_2 :	Vogel coefficient
φ :	volume or mass fraction oil
ρ :	density [kg/m ³]
ρ_{oil} :	oil mass density [kg/m ³]
ρ_{air} :	air mass density [kg/m ³]
σ :	cavitation number
σ_i :	cavitation inception number
σ_h :	normal load per length of the tribological contact
μ :	dynamic viscosity [Pas]
μ_{oil} :	oil dynamic viscosity [Pas]
μ_{air} :	air dynamic viscosity [Pas]
μ_φ :	tangential velocity [m/s]
ν :	oil kinematic viscosity [m ² /s]
τ :	Shear stress [Pa]
τ_t :	tangential shear stress [Pa]
$\varnothing$:	Diameter [m]
ω :	angular speed [rad/s]

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1 Introduction

1.1 Context of work and Research value

The World population is expected to reach 9.7 billion people by 2050 [1]. Energy consumption, following this trend, increased as well as of course its production and it is intrinsically linked to environmental pollution. Fossil fuels represented 2019 highest source of energy production with almost 79% of the global primary energy consumption and accounting for 94% of the global emissions of CO₂ [2]. In this setting, the transportation industry represents one of the main contributors to Climate Change, increasing the global greenhouse gas emissions (GHG) with 16.9% of input [3]. Out of this percentage road transport account for $\frac{3}{4}$ [4] and, in the UK specifically, it is responsible for 23.4% of the overall tailpipe emissions [5]. In 2016 Transportation was also responsible for 7.87 billion tons of CO₂ emitted representing the second largest sector responsible GHG emissions [6]. To address this issue, all the current laws and directives stress the environmental hazard tied to pollutant gas emissions in the atmosphere caused by fossil fuels. To achieve the main goal of the Paris Climate Change Agreement [7], which is to maintain the global average temperature below 2°C above pre-industrial levels, it is mandatory to accelerate the deployment of low-emissions vehicles. The Transportation industry is then gradually but consistently migrating to zero-emissions vehicles, decarbonization of fuel energy production processes, and efficiency improvement techniques. Carmakers are acting to respect the emissions targets to follow the changing market landscape to avoid risking penalties for missing these objectives.

COVID-19 pandemic highlighted the importance of the transportation industry and particularly heavy-duty vehicles (HDV) which played a major role in ensuring the continuous flow of essential goods for the population, while most of the other services had to undergo a massive production stop [8]. Freight travels are increasing in urban areas worldwide as a result of better living conditions and increasing trends in goods production and consumption. Heavy-duty trucks usage has indeed increased exponentially during the last decade even before the pandemic hit; according to the Department of Transportation in the USA, the increase of heavy trucks in VMT (vehicles mile travelled) in the period

of 2010-2016 was +0.5% whereas in more recent years, in half of the time, (2016-2019) it was 12 times more, reaching almost +6% [8]. Looking at 2050, it is predicted that Heavy-duty vehicles will increase by 2.6 times reaching 64 million vehicles [9].

HDV need then to be prioritised to create solutions for meeting environmental requirements; diesel engines accounted for 97.9% of all new medium and heavy trucks in 2019 [10], representing the almost totality of the transportation means in long-distance road travels. To reduce emissions, up to -60% of GHG related to global transport, the guideline strategy sets in first place improvement of vehicle efficiency of modern trucks, followed by the implementation of systemic improvements in road freight operation and logistics; lastly the shift to alternative fuelled trucks is recommended [9]. Battery electric Heavy-Duty trucks (BEHDT) are not a feasible alternative to consider at the moment because of the limitations of battery technology and the missing infrastructure to support the charging [11]. Furthermore, Electricity from batteries has low energy density which means the more power required, the heavier are the batteries. Additionally, charging infrastructures for BEHDT do not yet exist making them impossible to be framed for the long-haul operating conditions. The electrification for heavy-duty is then considered an inconsistent route to undertake; The G20 in 2020 highlighted as well how even in light-duty vehicles (LDV) the policy-driven improvement in the fuel consumption of Internal Combustion Engine (ICE) vehicles could have a greater effect on oil consumption than vehicle electrification [12]. In terms of oil demand from LDV, the impact from improvements in fuel consumption, looking at a 2040 horizon, would be -16% compared to just -4% of the impact from penetration of electric vehicles (EV) [13].

The research panorama in automotive nationwide is however highly dependent on UK fundings which were addressed for improvement for electrification of vehicles; in April 2021 the UK Research and Innovation (UKRI) announced that the “Driving the Electric Revolution” mission, with an investment in research of £12.7 million, was the necessary catalyst for building billions of pounds of new power electronics, motor and drives products in UK by 2025 [14]. This funding orientation led to a significant number of papers for investigating the electric source for traction purposes. The passenger car sector is then widely covered in

terms of research interest, however heavy-duty vehicles still required a valid alternative to traditional ICE that Academia needs to contribute to achieve. This is supported by The World Health Organization which strongly advises the shift to cleaner heavy-duty diesel vehicles as one of the main strategies to reduce air pollution and consequently health problems worldwide [15].

The aforementioned considerations are in favour of the thesis' worth and significance. The study of a hydraulically assisted turbocharger (HAT) prototype is related to improving ICE efficiency techniques and emissions reduction strategies supported by policies and trends to be adopted in heavy-duty fleets looking at 2050 [16]–[18]. Few studies are available simulating and testing HAT [19]–[23]; the research project adds valuable information for the challenges involved with its implementation in a Hydraulic Hybrid Vehicle (HHV) configuration and the comparison with the main technology competitor represented by electric-assist turbocharger.

1.2 Road transportation landscape

The Advanced Propulsion Centre (APC) roadmaps [17], [18] are used as main references to help identify the future of technology for transportation and frame the important contribution of the research subject to the available literature.

The past decades saw a significant change in the auto industry: over 30% of new car sales are projected to be zero emissions and plug-in hybrid by 2030 and electric vehicles could become as affordable as diesel and gasoline cars within 2022 [24]. Sixteen of the world's largest publicly-listed automotive companies, that represent 79% of the global passenger vehicle market by sales volume, are on business readiness for a transition towards low-carbon vehicles [25], setting a clear trend for passenger cars. Politics is acting accordingly: in 2020 UK decided, as an approach to lower its contribution to Climate Change, to ban, before 2030, the new production of cars and vans based on Thermal Propulsion Systems TPS. From 2035 all new cars and vans will be fully zero emission at the tailpipe [26]. Although this clear transition from traditional vehicles to electric or low-emissions vehicles is happening, it is taking place gradually. What emerges from the APC roadmaps is that there is no one clear winner: Battery Electric Vehicles (BEV), Hybridisation, Alternative Fuels and TPS will all power vehicles to 2050. The UK policy [26] applies only to cars and vans and

not to the heavy-duty industry; the latter will still need the TPS as the main propulsion system. The environmental impact of these vehicles is expected to be under greater scrutiny regarding fuel generation and suppliers will be pressured to decarbonise the fuel's derivation from primary sources. Moreover, the focus on efficiency improvement of ICE vehicles is encouraged taking into account that although the main part of road transport will be electrified, not all vehicles will be within the next 20 years [18].

While EV and Electric Hybrid Vehicles (EHV) face obstacles (charging options, back-end emissions, etc.), in the short to medium term another technology is a more cost-effective route to lower CO₂ and pollutant emissions: mild and full HHV use a hydraulic accumulator to store recovered energy instead of batteries powering the vehicles thanks to a high-pressure hydraulic pump. Plug-in hybrids are also considered in the solution landscape, however, they need to evolve to be compliant with greater zero tailpipe emissions capability. If the cost of reinforcing the electricity grid becomes too high, Fuel Cell Electric Vehicles (FCEV) could become more competitive as a hydrogen infrastructure proves to be more cost-effective, other than the potential of storing surplus energy generated from renewable sources. Hydrogen combustion engines are also promising and can provide a cost effective net-zero emissions solutions for many heavy-duty and off-highway vehicles running on net zero fuels [17].

Summarizing, environmental regulations dictate the criteria for how the transportation industry will move towards 2050; they address the reduction of pollutants as well as technology specifics requirements (Figure 1.1). A gradual but slow process is taking place to move into the non-dependency of the transportation industry from fossil fuels.

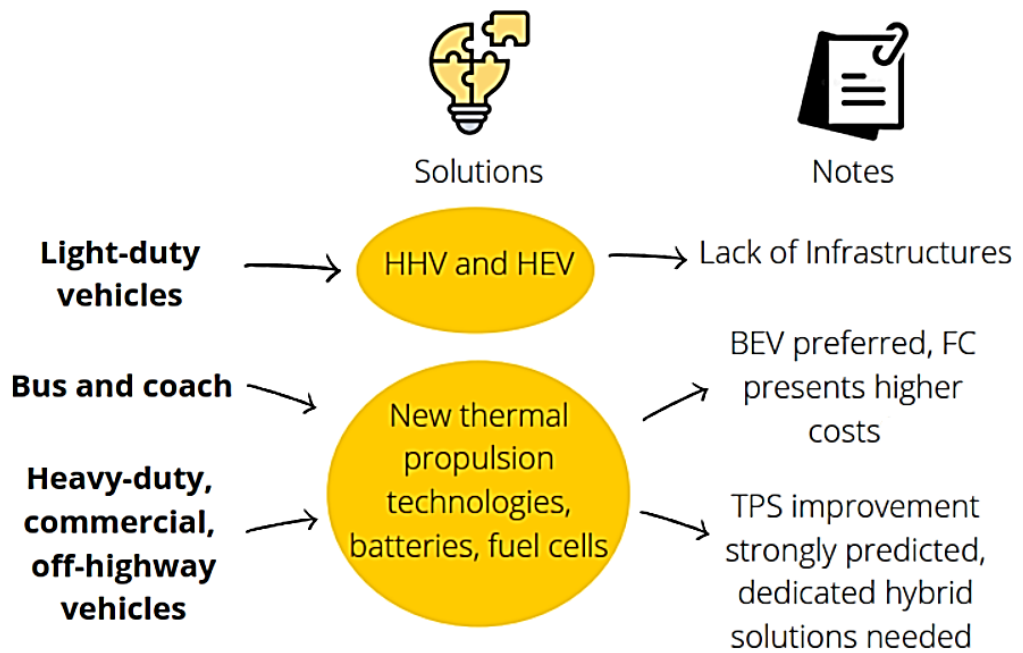


Figure 1.1 Outcomes from APC roadmap for 2050 vehicles prospectus

1.3 Heavy-duty impact and Regulation

For heavy-duty applications and off-road machinery, the most attractive and feasible solution remains the Diesel engine. 97.9% of all new medium and heavy trucks in 2019 were diesel-powered, whereas only 0.3% of the vehicles were full/hybrid electric vehicles (HEVs)[10], [27]. Medium and heavy-duty trucks accounted for 270 Mt of CO₂ emissions, compared to just 40 Mt in light-duty [27]; global energy demand for transportation and the ratio of HDV to LDV is expected to increase [28] and a shift towards a solution for HDV is required because of the great amount of pollution the road freight transportation represents.

Impact of Diesel engines, however, is still a challenge for car manufacturers, because their effect on Environment and Health. The increasing level of some of the substances contained in their exhaust gases is related to high production of GHG which modify the natural Greenhouse effect of the Earth. This increase on average global temperature reached +0.8 °C above pre-industrial levels [29]. In addition to this, in the UK it was estimated that long-term exposure to man-made air pollution has an annual impact on shortening lifespans equivalent to 28,000-36,000 deaths [30]. Social care cost for diseases associated with pollution in UK

could reach £5.3 million, and it increases to £18.6 million considering diseases with evidence of indirect association [30].

Reducing emissions from road transportation represents then a necessary step to reduce air pollution levels and the burden of long and short-term illnesses. In this sense, the international governments guidelines set the requirements for permissible exhaust emissions standards (EURO standards) according to European Union EU 2019/631 [31]. Heavy-duty emissions standards furthermore are consolidated on the Directive 2005/55/EC [32] [33]. As a reference example, EURO VI is expected to have -98% of Particulate Matter (PM), -67% in CO, -88% in HC and -95% in NO_x compared with EURO I. UK Government has identified road transport as the largest source of CO₂ emissions. To tackle the emissions from transportation [34], UK Government published in 2019 The Clean Air Strategy [30] to comply to the limits recommend by WHO (World Health Organization) [15], [35], [36]. The strategies shown are supposed to halve the number of people living in locations above the PM_{2.5} guidelines by 2025 [37]. The government is actively investing in reaching those goals: in partnership with the UKRI it has launched a joint research program of £19.6 million to promote the development of cleaner technologies together with a £5 million Small Business Funding competition to promote and procure industry-led R&D. In 2019 a £50 million pilot program of loans for small and medium enterprises was announced from the same corporation, aiming to increase the uptake of innovative technologies and the availability of testing grounds for innovators. The active actions have been mainly on city transportation, where the UK provided already £119 million for the purchase of low emission vehicles and £27 for retrofitting new buses. These fundings aim to achieve a reduction of -15% by 2025 in CO₂ emissions in both cars and vans [38] further decreasing it by 2030 to -37.5% for cars and -31% for vans.

HDV zero-emission options, although investigated, are not yet widely available. Solutions for reducing real-world pollutant emissions from new diesel vehicles are still needed; techniques like Rightsizing of the ICE can improve the fuel economy performances, lower the emissions and assist the implementation of H₂ hybrid ICE engines as long-term solution.

1.4 Road Transports Developments

To mitigate the effect of powertrain emissions from exhaust gases, several strategies like combustion control, turbocharging and after-treatment have been proposed following legislation-driven requirements. One category among them is “*Decarbonization*” which is referred to reducing “carbon intensity” decreasing the CO₂ output from fossil fuels. The reduction of CO₂ is currently attempted through electric or hybrids system or alternative fuel source (e-fuels) which are often associated with KERS (Kinetic Energy Recovery systems); these operate converting the braking power of the vehicles in usable and storable energy. It reflects in an improved fuel consumption, efficiency increase and less emissions at same operating conditions. In EHV the braking power is translated into electric energy that is stored in batteries; in HHV the used storage is a hydraulic accumulator.

The understanding of the potential energy saved and regenerated in KERS system is of great interest to explore the viability of possibilities to reduce GHG [39]. The European Union estimated a fuel economy improvement of 17% in real-life measurements thanks to Regenerative Braking (RB) alone and this value goes up to 50-60% when hybrid drivetrain systems are included [40]. A study on RB effect on fuel economy for a battery storage-based bus showed an improvement of 39.7% [41]. In another study an alternative hydraulic storage was shown to recover up to 45% of the total kinetic energy absorbed in braking [42].

Electro-fuels and biofuels represent a carbon-neutral alternative for traditional fuel. E-fuels impact on the environment is however not properly computed as the electricity needed for their production should come exclusively from low environmental impact renewable resources to be considered carbon-neutral. Biofuels depend on the extent to which biomass can be provided, on advances in conversion technology, on the competing demands for bioenergy and land and the feasibility of other sources of carbon-neutral fuels [27].

Electrification and hydrogen technologies can see some of the Road transports decarbonised within decades [43]; the IEA (International energy agency) scenario drawn in 2020, outlines that net-zero CO₂ emissions from global energy will be reached by 2040 for motorcycles, by 2050 for rail, by 2060 for small trucks. However, some other vehicles will be more difficult to decarbonise, for

example, emissions for cars and buses will not be eliminated by 2070 (Figure 1.2) [28].

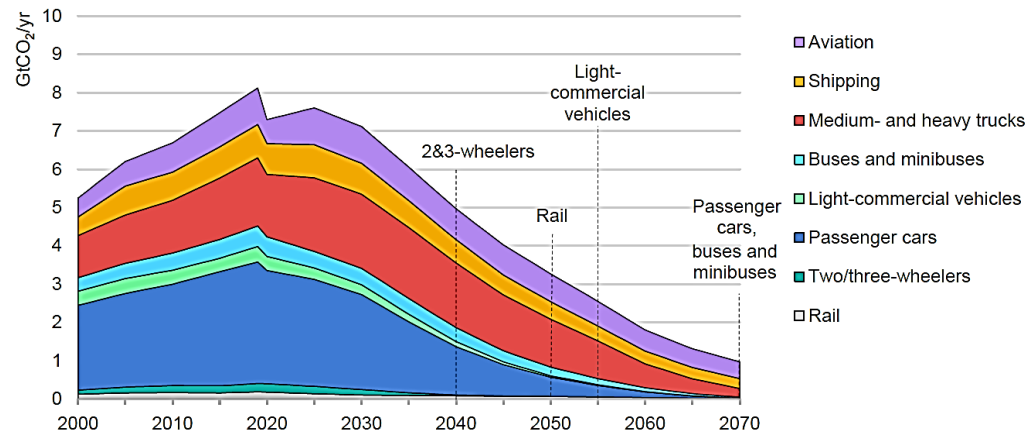


Figure 1.2 Global CO₂ emissions in transport by mode in the Sustainable Development Scenario 2000-2070 (IEA 2020) [28]

Note: dotted lines indicate the year in which various transport modes have largely stopped consuming fossil fuels and hence no longer contribute to direct emissions of CO₂ from fossil fuel combustion.

A strategy applied efficiently to ICE to lowering emissions without modifying the fuel source is the *Rightsizing*, an improvement of *Downsizing*. The latter involves reducing the capacity of the engine drive units and neutralise the resulting loss of torque through turbochargers or Direct Injection of fuel. The same vehicle power is obtained using a smaller engine operating with fewer cylinders reducing the fuel consumption. However, the costly components connected with the downsized drive units or injection systems are prone to malfunction due to high values of mechanical stress coefficient. The savings made due to the reduced fuel consumption take a back seat to the expensive repairs. *Rightsizing* consists in optimizing the matching of the engine cylinder capacity to the size of the vehicle. The theory background that makes rightsizing a winning strategy is that the combustion process has defined parameters that influences the energy extracted from the fuel. The technologies used are the same as the downsizing but with a focus on thermodynamic and geometric optimization instead of a mere reduction in size. Direct Injection (*DI*), Variable valve timing (*VVT*) and Forced Induction (*FI*) techniques are used, the focus in this paragraph is given just to the last category.

In Naturally Aspirated (NA) engines air induction is solely produced by the pressure difference between the air intake system and the backward motion of the piston into the cylinder. Incrementing the effective compression ratio increases air density improves power and torque and it is possible thanks to supercharging and turbocharging. The main disadvantage of the first strategy is the power share subtracted from the engine which represents a parasitic loss for the engine. In *Turbocharging* instead, the kinetic power is provided from the high-temperature exhaust gases. Turbocharging is often used in diesel engines as air capacity is an important constraint on the amount of fuel that can be burned and therefore on the engine's power, especially of interest in large engines [24]. The rightsized ICE with the most recent turbocharging solution, performs more efficiently with lower fuel consumption and emissions [44]–[47].

Every turbocharger needs to be matched to the engine using the efficiency map of its compressor; the compressor needs to be small enough to operate efficiently at low speed and big enough to still operate efficiently at peak engine power. The compromise in matching the turbocharger comes with a common issue in fixed geometry turbocharger called Turbo-Lag. The latter is the time required by the engine to produce enough exhaust gases pressure to spool up the device to the minimum rotational speed for the boost. The results of this delay are limited instantaneous torques; in a high-speed diesel engine applying full load in 1s, the time response can be up to 4s [48]. Traditional solutions to this are wastegate valve, multi-stage turbos and Variable geometry turbo, however some of these lower the engine efficiency. An innovative alternative are assisted turbochargers: an external source, hydraulic or electric is used to aid the turbines assembly to reach the desired speed value and increase the time response.

1.5 KERB-HAT: Hydraulically-assisted turbocharger for heavy-duty vehicles

The thesis is based on a project involving a theoretical and experimental investigated hydraulically-assisted turbocharger framed in the successful strategy of rightsizing in ICE. KERB-HAT (Kinetic Energy Recovery Boost with Hydraulic Assisted Turbocharger) consists of a standard turbocharger equipped with a hydraulic Pelton turbine mounted on the same shaft of the gas turbine and the compressor. High-pressurised oil is injected through bespoke nozzles in an innovatively designed casing optimised to avoid hydraulic lock and

maximise the rotational speed and time response; the basic schematic of its functioning is shown in Figure 1.3.

According to APC roadmaps, heavy-duty vehicles are more likely to benefit from HAT as thermo-propulsion systems are far from being replaced and the HHV application is realistic for this category of vehicles. Through turbocharging, rightsizing of the engine is possible and a reduction in GHG is achievable; the added value of KERB-HAT is to address furthermore the issue of Turbo-Lag [49]–[51].

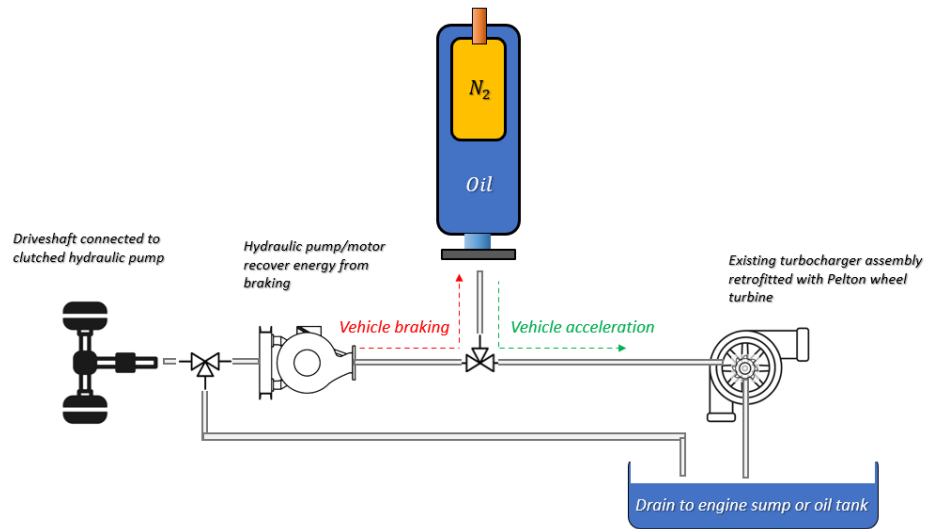


Figure 1.3 Basic schematic of KERB-HAT functioning

1.6 Aim and Objectives

The overall aim of this thesis is to understand if a hydraulically-assisted turbocharger can be a feasible solution to be used as turbo-lag mitigation strategy. This is achieved focusing the experimental work on acquiring rotational speed data to research to prove feasibility (TRL4) and provide the base to Demonstrate the technology (TRL5-TRL6). The objectives are summarised below.

1. Design and manufacture a bespoke experimental rig to investigate a hydraulically-assisted turbocharger prototype performance.

2. Test the proof of concept for a HAT in different experimental conditions, identifying the set of parameters to address and modify for a new prototype optimization design.
3. 3D Design of an optimised prototype for a HAT, manufacture and test it in the basic standard configuration and investigate its performances.
Replicate real-functioning conditions by designing and manufacturing a turbo gas test stand to map performance parameters in engine-like conditions such as time response, maximum rotational speed, compressor boost and mass flow rate.
4. Produce a computational model to replicate the prototype operation in basic condition by modelling the hydraulic accumulator discharge and relating it to a hydraulic wheel dynamic including bearing and windage losses of the complete assembly.

1.7 Thesis layout

The research project is a development of a previous PhD research project in Brunel University [49]. The work concerned the design, analysis and basic demonstration of a hydraulically assisted fixed geometry turbocharger aimed to overcome the turbo-lag. The PhD researcher carried out a preliminary computational analysis to select the specification for the turbine and the geometry associated with the hydraulic parts. The experimental work investigated a basic configuration with a Pelton wheel mounted on a shaft and equipped with an inertia disc to replicate the ideal turbocharger momentum. The main outcome was the achievement of 90,000 rpm with hydraulic boosting of 70% efficiency. The study showed the potential for this technology and as a result of the PhD thesis at Brunel, a prototype was manufactured but never tested. Starting from this, the summarised steps of the current research project were:

- I. Test the Original Turbocharger Prototype to assess criticalities.
- II. Manufacture and test a basic configuration, to collect data to optimise a new prototype.
- III. Design and manufacture a new HAT prototype.
- IV. Test the new prototype stand-alone.
- V. Test the new prototype in engine like condition.

VI. Produce a computational model to predict and compare results.

The thesis is structured in eight chapters, each of the ones includes a brief introduction and a summary at the end to gather the main outcomes.

After this first chapter in which the context of work is introduced, Chapter 2 shows, as part of the Literature review, an overview on efficiency improvement techniques, focusing on energy recovery systems and application on hybrid vehicles. The rightsizing strategy is introduced and as part of that, the Turbocharger technology is discussed in details. The turbocharger-lag issue of these devices and its potential solution are shown focusing on assisted turbochargers.

Chapter 3 describes the Experimental Set-up and Methodology. The experimental rig designed to test the HAT is shown and the Design of Experiment is customised for the application for the reliability of the experiments. The shakedown of the rig is shown including the acquisition data tests and hydraulic lock tests.

Chapter 4 describes the computational model designed to replicate the experimental conditions and to predict results. The mathematical representation for the hydraulic accumulator oil pressure source is described together with the Pelton wheel dynamic and the related losses of bearings and windage. This is a powerful tool to predict experimental performances and it is used in the Discussion Chapter 7 for comparisons and predictions.

Chapter 5 includes the investigation of the original prototype and a simplified free running Pelton (FRP) system to select the optimization criteria for the new prototype. The limitations of the original prototype are assessed and the FRP results correlate the performance improvements in regards to maximum speed reached and time response. Comparisons with the computational model verifies the reliability of the latter to predict the efficiency of the system.

Chapter 6 includes the experimental investigation on the new turbocharger prototype including the optimization criteria and the turbocharger gas stand

testing configuration explained. Time response and compressor boost are extrapolated and used for the Discussion.

Chapter 7 includes the Discussion of main results and the consideration for the automotive application. The computational model is used to compare and predict testing results for different configuration and limitation of the model are highlighted. Optimizations are discussed and the application matching for KERB-HAT is screened in regards of ICE performance parameters framing the prototype organically in the vehicle.

Chapter 8 summarises main conclusions and innovation of the research, highlighting the contribution to the existing research issue and providing opportunities and reasons for future works.

1.8 Summary

The takeaway messages from the Introduction Chapter are the following:

- Environment pollution contributes in Climate Change global issue; leaders worldwide are working towards legislation focusing on reducing emissions in energy production systems. All the research community is called to find solutions to contribute.
- The transportation sector is one of the largest responsible for GHG harmful emissions and the Road Transports account for 75% of these emissions [4].
- Electrification and hydrogen technologies can see some categories in the Road transports decarbonised within decades [23], however long-distance large trucks and aviation requires a larger power which is not sustainable with electric power sources.
- The UK ban for new production of diesel/petrol vehicles does not apply to heavy-duty [20]; improving efficiency and lowering emissions techniques will still be required.

Introduction

- Rightsizing is proposed as a successful strategy to lowering emissions in ICE, especially in Diesel engines which are widely used in heavy-duty transportation.
- The introduction of a Hydraulic-Assisted Turbocharger is a meaningful, innovative and high-potential strategy in the rightsizing of Diesel ICE's for heavy-duty vehicles, contributing to the aim of lowering emissions from transportation and reducing the greenhouse effect.

2 Literature review

2.1 Introduction

The literature review chapter provides knowledge insights to understand the research subject. The aim of the chapter is to frame kinetic energy recovery systems, hybrid vehicles and turbocharging solutions for heavy-duty vehicles. It splits into three main sections other than the introduction to the chapter. In section 2.2 an overview of efficiency improvement techniques is given for heavy-duty vehicles. The emissions related to heavy-duty vehicles are listed and the rightsizing solutions for their mitigation are introduced. Turbochargers are compared to superchargers highlighting the advantages of the firsts. In section 2.3 the turbocharger technology is discussed in detail as a low-emissions and rightsizing strategy for diesel engines. The turbo-lag issue is described, and solution strategies currently offered in the industry are listed. A focus is carried out on assisted turbochargers and a comparison between electric and hydraulic sources is provided. In section 2.4 KERS (kinetic energy recovery systems) are discussed. HHV are explained in this optic with a brief comparison with EHV systems. Finally the literature available on windage losses described as they represent the main source of friction in the rotating system.

2.2 Overview on turbocharging for heavy-duty vehicles

2.2.1 Heavy-duty vehicles and emissions

The Introduction chapter highlighted how Climate Change is an active and progressing challenge for the industry. Worldwide services and products are being adapted to limit the harmful effect of pollution produced by industrial processes. In the transportation industry, the main issue is related to emissions from TPS vehicles which will still be used in heavy-duty transportation [18] [50].

Emissions are contained in the exhaust gases of the ICE as a result of incomplete combustion of the fuel. The ones that have the major influence on GHG substances are [29]: Hydrocarbons (HC), Carbon monoxide (CO), Nitrogen Oxides (NO_x), Particulate Matter (PM), and Carbon dioxide (CO₂). *Hydrocarbons* are stated in terms of CH₄ equivalents and they are mainly caused

by the insufficient temperature near the engine cylinder wall which is lower compared to the centre of the cylinder. This leads to the production of this pollutant when part of the fuel has not reacted with the oxygen. These emissions usually occur at light loads where the lean air-fuel mixing does not enable the flame speeds to complete the combustion during the power stroke. *CO* derives from the not complete oxidation in the combustion process, which results in partially burned fuel. Because of the nature of the Diesel engine which has a consistently high air-fuel ratio, the formation of *CO* is minimal in these types of engines, however, it can be produced if the droplets of diesel fuel are too large or if turbulence occurs during the combustion. *Nitrogen Oxides (NO_x)* emissions are generated in Diesel engines, by the reaction at high temperatures, 1600 °C, in the combustion chamber between the Nitrogen in the compressed hot air and the oxygen. The main sources of Nitrogen Oxides are road transports accounting for 34% [51]. This pollutant in the engine can be effectively controlled using Engine Gas Recirculation (EGR) [52]. *Particulate Matter (PM)* consists of the particles generated from the combustion such as soot, ash, volatile components, sulfates and Nitrates. Their size is 15-40nm and 90% of PM is smaller than 1µm; in Diesel engines, these emissions are six to ten times higher than in gasoline engines. They classify in PM₁₀, PM_{2.5} and PM₁ depending on the particle diameter. *Carbon Dioxide (CO₂)* represents the largest rate of GHG and it has increased exponentially in the last century [53]; in 2016 Transport accounted for 7.87 billion tons of CO₂ emitted, increasing more than 50% in the previous 20 years [6]. Particularly, lorries, buses and coaches are responsible to about a quarter of CO₂ emissions from road transport in EU and for 6% of total EU emissions [54].

Despite the improvements in fuel consumption efficiency, these emissions are still rising because of increasing road freight traffic. The Regulation (EU) 2019/1242 for emission standards for heavy-duty vehicles [55] sets the target for reducing the average emissions from new lorries for 2025 and 2030 going from 15% to 30% reduction from 2030 onwards. In the powertrain domain, several technologies have been researched and developed to save energy and lower fuel consumption. In the following section, a focus study on rightsizing is carried out as the main solution.

2.2.2 Rightsizing: superchargers vs turbochargers

In the introduction chapter, Rightsizing, as part of engine optimization strategies, was identified as the winning approach for heavy-duty vehicles. In this section, a focus is carried out on its sub-category regarding Forced Induction (FI) devices: turbochargers and superchargers.

The concept of engine supercharging involves the introduction of forced air into the engine cylinders at a density greater than ambient [56]. It is adopted because the increase of air mass and fuel inducted per cycle improves torque and power output [57]. The higher pressure achieved must be taken in account when designing the engine, as it needs to withstand higher stresses. The supercharging effect is outlined analyzing a naturally aspirated (NA) dual-combustion cycle. This is compared to a supercharged/turbocharged engine cycle as per Figure 2.1 and the main advantages are that:

- The supercharged cycle starts at a higher pressure.
- Extra fuel can be burned during the combustion process.
- Maximum pressure is increased.

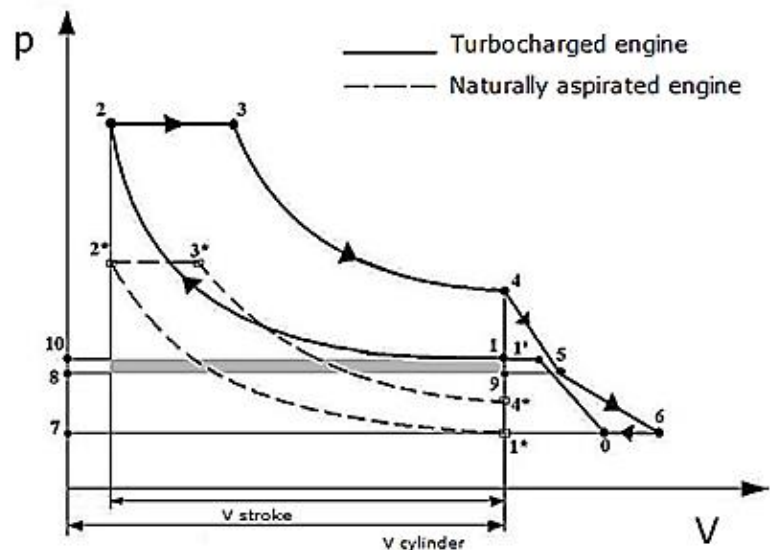


Figure 2.1 Comparison Naturally aspirated Diesel cycle with turbocharged cycle in a pressure-volume diagram (modified from [58])

The supercharging strategy can be implemented through the utilization of a supercharger or a turbocharger. A Supercharger, to produce boost, is

mechanically driven by a belt/chain connected to the engine crankshaft increasing the volumetric efficiency. The response time is improved, but the power required for its operation is a parasitic loss for the engine, the reason why it is not widely used in recent technologies [59]. Turbochargers, instead, recover the otherwise wasted thermal energy from the engine's exhaust gases to compress air at the engine's intake. A gas turbine uses the energy to rotate a compressor mounted on the same shaft, transferring the kinetic rotational energy to air pressure energy. The main drawback of this device is the energy exchange nature between the engine and the turbocharger. The engine exhaust gases needed to spool the turbine at the corresponding speed for the desired boost pressure are fixed for each engine load. This causes a mismatch between the produced and the required gas at different loads, translating in a poor transient response and speed performances. Nonetheless, because of their evident advantages, turbochargers have been a reason of interests from car manufacturer. Ford Motor Company invested in the development of a gasoline turbocharged direct injection engine technology with the Department of Energy for developing the EcoBoost system [60]. This new system can achieve 25% of total fuel economy improvement satisfying the standards imposed by the latest emissions limits. Turbochargers are more widely used compared to superchargers because of their better performance during a typical FTP-75 cycle; the latter is an Environmental Protection Agency EPA Federal Test Procedure used to assess the tailpipe emissions and the fuel economy of passenger cars. In Figure 2.2 a basic schematic of turbocharger vs supercharger configuration is shown.

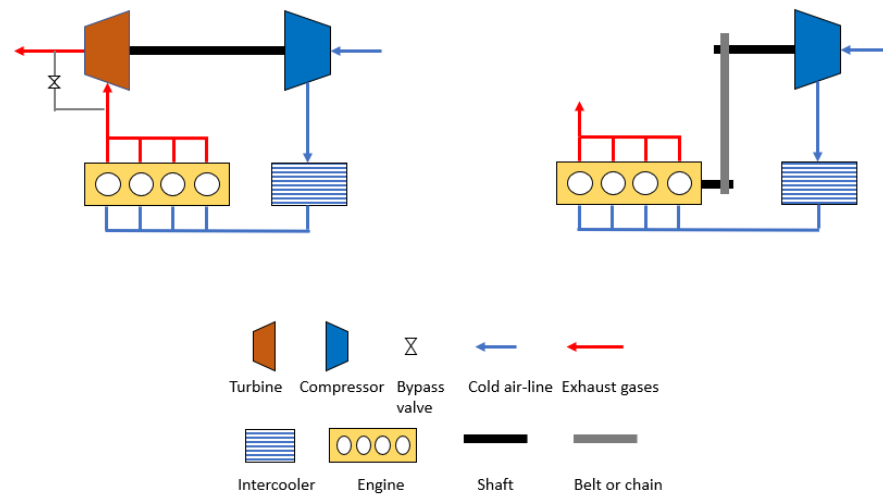


Figure 2.2 Turbocharger (left) vs mechanical supercharger strategy (right)

Summarizing, both superchargers and turbochargers can be used to increase the power and fuel economy of the ICE, with different pro and cons [61]. The turbocharger leverages the free energy available from the exhaust gases so no external output is needed for driving the compressor, increasing the organic efficiency of the system. It provides a significant increase in horsepower and a better fuel economy with a lower cost compared to the mechanical supercharger. Their disadvantages however include the turbo-lag (especially for larger engines) and a lower range of rpm operation; the turbo-lag is mostly experienced at low rpm as the boost effect is lower than at high rpm [62]. On the other hand, superchargers can provide an almost instantly boost, overcoming the issue of the turbo-lag typical of a turbocharger, especially at low rpm. However, their biggest disadvantage is lower reliability and overall efficiency as they represent a parasitic loss for the engine.

A good compromise to implement the advantage of the free energy used as input to the turbocharger, and the greater boost provided from the supercharger, is to use a turbocharged assisted by an additional source, which can be either hydraulic or electric. This will be discussed in detail in section 2.3.4.

2.3 Turbocharger technology

This section describes in detail the turbocharger fundamentals and the state of the art of the technology. It highlights the advantage of this device related to ICE

and it frames the turbocharger prototype subject of the research in the solutions available to overcome its main disadvantage represented by the turbo-lag.

2.3.1 Turbocharger overview

The reason for the benefit in using a turbocharger can be found in the fundamental science of the internal combustion engine transmission system. Equation 2.1 for Brake Power P_b shows that this, is proportional to the mass of air inducted per cycle [57] thus on the inlet air density ρ_{air} . The performance of an engine can be then improved by increasing the density ρ_{air} of the air at the inlet of the combustion chamber.

$$P_b = \frac{\eta_f \eta_v N_e V_d Q_{HV} \rho_{air} r_{(F/A)}}{n_r} \quad 2.1$$

Where η_f is the fuel conversion efficiency, η_v is the volumetric efficiency, N_e is the engine rotational speed, Q_{HV} is the fuel heating value, ρ_{air} is the air density, $r_{(F/A)}$ is the fuel-air ratio and n_r is the number of crank revolutions for each power stroke ($n_r = 2$ for a 4-stroke cycle).

A turbocharger is one of the forced-air induction devices used to increase engine performances; it is powered by the thermal energy of the engine's exhaust gases that can reach temperatures up to 1200 degrees Celsius for Spark Ignition (SI) engines and up to 850 degrees Celsius for Compression Ignition (CI) engines. The main components and configuration of a turbocharger are shown in Figure 2.3. The components part of the rotating shaft are:

- *The gas turbine:* made in Inconel to stand the high temperature from exhaust gases, it converts heat energy into rotational energy that is transmitted to the compressor mounted on the same shaft.
- *The compressor:* made of aluminum, it is connected to the intake of the combustion chamber, it increases the air pressure caught during the induction phase increasing its density; more oxidization is allowed thus more fuel to be burned on each stroke.

- *The bearings:* ball bearing types in the newest turbochargers ensure the coupling of the rotating system with the turbine casings.

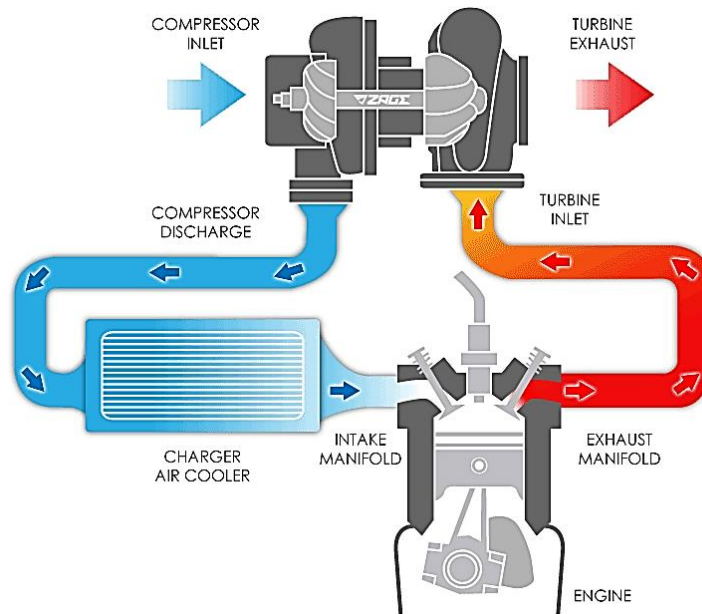


Figure 2.3 Main components of a traditional single-stage turbocharger and their basic configuration on an engine [63]

As per Figure 2.3, the turbine wheel is driven by the engine's exhaust gases and it puts in rotation the compressor wheel. The rotating compressor wheel captures the air at ambient pressure and temperature which, after leaving the compressor at a higher density, goes through an intercooler; it is then cooled down to further increase the density of the gas before entering the engine combustion chamber. A turbocharger must be designed with extreme precision because of the pressure difference between the turbine and the compressor. An essential design parameter is the ratio A/R between the turbine area A in the correspondent point in which the gas reduction area meets the casing tongue and R , the radius from the centreline of rotating parts to the centroid of A (Figure 2.4). A low ratio indicates a smaller passage and a high ratio a greater passage. A small A/R means a great acceleration of exhaust gases, optimizing boost response at low speed. This means, however, that the tangential inlet angle is quite small and at higher engine speeds involves reducing air-flow and increasing back-pressure affecting engine breathing and maximum power output. A high A/R is more desired for

higher speeds, but it limits boost performances and pressure at lower speeds. A standard turbocharger has a fixed A/R which is selected according to the intended use of the turbocharger and it is perfectly matched with the engine. In terms of speed, turbochargers can rotate up to 300,000 rpm and, however the inlet air needs to be properly filtered to avoid particles destroying the blades [64].

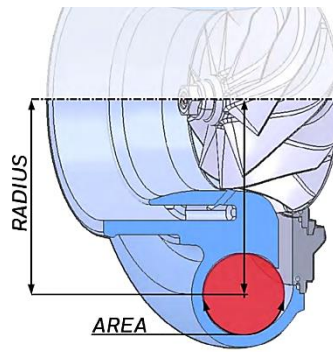


Figure 2.4 Parameter for the ratio A/R of a turbocharger turbine (modified from [65])

The content of pollutants in emissions from IC engines has already been controlled through different strategies: better quality of air supply and air mixing, precise ignition timing, cylinder de-activation, alternative fuels [66][67]. The turbocharger, however, is the only device that addresses effectively the after-use of waste heat as an emission control strategy. Turbocharged Diesel engine, if properly controlled, produce fewer emissions compared with naturally aspired and supercharged diesel engines [67]. They operate more efficiently producing proportionately less CO_2 [68] and are nowadays considered an effective strategy to reduce fuel consumption [69]. The combined effect of legislation regarding pollutant emissions (EU Directives 443/2009 EC for passenger cars and 510/2011/EC for light-duty vans), is driven by economic reasons and energy saving trend, led to an increasing use of turbocharger on CI engine [68][56]. Turbochargers are now normally fitted to CI because of their intrinsic energy recovery and because the lower cost of a CI compared to a gasoline engine, allows absorbing the additional costs of the turbocharger. Summarizing, the compressor action improves the performance of the engine because of the increase of the maximum deliverable torque resulting in lowers emissions and engine downsizing.

The benefits in using turbochargers are so well proven in CI engine, that a non-turbocharged diesel is considered obsolete [66]; in gasoline engines a brief comparison between turbocharged SI and naturally aspirated SI engines highlights several advantages: 40-45% downsizing of the base engine [70], 15-20% improvement in fuel economy [57] with 40% downsized turbocharged gasoline a 16-18% better fuel economy [71] and greenhouse emissions reduction of 25-30% [66]. Furthermore, the higher initial cost of a turbocharged engine can be offset with cost savings in engine downsizing [66].

Although the clear advantages of turbocharged engines, a significant disadvantage in the functioning of this device is the delayed time response called turbo-lag; this is widely discussed in section 2.3.3.

2.3.2 Turbocharger matching

As introduced in the previous section, the performance of a turbomachine is highly dependent on the parameter A/R thus on the gas angles at the impeller inlet. The blade angles are set to match these angles and the mass flow rate is corrected for a specific rotor speed.

A standard fixed-geometry turbocharger needs then to be matched with the engine transient operation related to the different loads and having a target horsepower. The overall aim of the turbocharger matching is to fit a turbocharger with the most suitable characteristics to an engine to obtain the highest performance. This is possible just at a particular design point in the operating range of the engine. When an engine is required to run for most of its life at a constant speed and full load, the turbocharger is chosen to have its high-efficiency operating area coincident with the pressure ratio and mass flow requirements of the engine at that condition. The basic size of the turbocharger is determined in equation 2.2 by the air required by the engine. The mass flow rate $\dot{m}_t$ is a function of the swept volume V_{sw} , rotational speed N_t , boost pressure, density of air ρ_{air} at the manifold and volumetric efficiency η_v [48]. The density ρ_{air} in equation 2.3 depends on the boost pressure and temperature and the latter is calculated from equation 2.4 where the subscript a identifies ambient conditions, η_c is the compressor efficiency and $k=c_p/c_v$

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$$\dot{m}_t = \frac{N_t}{2} V_{sw} \rho_{air} \eta_v \quad 2.2$$

$$\rho_{air} = \frac{p}{RT} \quad 2.3$$

$$T = T_a + \frac{T_a}{\eta_c} \left[\left(\frac{p}{P_a} \right)^{(k-1)/k} - 1 \right] \quad 2.4$$

An essential tool for turbo-matching is the compressor map which identifies areas in which the boost is more efficient (Figure 2.5). The Efficiency Islands are concentric regions that represent the compressor efficiency at any point on the map; the smallest island near the center of the map is the highest or peak efficiency island. As the rings move out from there, the efficiency drops by the indicated amount until the surge and choke limits are reached. The Mass Flow Rate is the mass of air flowing through a compressor over a period of time and is expressed usually as lb/min. As an example an engine with a target peak horsepower of 400 Hp will require 36-40 lb/min of airflow to achieve that target [65]. Turbo Speed Lines are constant turbo speeds measured in rpm. As turbo speed increases, the pressure ratio and mass flow increase. Turbo speed lines are very close together at the far right edge of the map indicating a potential turbo over-speed condition. The Choke Line is the right-hand boundary of the compressor map and is defined at the point where the efficiency drops below 58%. In addition to the rapid drop of compressor efficiency past this point, turbo speed also approaches or exceeds the recommended limit. The surge line is the left-hand boundary of the compressor map and represents a region of flow instability. This region is characterised by mild flutter to wildly fluctuating boost from the compressor. Continued operation within this region can lead to premature turbo failure due to heavy thrust loading. The surge will decay once the turbo speed finally slows enough to reduce the boost and move the operating point back into the stable region. This situation is commonly addressed by using a Blow-Off Valve (BOV) or bypass valve. A BOV functions to vent intake pressure to the atmosphere so that the mass flow ramps down smoothly, keeping the compressor out of surge. In the case of a recirculating bypass valve, the

airflow is recirculated back to the compressor inlet. The pressure ratio is the ratio between the absolute outlet pressure at the compressor and its absolute inlet pressure.

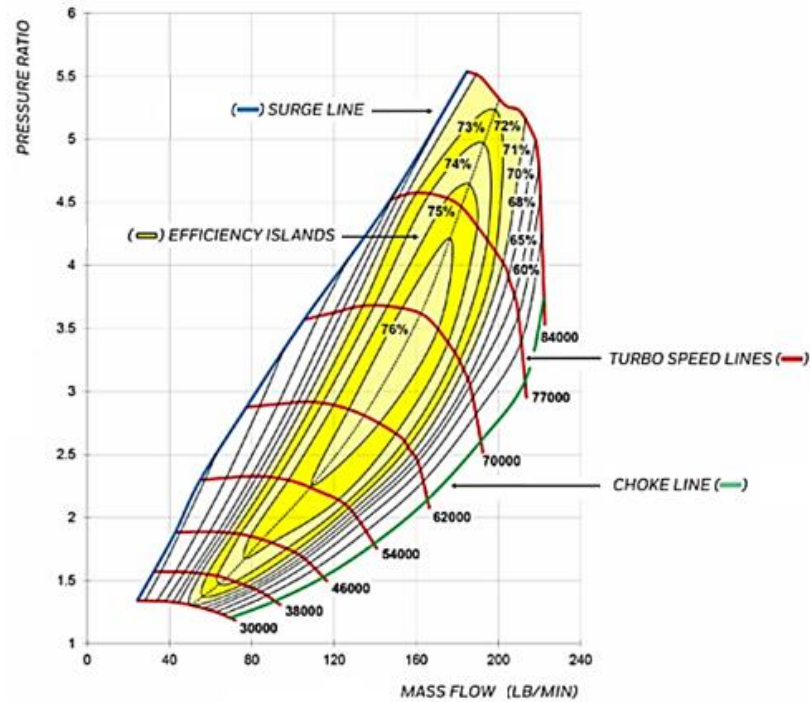


Figure 2.5 Typical compressor map of a turbocharger (modified from [65])

Once the required parameters for the compressor are established, a basic frame size of the turbocharger can be selected. The final choice is made considering the complete operating lines of the engine over its whole speed and load range, superimposed on the compressor characteristics. The compressor diffuser is chosen to allow a sufficient margin from the surge line while ensuring that the operating lines pass through the high-efficiency area. When the basic frame is selected, the effective turbine area must be matched by altering its nozzle ring or volute.

2.3.3 Turbo-lag and solutions

Despite the clear advantages that a turbocharger provides to the engine, there are some issues regarding thermal durability (materials need to withstand 1050°C) [66] and cold start emissions. Without doubt, the most concern stemming from

this device is related to the transient behavior that occurs as a result of the rapid actuation of the accelerator. This inconvenience is called **Turbo-lag**; this time is required by the engine to produce enough exhaust gas pressure to spool up the device to the minimum rotational speed to exceed atmospheric pressure and pump the compressed air into the engine. This problem of poor performance under changing load or speed conditions can be attributed to the nature of the energy transfer path between the engine and the turbocharger. Under accelerating conditions only part of the energy at the turbine is delivered to the compressor for a boost effect, the remaining is required to overcome the inertia of the turbocharger and accelerate its rotating part [48][72]. The boost pressure developed when the engine accelerates is likely to be below that developed when operating steadily at the same instantaneous engine speed and fuel rack position. The results are combustion deterioration with smoke emission and limited instantaneous torque developed by the engine due to the slow response time. In a high-speed diesel engine applying full load in 1s, the time response of the turbocharger to reach stationary regime can be up to 4s [48]. The turbo lag issue strongly unfits that applications in which rapid changes in the engine are required (i.e. city travel). The solutions proposed to overcome the turbo lag focus on modification of its geometry to compensate for the functioning delay. The enablers are distinguished in turbine and compressor technologies [73], [74]; numerous solutions are proposed and a summary of the main available systems is given below.

1. Wastegate valve: a turbo can be tuned with a smaller exhaust housing that allows the turbine to find optimal performance points even at medium loads. However, to avoid high back pressure situation at the engine discharge, a wastegate valve is used; this simply limits the pressure, wasting part of exhaust gases. A downsizing of the turbo is designed but the same amount of flow rate in the discharge system is ensured. Despite the clear benefit, the downsides are the lower boost pressure of the engine at high loads and the lower efficiency of the system because of the wasting of the exhaust gases thermal content.

2 Literature Review

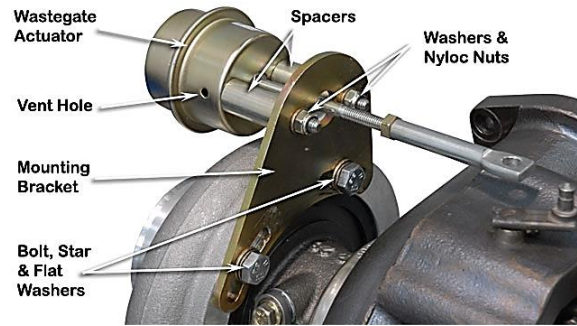


Figure 2.6 Wastegate valve positioning in standard turbochargers [75]

2. Multi-stage turbochargers: a traditional system for overcoming the turbo lag involves the installation of twin turbines/compressors in parallel. This provides a lower grade of delay compared to a greater turbine, supplying however the same boost pressure. An advanced configuration consists of multi-stage turbines, hence different dimensions of turbines are coupled. A smaller turbine at the beginning of the stage works in optimal conditions responding in acceptable periods; when the load increases, thus the exhaust gases, part of the exhaust gases is bypassed into a bigger turbine, optimised for higher loads. The smaller turbine then acts absorbing the first stage of transient loads while the higher turbine can work in its optimal performance point at greater loads.
3. Variable-geometry turbocharger (VGT): a discharge impeller with movable vanes can be used to overcome the turbo-lag issues; this involves the variation of cross-sectional area A of the turbine which reflect on the variation of the aspect ratio A/R introduced in section 2.3.1. The variation of the geometry adds a degree of freedom to the turbocharger, giving optimal control in different working conditions. The technical advantages are related to the control firstly of the turbine inlet area, which is adapted to the optimal gas flow rate, and secondly of the exhaust gases' impact angle on the turbine impeller, which is adapted to the velocity of the gases. The increase of the turbocharger speed is achieved by reducing the inlet area of the turbine increasing then the exhaust manifold pressure. Coupling the variable geometry with an electronic unit control ensures the same inertia at lower flow rates typical

of a small turbine maintaining the same maximum flow rate of a greater turbine. The VGT, although being the main technology used in Diesel engines [74], is, however, a delicate system because of the movable part at the stator; it better fits indeed in CI engines with a lower exhaust temperature compared to gasoline engines. The aspect ratio of the turbine is usually varied between 0.3 and 1; at a lower speed, the adjustable guides are closed and small amounts of exhaust gases are sped up in the turbine blades, while at a higher speed the guides are open, allowing exhaust gases to flow directly into the turbine (Figure 2.7). Studies have shown furthermore that the Active Control strategy can be combined with VGT improving power gains from the engine up to 7.76% [76].

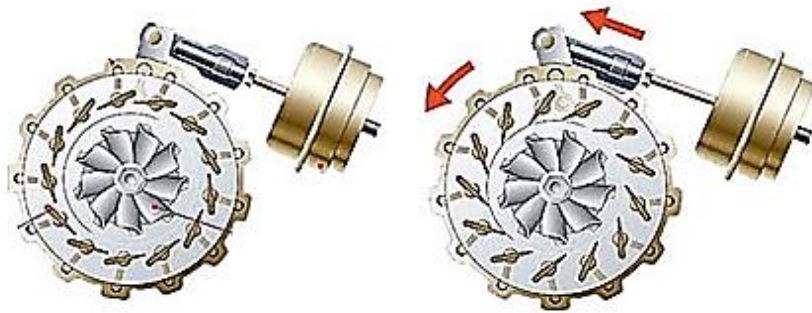


Figure 2.7 variable geometry impeller in a VGT turbocharger;
deflector orientation at low rpm (left) and at high rpm (right) [77]

4. Assisted turbochargers: Another useful expedient to overcome the turbo-lag is the coupling with an external assisted device. This can be driven by electrical input as an energy source (EAT: *electrically assisted turbocharger*) or by hydraulic energy (HAT: *hydraulically assisted turbocharger*). This solution is relatively new but it was proven to be efficient in providing the initial boost to spool up the rotational speed with the shortest delay. It is specifically used at lower engine speeds and loads; different configurations and they are available in detail and discussed in section 2.3.4.

2.3.4 Assisted turbochargers

Assisted turbochargers are a possible solution to overcome the lag using an electric/hydraulic assist. Powertrains are supposed to become slowly but increasingly integrated with electrical architectures to satisfy the fuel requirement for the EHV incoming of 2050 [18] and Electrically-Assisted Turbochargers could fit perfectly in this scenario [78]. The EAT has an electric motor mounted on the same shaft of compressor and turbine that increases the impeller speed sending compressed air to the inlet of the engine. They can improve the transient response of a traditional turbocharger and generally the Brake specific fuel consumption through two possible configurations.

1. “In-built motor”: a conventional turbocharger with a high-speed motor generator mounted directly on the shaft. The electric motor assists the turbo at low engine speed when the exhaust gas energy is insufficient, improving the transient response delay. The excess gas energy during high-engine speed zone can also be recovered as electric power and stored in the battery.
2. “Turbocharger with additional electric driven compressor (TEDC)”: it involves an external electrically operated compressor as an aid device for the turbocharger.

A more complete classification of different electrically assisted turbocharger topologies is available in [79] and a brief example is given in Figure 2.8.

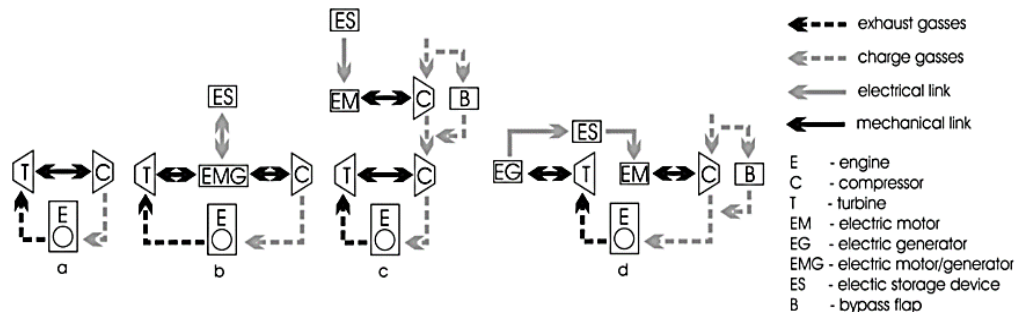


Figure 2.8 Schematic layout of electrically assisted turbocharging topologies (modified from [79]); a) single stage turbocharger (baseline), b) EAT, c) TEDC, d) Electrically split turbocharger (EST)

2 Literature Review

EAT was shown to have good potential and significant literature is available on the topic. Table 2.1 summarised the results found in literature. The results show that an electrically assisted turbocharged can be a valid alternative compared to the supercharger as turbo lag solution, however, the power required still represents a parasitic loss for the engine, which results in increased fuel consumption[59].

<i>EAT characteristics and application</i>	<i>Benefits</i>	<i>Reference</i>
Numerical simulation validated: 12V battery, 2L diesel engine with LP-EGR	-2.2% Fuel consumption +12.5% vehicle acceleration	[80]
Numerical 1D simulation of a Renault megane :1L turbo-charged engine, E-booster 1.5bar, 80k rpm, 1.6 kW	-67% turbo response time	[81]
Tested gasoline engine 1.5L 12V battery, 3kW, 90k rpm	-43% turbo response time	[82]
Turbo motor 1-3kW assistance	-70% turbo response time with 2kW assistance 50-100% engine torque improvement 8-15% engine thermal efficiency improvement	[83]
Simulation on Hybrid Electric vehicle Mercedes-Benz E CClass	-4.2% vehicle acceleration time 0-100km/h	[44]
Simulation with 4-cylinder EGR HSDI Diesel Engine experimental validated	at 1000 rpm speed engine with TEDC: +38% boost pressure, +48% BMEP	[79]
0D simulation program validated with experimental input data	-56% turbo response time	[84]
1D engine simulation fluid dynamic code	1-6% fuel consumption improvement	[85]
GT-power analysis on EAT 1-5 kW coupled with a 2L SI engine	70-90% improvement in response time	[59]

Table 2.1 Summary of results available in literature for EAT investigations

An alternative to the electrical-assisted turbocharger is represented by Hydraulic-Assisted Turbochargers (HAT). The idea of a hydraulic assisted turbocharger lies originally in a patent assigned in 1961 to Birmann [86] that was further developed by Byrne with Garrett corporation in 1981 [87]; in 1992 Kobayashi included a controlling system [88]. Kapich in 2012 fully developed

the concept of a hydraulic-pump motor in mechanical communication with the engine shaft and the turbocharger system [89]; the controlling system was further modified by Sun et al. [90] in a different patent. In a HAT a hydraulic turbine selectively driven by a high-pressure hydraulic fluid is mounted on the turbocharger shaft to drive it supplementally during engine operating conditions when additional airflow to the engine is required. Kapich showed in a preliminary evaluation in a SAE paper [91] that potential for improvement in fuel efficiency of heavy-duty turbo-diesel engines exists, however, the model proposed was in the early stage of development, thus more reliable results are required. Zeng et al. [20] in a more recent study in 2018 investigated a RHAT (Regenerative Hydraulic assisted turbocharger) custom-designed and coupled to a light-duty diesel engine. Vehicle and engine simulations showed that the RHAT system can significantly improve engine transient response with a potential 4% fuel economy improvement on the FTP 75 drive cycle. Timoney performed experimental tests [22] on a crankshaft-driven hydraulic system to add auxiliary power to the compressor of a standard turbocharger, experiencing a reduction in both intensity and duration of critical exhaust smoke, together with a reduction of engine speed at which peak torque occurs of 10-15%. Further research is required on this device to compare their performances with EAT.

The literature on REAT (Regenerative electric assisted turbocharger) showed the potential of these devices to overcome the turbo-lag; the competitor RHAT (Regenerative hydraulic assisted turbocharger) system has the main advantage which lies in power density over energy density. For medium and heavy-duty automotive diesel applications REAT brings indeed some concerns [20]. Assisted motors in REAT have usually a power output in the range of 1.5-10kW [83][92][93], whereas the required power for assisting the acceleration of larger turbochargers is more than 10kW depending on the size. This boost assist represents a challenge for the 12V power supply system in vehicles. Additional challenges in REAT are furthermore represented by the cooling of the electrical machine and the fuel penalty from maintaining the battery state of charge over a working cycle [92]. To recover the exhaust energy during braking, the power capacity needs to be in the range of 15-40 kW [20] as the electric turbo operates at high speeds. The packaging of the REAT for this power would be challenging

in terms of increased rotational inertia and mechanical stress, and the 2kW electric motor supported by 12V would not effectively recover exhaust energy when it exceeds. For these reasons, RHAT is an attractive alternative for heavy-duty vehicles compared to the REAT because of lower inertia, lower cost, smaller packaging space and better durability.

2.4 KERS: Kinetic Energy Recovery Systems

In this section, the kinetic energy recovery concept in road transportation is introduced, analyzing the available technologies and focusing on the energy recovery process. Hybrid vehicles, which combine two power generating sources and store recovery energy through regenerative braking to power the tractive system, are explained. A focus on HHV is carried out to analyze the studies available on this technology.

2.4.1 KERS overview

The European Union in a joint research with the Institute for Prospective Technological Studies estimated that up to 46% of energy losses in a conventional vehicle can occur from braking [40]. Hybrid vehicles leverage on this type of tractive loss as their energy recovery sets the foundation of the efficiency and technological improvement over conventional vehicles. The advantage of KERS systems lies indeed in the *Regenerative Braking* (RB) which recycles the kinetic energy otherwise wasted in thermal energy during the braking events in road trips. To give an order of magnitude, braking energy can represent up to 70% of useful motor energy for some urban driving conditions, 40% for suburban and 18% for motorway conditions [94]. RB systems are installed along the drive train or fitted to the drive wheels of a vehicle. The main technologies used in KERS are listed below according to the physical principle used for energy conversion and storage. They all aim to maintain consistent energy where the normal energy source is intermittent.

- Electro-Mechanical: the system collects the kinetic energy of the vehicle to spin a *Flywheel* connected to the drive shaft through a transmission, a motor/generator and a gear box. The flywheel maintains a constant

rotational speed even if the primary input is intermittent and it stores the energy in the form of rotational momentum. When the power is needed, the flywheel provides torque to the drive shaft. A vacuum pump can be used to remove air from the encasing chamber to reduce air resistance of the flywheel, thus windage losses, maximising its efficiency. Flywheels are a smart choice for RB because of their high power and energy density, no degradation related to the load variation, no periodic maintenance, short recharge time and low environmental impact [95]. A study showed that their overall fuel economy efficiency improvement over a full regenerative cycle was about 70% on a truck [96] leading to CO₂ reduction of about 30%. Despite the advantages, mechanical failure can be catastrophic because of the high magnitude of the forces involved; for this reason the flywheel is encased within a resistant casing.

- Electromagnetic: the drive shaft is connected to an electric motor/generator unit (MGU) and the energy is store in the form of electricity. Due to the maximum recharging rate of the circuit and the capacity of the battery, the braking force recovered from this type of layout is always limited. Furthermore, the multiple energy conversions (kinetic-electrical-chemical-electrical-kinetic) reduce the overall efficiency over a full regenerative cycle to only 36% [96]. Another reason for inefficiency is the production of heat while the batteries are charged. Cooling systems must be carefully designed to avoid lowering storage capacity efficiency and secondary issues related to the high temperature. Finally, the weight of the batteries prevents the installation of these systems into heavy-duty vehicles. Despite the disadvantages, it is more efficient in the long-time storage of recovered energy compared to a flywheel system [96] .
- Hydraulic: in this system RB produces electricity which is combined with a hydraulic reversible pump. The pump sends hydraulic fluid from a low-pressure reservoir into a high-pressure accumulator. The Nitrogen gas in the accumulators allows to benefit of the potential pressure energy of the compression for pressurizing an incompressible fluid. The compressed fluid system has the highest efficiency as energy storage as it dissipates a very low amount of energy over time. This system

configuration was also proposed for heavy-duty trucks and it was predicted to store 80% of the momentum lost by a vehicle during deceleration [23]. The hydraulic regenerative braking, that was shown numerically and experimentally to have energy recovery efficiency up to 50% [97], can be efficiently adapted to different vehicles configurations; in [98] a combination of hydraulic storage in an electric vehicle showed that the fuel economy was enhanced by the RBs for than 25% under a ECE driving cycle. Another study showed a fuel consumption saving of 21.7% over and idealised stop-start cycle [99]. This system represents a relatively new technology and for this reason it is still not widely applied in the automotive industry.

- Compressed air storage: compressed air energy storage is used already for some industrial application [100], however this idea was expanded to be applied in cars [101] and it is claimed to improve fuel economy of 35% using compressed air to storage energy from regenerative braking; however, so far, the not proven reliability of the system did not allow providers to bet on this technology.

2.4.2 Fuel economy evaluation

In road transportation, KERS are commonly applied in Hybrid and full electric vehicles where the electric motor turns into generator mode when a braking event occurs. All KERS act using RBs regenerative braking systems; to evaluate the efficiency of the process, three aspects are mainly considered [102]: system design, Blended Brake Control and energy efficiency evaluation.

Research and development on system design and brake control systems have been the focus of car manufacturers and researchers worldwide. Toyota developed an electro-hydraulic “brake-by-wire” system and implemented in a EHV [103]. Nissan Leaf electric car implemented an electrically-driven system that, through a system of sensors and solenoid valves, ensures more reliability using direct mechanical control of the service brakes [104]. Energy regeneration system simulation based on PEM (Polymer Electrolyte Membrane) fuel-cell, super-capacitors and batteries were found to have an optimised configuration ensuring lower specific energy consumption [105]. A combination of fuel-cell system and ultra-capacitors have also been investigated to satisfy both the load

demands, which is a flaw in FC, and to provide suitable energy storage [106]. The recovery energy systems through braking were also investigated for minivans giving an improvement of 15% more compared to the baseline control strategy [107]. A similar study on recovery braking applied in a fuel cell city bus [108] showed that the potential braking energy of the vehicle can be recaptured decreasing hydrogen consumption and improving the driving range significantly. In [98] three different control strategies for regeneration efficiency and brake comfort were studied for a hydraulic-braking of an electrified passenger car. In a ECE driving cycle with the regenerative braking system developed, the fuel economy was enhanced by more than 25%.

The contribution brought by regenerative braking to electric vehicle's energy efficiency improvement can be evaluated through the strategy proposed in [109]. The evaluation parameters are the contribution ratio to energy efficiency improvement and the contribution ratio to driving range extension. Experimental results on a NED driving cycle showed an energy efficiency improvement of 11.2% and a driving range improvement of 12.6%.

The ability to regenerate energy with braking is then an omnipresent feature among hybrid vehicles in road transportation; computing its potential is of great interest for improving the vehicles intrinsic efficiency. The required instantaneous tractive force F_{TR} for driving on a straight smooth level road in absence of wind is given in equation 2.5 and it is the combination of three contribute: the resistance R_{TR} , the aerodynamic drag D_{TR} and the linear+rotational inertia [110].

$$F_{TR} = R_{TR} + D_{TR} + \left[m + 4 \left(\frac{I_w}{r_w^2} \right) \right] \left(\frac{dv}{dt} \right) \quad 2.5$$

$$F_{TR} = r_{tr}mg + C_D A \frac{V^2}{2} \rho + \left[m + 4 \left(\frac{I_w}{r_w^2} \right) \right] \left(\frac{dv}{dt} \right) \quad 2.6$$

The terms are expressed in Equation 2.6 where r_{tr} is the tire rolling-resistance coefficient, g is the gravitational acceleration, C_D is the aerodynamic drag coefficient, A is the vehicle frontal area, m is the vehicle mass, ρ is the density

of ambient air, v is the linear velocity, I_w is the polar moment of inertia of a wheel assembly, r_w is the effective rolling radius of a wheel assembly. The term $\left[M + 4 \left(\frac{I_w}{r_w^2} \right) \right] \left(\frac{dV}{dt} \right)$ is related to the total inertia so it can be called M_{intot} . The term $4 \left(\frac{I_w}{r_w^2} \right)$ accounts for the fact that the four wheels must be rotationally as well as linearly accelerated. R and D are always positive, but the inertia term can be negative. In a traditional vehicle, driving phases can be categorised into three generic modes associated with its tractive force:

1. Powered driving: $F_{TR} > 0$ occurring during vehicle acceleration where $\frac{dV}{dt} > 0$, in constant-speed driving where $\frac{dV}{dt} = 0$ or even in deceleration event when $\frac{dV}{dt} < 0$, as long as it is always $R_{TR} + D_{TR} > M_{intot}$
2. Braking: $F_{TR} < 0$ occurring when $M_{intot} > R_{TR} + D_{TR}$
3. Idling: $F_{TR} = 0$ occurring when $M_{intot} = R_{TR} + D_{TR}$

An example of the power profile $P_{TR}(t)$ is plotted in Figure 2.9 according to 2.7. The positive area under the power curve (grey areas) represents the *Tractive Energy* E_{TR} in 2.7 and the areas between the negative parts of the curve and the axis (green areas) are elements of the *Braking Energy* E_{BR} in 2.8.

$$E_{TR} = \int_{F_{TR} > 0} P_{TR}(t) dt = \int_{F_{TR} > 0} F_{TR} v dt \quad 2.7$$

$$E_{BR} = - \int_{F_{TR} < 0} P_{TR}(t) dt \quad 2.8$$

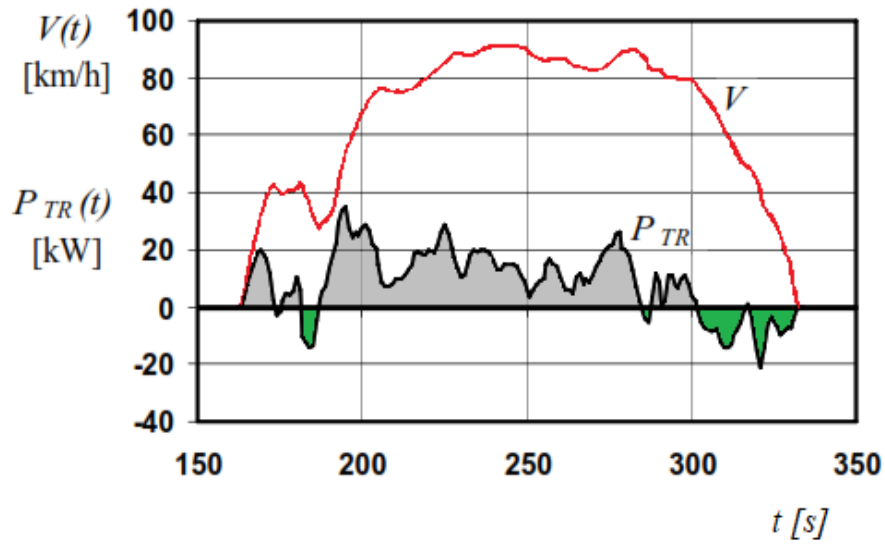


Figure 2.9 Velocity and tractive power variation of a typical start/stop driving Urban schedule (modified from [110]).

$V(t)$ is the speed of the vehicle in km/h. P_{TR} is the tractive power of the vehicle expressed in kW. The braking energy is represented by the green areas and the tractive energy is in the grey areas.

Both E_{TR} and E_{BR} can be evaluated in selected parameter ranges with linear equations with coefficients dependent only on the driving schedule [110]. During decelerations in each driving cycle, the braking force E_{BR} which is usually of frictional nature, can be converted in reusable energy to power the vehicle propulsion. The regenerative braking reduces indeed the portion of tractive energy that needs to be delivered directly from the conventional propulsion system through active fuel transformation (fossil fuel, electricity, etc.) during the powered driving. The potential improvement enabled by regenerative braking can be quantified unifying the approach in [110] with fuel consumption estimation. Fuel consumption is calculated according to the three contributes of powered driving, braking, idling) together with the regenerative training capability and powertrain performance. The combination of equations 2.9 and 2.10 of the efficiency of power conversion and transmission, leads to the mass of fuel consumed for tractive purposes $m_{f,TR}$ in equation 2.11. The last equation highlights how when wheel to wheel regeneration effectiveness η_{BR} increases, $m_{f,TR}$ is reduced. η_{dt} is the drivetrain efficiency. These are not going to be used in the study of the thesis but they are a starting point for future works using the results on the HAT prototype and the potential power required from the hydraulic source.

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$$\eta_{pc} = \frac{E_{b,TR}}{H_f m_{f,TR}} \quad 2.9$$

$$\eta_{dt} = \frac{E_{TR} - \eta_{BR} E_{BR}}{E_{b,TR}} \quad 2.10$$

$$m_{f,TR} = \frac{E_{TR} - \eta_{BR} E_{BR}}{H_{fuel} \eta_{pc} \eta_{dt}} \quad 2.11$$

A comparison can be made in terms of fuel consumption with a conventional vehicle with ICE traction. To simplify the nomenclature, the subscript “0” is attributed to the conventional vehicle, where “hv” is attributed to the hybrid vehicle, thus $m_{f,TR} = m_{f,hv}$. The conventional vehicle is considered to have tractive-fuel consumption $m_{f,0}$ which is shown in equation 2.12. The ratio of $m_{f,hv}$ to $m_{f,0}$ is a measure of the fuel consumption benefit resulting from KERS in equation 2.13

$$m_{f,0} = \frac{E_{TR,0}}{H_{fuel} (\eta_{pc} \eta_{dt})_0} \quad 2.12$$

$$\frac{m_{f,hv}}{m_{f,0}} = \underbrace{\left[\frac{(\eta_{pc})_0}{\eta_{pc}} \right]}_{\text{Power conversion efficiencies}} \underbrace{\left[\frac{(\eta_{dt})_0}{\eta_{dt}} \right]}_{\text{Drivetrain efficiencies}} \underbrace{\left[\frac{E_{TR}}{(E_{TR})_0} \right]}_{\text{Tractive efficiencies}} \underbrace{\left[1 - \eta_{BR} \left(\frac{E_{BR}}{E_{TR}} \right) \right]}_{\text{Regenerative braking contribute}} \quad 2.13$$

Equation 2.13 is split in four terms:

1. Power conversion operating efficiencies: usually it is considered $\eta_{pc} > \eta_{pc,0}$ as the energy storage device allows the power conversion system to not being dependent on the instantaneous tractive energy requirement.
2. Drivetrain operating efficiencies: this term can be greater than the unity or lower, dependently on the mechanical and electrical drivetrain efficiencies.

3. Tractive energy efficiencies: the vehicle mass M_{hv} of a hybrid vehicle is usually 1.05 the value of the mass M_0 of a conventional vehicle [111] because of the additional component required for the regenerative braking conversion system. This means the third term in the equation is >1 .
4. Regenerative braking influence: the ratio between the energies required for braking and for traction is less than the unity, this means that the overall term will be less than the unity.

The fourth multiplicative term in equation 2.13 is the one directly influenced by the regenerative braking. The other terms are influenced indirectly and act on the overall powertrain system. The order of magnitude of the potential fuel economy for regenerative braking applied in hybrid solution is shown in Figure 2.10 [112]. The baseline is a gasoline engine and this is compared with the same engine electrically hybridised, with the addition of fuel direct injection strategy to the latter system and to a diesel electrically hybridised system; electrically hybridised hydrogen fuel and compressed natural gas system are considered as well. The electrically hybridised hydrogen fuel engine seems to give the best performances with a fuel saving of 39%.

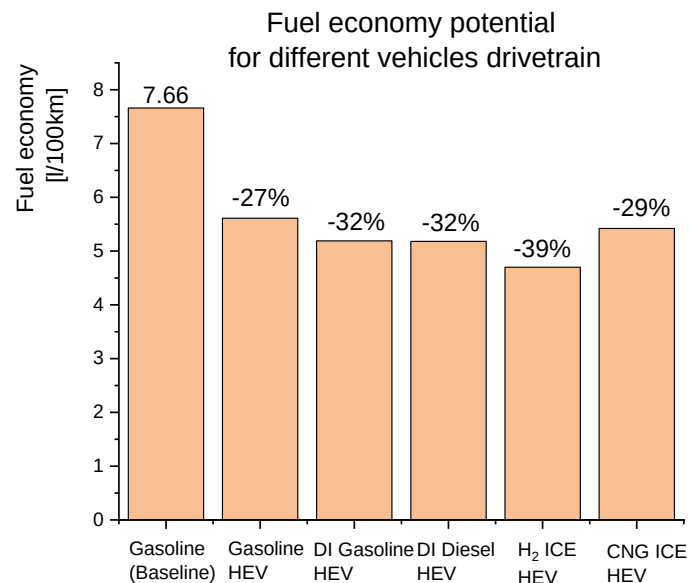


Figure 2.10 Potential fuel economy of some powertrain options in [112]

DI-Direct Injection ICE

EHV-Parallel Hybrid Electric Vehicle with Charge Sustaining Control Strategy

CNG-Compressed Natural Gas

2.4.3 Standard Driving cycles

The regeneration potential of a KERS depends strongly on the driving cycle of the vehicle and the recovery power capacity of the drivetrain.

Concerning the driving cycle, the papers available in literature estimates fuel economy and vehicles efficiency improvement based usually on NEDC and WLTP driving cycles more recently. The NEDC applied for all passenger vehicles and light commercial vehicles, and it was introduced by EU in 1992 to provide comparable values for fuel consumption. The cycle was criticised for its unrealistic reproduction of a driving cycle; it must be performed indeed on a cold vehicle (25 C degrees) on a flat road in the absence of wind. These situations are unachievable in reality and they are usually performed on a roller test bench equipped with an electrical machine to emulate resistance due to aerodynamic drag and vehicle mass. The NEDC is composed of two parts: 4 times of Urban Driving Cycle repeated and Extra-Urban Driving cycle plotted in the second part of the test. The cycle was characterised by an average speed of 34 km/h with smooth acceleration, few and prolonged stops and a top speed of 120 km/h. The structure of the cycle has become outdated as nowadays distances and road variety are consistently different from 1990. WLTC (Worldwide Harmonised Light Vehicles Test cycle) is the updated standard that from 2017 provides more realistic specifications as it utilises a profile deemed to be more similar to daily usage compared to the NEDC [113]. WLTC sets standards fuel economy and emission tests not only for Europe but also for India, South Korea and Japan, harmonizing also studies and investigation on both traditional vehicles and hybrid/full electric vehicles. WLTC sets the average speed at 46.5 km/h, top speed at 131.3 km/h and has a velocity profile more dynamic which consists in more rapid accelerations with short brakes. The distance covered during the test cycle is 23.25 km, whereas NEDC distance was 11 km. Overall WLTP compared to NEDC was improved including higher speeds, a wider range of driving conditions, a longer distance, a higher drive power, steeper accelerations events. Together with the fuel economy test, RDE (Real Drive Emissions) tests are carried out to verify that legislative caps for pollutants are not exceeded under real use.

The Recovery Power Capacity of a KERS system is also of essential consideration when the regeneration potential is analysed together with the fuel

economy improvement. Optimization analysis suggests that storage capacity is not linearly related to energy saving potential; Saxena et al. [114] showed that a higher efficiency engine operation and more engine shutdown is enabled by a larger battery but only up to 10 kW power capacity. Energy saving from regenerative savings occurs with a larger power capacity up to 20 kW; beyond these values there are diminishing return to increasing the power capacity.

Summarizing, the KERS process has then an intrinsic efficiency, however there are challenges on kinetic energy recovery improvement related to power/torque handling ability of the propulsion system, employed energy conversion techniques and energy storage capacity. In the following sections the main hybrid technologies available for vehicles according to the type of kinetic energy recovery method and the storage mean for road transportation will be analysed: Hybrid Electric vehicles (EHV) and Hydraulic Hybrid vehicles (HHV).

2.4.4 Introduction to Vehicles Hybridization

A hybrid vehicles uses two different type of power that collaborate for the traction. The main means of storing energy can be electric (battery storage) or hydraulic (pressurised fluids). In EHV two energy sources collaborate for the traction: a TPS and an electric motor/generator. This is a compromise between ICE and BEV; the first one causes high emissions, whereas the BEV faces the challenge of expensive electricity storage and time-consuming battery charging [115]. In EHV the electric energy is recovered through RBs specifically using an electric generator and it is stored in battery, supercapacitor or alternative fuels.

EHV main configurations are series or parallel [116]. In the *Series* configuration, the TPS engine has no connection to the wheels and it just charges the batteries, which power the drivetrain electric motor. The advantages of this system are that the ICE works at fixed points meaning the efficiency is the maximum because the ICE is not designed on the maximum power but on the average one [117]. There is the possibility of zero emissions target during the stage of use [118]. Despite the advantages, three elements of propulsion are used (ICE, electric generator, electric motor) leading to a triple energy transformation (chemical-mechanical-electric-mechanical). The energy efficiency of this conversion is

then lower than the ICE process. The large dimension of the batteries represents another point of disadvantage in these systems. The *Parallel* configuration has the electric motor coupled in the same shaft of the traditional ICE transmission system and it cooperates to the traction of the wheels acting as well as a generator during the RB. The system allows to downsize the engine because of the collaboration of the two engines (electric and TPS), just one electric/motor component is used compared to the series system and the energy conversion path is shorter. In this system, it can be prioritised TPS transmission (mild hybrid, smaller batteries) or the electric motor (full hybrid, plug-in charging). The disadvantages include the complex transmission system, lower efficiency of the TPS because of the different working points and less electrical autonomy compared to the series system. Summarizing, the parallel hybrid acts as a traditional TPS (high autonomy and good performance) but with better efficiency and the possibility to work at zero emissions (full electric mode) [117]. Disadvantages that are not overcome by the EHV technology are the dependence on fossil fuel and higher cost compared to ICE. Furthermore, plug-in hybrids require long charging operations and disposing of batteries is still an important issue, indeed lithium-ion batteries are considered toxic. The major issues nowadays are represented by battery sizing and the management of multiple energy sources.

2.4.5 Focus on Hydraulic Hybrid Vehicles

HHV adopt the same logic as EHV but instead of using lithium-ion batteries, they use lightweight components as storage mean; these exploit the potential energy of a pressurised fluid to give additional power to the engine [119]. In Figure 2.11 components of HHV are shown. The three main components to power the vehicle and help the engine in operation are:

1. *Low-pressure reservoir*: the fluid is stored at low pressure;
2. *Hydraulic drive (pump/motor)*: acting as a pump it moves the fluid from the low-pressure reservoir to a high-pressure accumulator, acting as a motor it uses the pressurised fluid to move the wheels;

3. *High-pressure accumulator*: this component stores the high-pressurised fluid. It's usually a bladder type with nitrogen on the gas-side and oil on the fluid-side rated for 5000 psi (344 bar) [120];

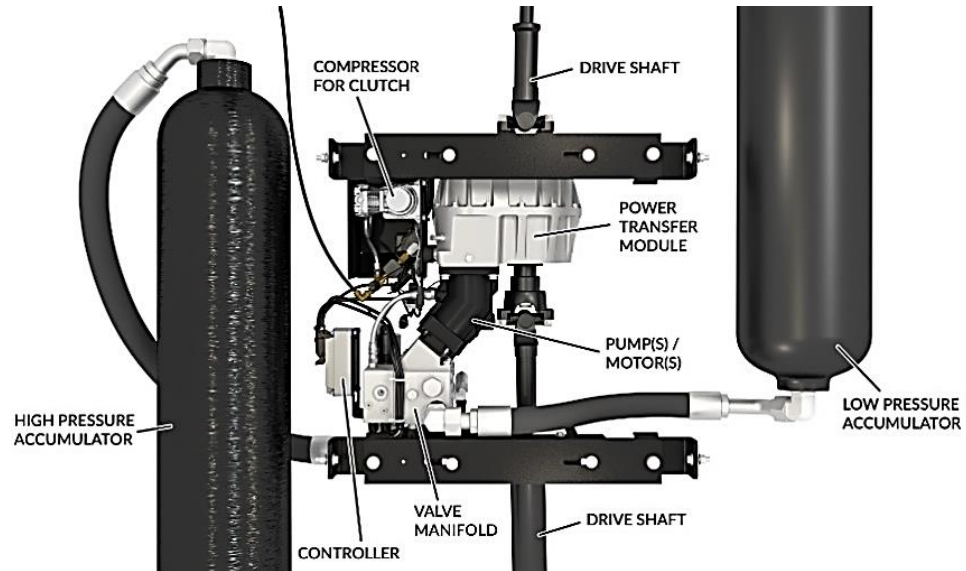


Figure 2.11 Main components of a Hydraulic Hybrid vehicle regenerative system [121]

In Figure 2.12 the different working phases are explained. During the RB event electric energy is gathered powering the hydraulic drive that, acting as a pump, moves the fluid from the low-pressure reservoir to the high-pressure accumulator; usually the entire closed loop system holds about 100 L of hydraulic fluid. An HHV is able in this way to recover up to 72% of otherwise wasted energy through the regenerative with a return of 56% of the braking energy to the drive wheels [23]. The high-pressure accumulator acts as an energy reservoir of a pressurised oil having the same energy storage function as the batteries in an EHV. The relevant difference is that, when stored-energy is required, the recovered power goes directly to the driveshaft, avoiding an additional energy conversion that is required instead in electric hybrid vehicles. During the acceleration, pressurised oil stored in the high-pressure accumulator feeds the hydraulic drive that, acting as a motor, drives the wheels and unpressurised fluid is stored in the low-pressure reservoir.

2 Literature Review

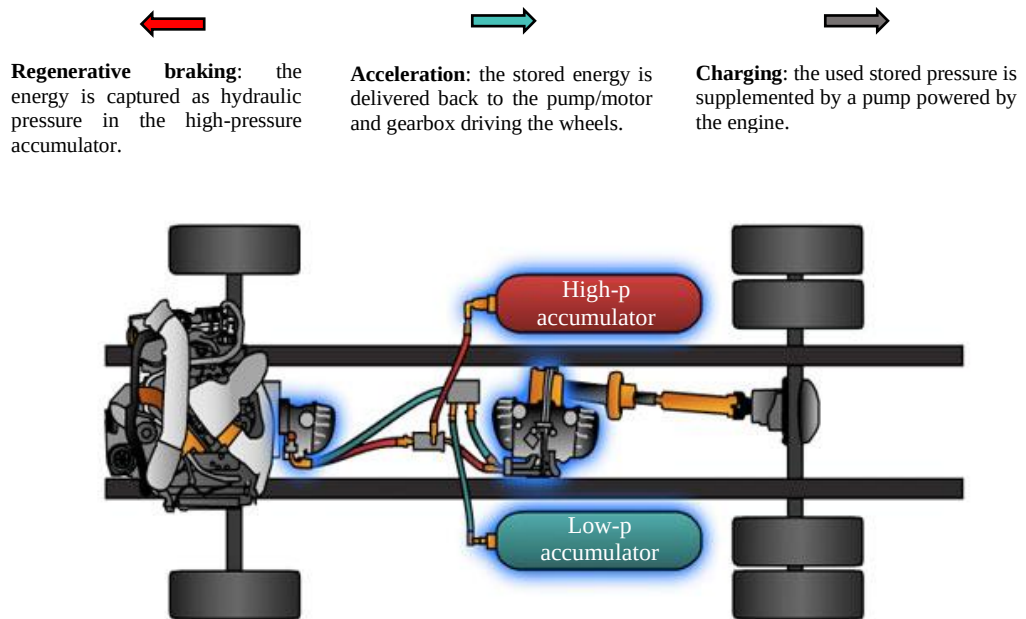


Figure 2.12 Basic working principle of HHV (modified from [120])

There are three ways to connect the components in an HHV:

1. *Parallel hydraulic hybrid:* the hybrid components are connected to a conventional transmission and driveshaft [122]. This allows the system to assist the engine in acceleration, however, it doesn't permit the engine to shut off at slow speed, thus the vehicle always consumes fuel.
2. *Series hydraulic hybrid:* the hybrid components are not connected to a conventional transmission or driveshaft, they transmit power directly to the wheels. This configuration reduces the number of components needed, to the benefit of the organic efficiency. Furthermore, this system enables the vehicle's engine to shut off, resulting in more fuel savings.
3. *Power-split hydraulic hybrid:* this approach combines the first two, the engine can be decoupled from vehicle load/speed maintaining however the possibility of transmission via the mechanical path [123].

Although EHV's have a more established technology than HHV, they are not so powerful as it would be expected. Their RB efficiency goes from 16% to 70% [124] but just when driving the wheel at low speeds; from 30 mph the gas engine

takes the lead and the vehicles operates as a normal ICE car driven [119]. On the other hand, the use of a variable displacement hydraulic pump/motor coupled with an accumulator has been getting the attention of the car manufacturer in the past 40 years. This type of system is able to potentially recover 80% of the braking energy [23] with fuel consumption of 21.7% over an idealised stop-start cycle [99]. EPA and United Parcel Service (UPS) developed a Hydraulic Hybrid delivery vehicle to explore the benefit in emissions and fuel economy [125]. Compared to a conventional UPS vehicle the advantages experienced in laboratory tests have been: 40% decrease in CO₂ emissions, 50% decrease in HC emissions, 60% decrease in particular matter emissions and a fuel economy improving up to 70%. EPA estimates that, when commercialised in high volume, the additional cost of an HH technology would be \$7,000 for the UPS package car, but the net lifetime saving would be, considering 20 years of an average lifetime for a UPS truck, over \$50,000. An additional study on United Parcel service and Parker Hannifin hydraulic hybrids has been carried out in 2014 [126] using a maximum pressure accumulator of 370 bar. The results confirmed the predicted benefits in fuel economy which was 19-52% better than conventional diesel parcel delivery van and 30-56% better than conventional gasoline delivery van. Artemis [127] carried out a study on a typical urban bus where the retrofit of a hydraulic system led to save up to 27% in fuel consumption. EPA, in collaboration with other partners, installed the HH technology on others demonstration vehicles [128]. A Full Series HHV Ford Expedition SUV experienced 85% improvement in fuel economy with a diesel engine, about two years of payback if produced in high volume and minimum net lifetime savings of £4,000 (in terms fuel cost and brake maintenance). A Parallel HH system in a Ford F-550 truck showed the ease of retrofitting trucks with this technology and a 20-30% of improvement in fuel economy. A Full Series HHV passenger car test chassis experienced a fuel consumption of 34 km/l, achieving 0-60 mph in 8 s with £700 on top of the base cost. Furthermore, a large SUV type and a midsize car were modelled and tested combining different HHV configuration with gasoline and diesel engines adopting “engine-on” and “engine-off” approaches and VDE (variable displacement engine). With an engine-on strategy the engine would only be shut down when the driver turns the engine off, whereas, with an engine-off strategy, engine operation would be

shut down whenever the vehicle is not moving. Results are collected and compared considering as a baseline a conventional transmission gasoline engine with a 5.4L V8 for the SUV and a 3L V6 2v/cyl for the midsize car and they are summarised in Figure 2.13. The most efficient combination of a HHV configuration on a SUV appears to be a Full Hybrid Clean Diesel VDE off with 101% of fuel economy improvement. The Clean Diesel VDE has already the best efficiency with a conventional transmission and keeps this trend varying the HH configuration. The gasoline engine shows the worst fuel economy improvement. Full HHV with engine-off configuration results in being the most efficient solution for all the different engines configuration comparing them with a conventional transmission configuration. The highest net lifetime savings is achieved with a full HHV engine-off configuration on a clean diesel VDE. Considering the saved fuel economy, the most convenient payback is actually achieved in 1.2 years using a full HHV engine off on a conventional gasoline engine. The same analysis carried out for a midsize car, provided similar results in terms of fuel economy improvement. However, in term of net lifetime saving, a midsize car finds its best combination having a full hydraulic hybrid system engine off in a clean diesel engine; the smallest payback time is 2.9 years and it is given from a gasoline variable displacement engine with a conventional transmission.

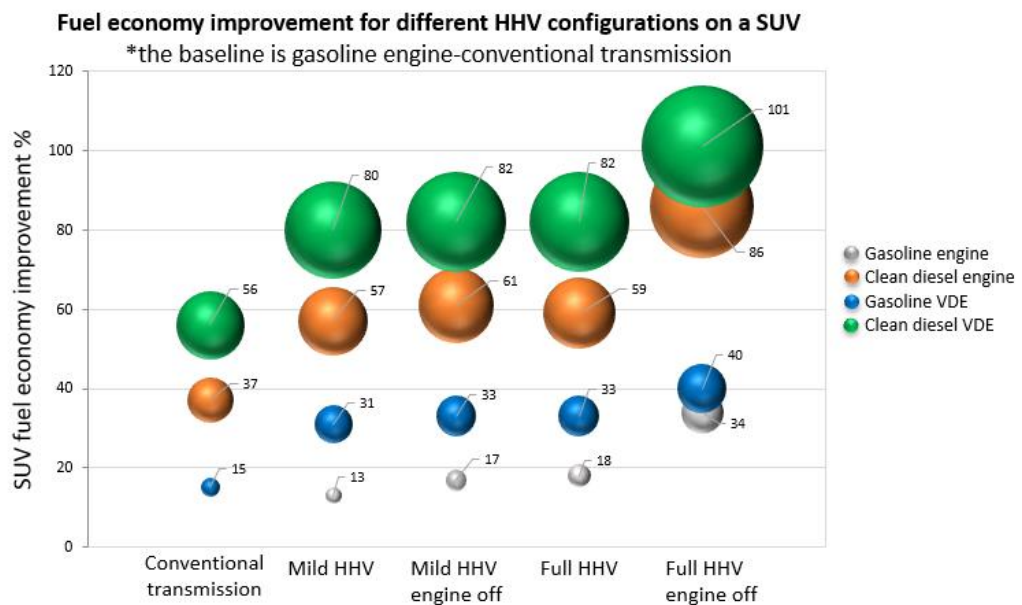


Figure 2.13 Fuel economy improvement to consumers for different HHV configurations on a SUV using gasoline or diesel engines (data from [128])

A Ford Hydraulic power assist study demonstrated that it is possible to achieve in a Sport Utility Vehicle, significant fuel economy savings operating with a hydraulic accumulator from 135 to 345 bar approximately [23]. Overall, the benefits on the EPA city cycle test were 23.6% fuel economy benefit and in terms of emissions -19% CO₂, -30% NO_x, -21% HC, -32% CO. The benefits were even higher with a heavier acceleration cycle. Another important aspect of improvement in the navigator vehicle was a 35% decrease in 0-30mph acceleration time (from 5.4s to 3.5s). The results from the Ford study are supported from a fuel economy improvement simulation of delivery trucks with HH propulsion. With the optimal power management strategy, the fuel economy was predicted to be up to 47% with an accumulator maximum pressure of 360 bar [129]. Another study found out a 32% of fuel economy improvement implementing HH propulsion into a medium truck with a max accumulator pressure of 350 bar [130].

Barbosa et Al. [131] modeled a small hydraulic hybrid vehicle (SHHV) to evaluate fuel economy improvement using a real-world driving cycle (instead of a conventional cycle like NEDC). The conventional passenger vehicle used was a Nissan March (2013) with a 1L engine; the results highlighted a fuel consumption reduction of 37% related to the standard passenger car. Another study which shows the advantage of using HHV in heavy-duty system is Macor et al. [132] which simulated a hydro-mechanical transmission in different configurations. Results confirmed a 40% recovery efficiency in the recovery of energy from brakings and an overall 20-35% energy saving. Matheson et al. [133] simulated a hydraulic regenerative system for use on heavy commercial vehicles using a matlab/Simulink environment implemented in the Advanced Vehicle Simulator software (ADVISOR). Results showed a decrease in acceleration time compared to a standard vehicle.

Bender et Al. [134] demonstrated how the application of hydraulic hybrid system is suitable for refuse collection vehicles because of their start and stop driving nature and of their hydraulic compactor system. Real driving cycle data were acquired such to compare the simulated ones; results showed that fuel consumption was -20% during daytrip and collection, and 22% during the transfer. Control strategies for hydraulic/electric synergy system in heavy hybrid vehicles have also been proposed to overcome the disadvantage of a single

energy storage source in heavy hybrid vehicles; a fuel economy improvement from 21.3 to 39.8% resulted from modeling different configuration on a New York city truck cycle [135]. In Table 2.2 a summary of the results is shown.

Summarizing, HHV have the following advantages:

- Lower emissions than ICE;
- Regenerative braking (70%);
- High energy density;
- Light materials but strong enough to survive major impacts;
- Economic materials and retrofit, considerably short payback;
- No chemicals involved;
- Easy retrofit in existing vehicles;
- Lower maintenance
- Reduction of engine exhaust during acceleration

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Vehicle type	Benefits	Reference
Simulation on Volvo FH-12 tractor	Idealised start/stop cycle: -21.7% fuel consumption Heavy-Duty Diesel Truck Transient mode: -11.2% fuel consumption New York city driving cycle: -17% fuel consumption Urban Driving Dynamometer Schedule legislative cycle: -11.7% fuel consumption	[99]
Ford Expedition SUV	85% fuel economy improvement	[136]
Ford F-550 truck	20-30% in fuel economy improvement	[136]
City bus	-27% fuel consumption	[127]
Lincoln Navigator SUV in EPA city cycle	Normal cycle -23.6% fuel consumption -19% CO ₂ -30%NO _x -21%HC -32% CO Heavier acceleration cycle 35.5% fuel economy improvement -26% CO ₂ -40%NO _x -14%HC -43% CO	[23]
Simulation on Delivery trucks	47% of fuel economy improvement	[129]
Medium truck	32% of fuel economy improvement	[130]
Delivery van	19-52% fuel economy improvement (compared to diesel engine) 30-56% fuel economy improvement (compared to gasoline engine)	[126]

2 Literature Review

Simulated vehicle	+36.3% accelerating performances +35.59% fuel economy improvement	[137]
Modeling SHHV in a real-driving cycle	-37% Fuel consumption	[138]
Simulation HHV for a refuse vehicle in a real-driving cycle	-21% average fuel consumption	[139]
Simulation of hydro-mechanical transmission	40% recovery efficiency in braking 20-35% energy savings	[140]
Simulation NEDC passenger car	-50% fuel consumption	[141]
Simulation synergic electric-hydraulic hybrid on NYCT (New York City Truck) cycle	-39.8% fuel consumption	[142]

Table 2.2 Summary of benefit using a Hydraulic Hybrid configuration in different types of vehicles

2.5 Windage losses overview

2.5.1 Windage losses overview

Windage is a force created on an object by friction when there is relative movement between a fluid and the object; windage loss is the reduction in efficiency due to windage forces. The mechanism of windage loss occurring in the dynamic of the Pelton spinning in a small enclosure with a mixture of air and oil is similar to the losses experienced in gearboxes [143]. Windage power loss (WPL) is then defined as the power loss that a rotating body with or without teeth/buckets experiences dragging air or air-oil mist. The parameters that influence WPL are: geometry, degree of confinement, density of the fluid surrounding the body and the linear speed. Particularly, the higher the speed, the higher the gear vulnerability to the windage loss [143], [144]. A comprehensive review was carried out by Eastwick and Johnson [143] regarding windage losses, the main equation and theoretical approaches available in literature are summarised and adapted for the PhD research application. This review summarises the windage loss torque for a free rotating disc, for an enclosed rotating disc in a single-phase fluid, for a cylindrical body rotating with losses associated to blades and finally for a partially wet disc rotating in an enclosure. In this section, the analysis for the simple assembly Pelton-inertia disc and compressor/turbine wheel are analyzed to select the most suitable equation in the model.

The approach to study windage losses for a rotating body is based on the boundary layer theory. The boundary layer is the layer of fluid in the immediate vicinity of a bounding surface where the effects of viscosity are significant [145] (Figure 2.14). The boundary layer is then divided in laminar and turbulent; the laminar boundary is a very smooth flow creates less skin friction drag, while the turbulent boundary layer contains swirls or "eddies" but it is more stable. The boundary layer flows over a wing surface and it begins as a smooth laminar flow; as the flow continues back from the leading edge, the laminar boundary layer increases in thickness. At some distance back from the leading edge, the smooth laminar flow breaks down and transitions to a turbulent flow. This theory can be applied to the case of a single disc rotating in an initially stationary fluid [146]. Having a disc of radius r , rotating at an angular velocity ω (Figure 2.15), a boundary layer develops on each side of the disc, the no-slip condition makes

the tangential velocity of the fluid being $u_\phi = \omega r$ at the disc and zero in the free stream outside the boundary layer. A radial flow develops in the boundary layer thanks to the centrifugal forces generated by the shear between the rotating disc and the fluid; the radial component of velocity is zero both at the disc surface and in the freestream to satisfy the conservation of mass. The radial outflow is referred as the “pumped flow” and it causes the free-disc pumping effect. Following this introduction of the flow behaviour, Von Karman laminar and turbulent friction studies [147] set the basis for investigate the windage losses torque. During his studies Von Karman determined that the solution of Navier-Stokes equations identifies that the resulting tangential velocity varies linearly across the gap from 0 on the stator to ωr on the rotating disk. The tangential shear stress τ_t can be derived on the rotating disc considering the velocity gradient $\frac{du}{dy}$ in equation 2.14 using the notation in Figure 2.14.

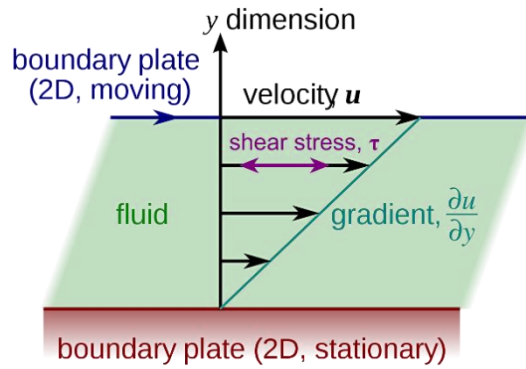


Figure 2.14 Notation diagram for boundary layer theory

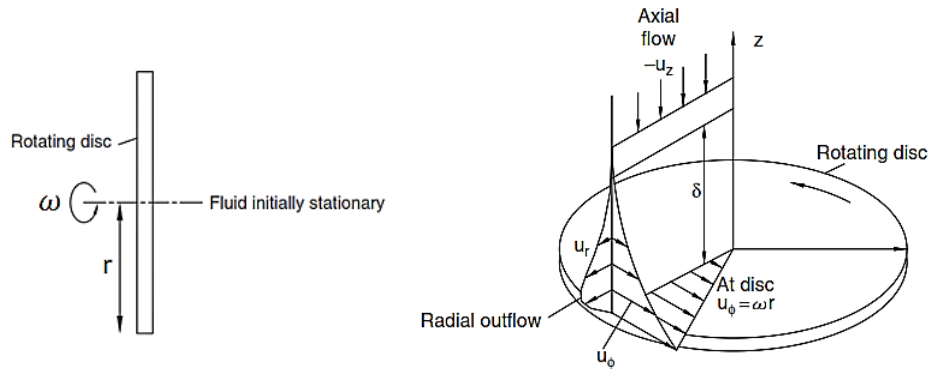


Figure 2.15 A single disc rotating in an initially stationary fluid (left), the radial and tangential boundary layer associated inflow and outflow induced by a rotating disc (right)

$$\tau_t = \mu \frac{du}{dy} ; \quad \frac{du}{dy} = \frac{\omega r}{s}; \quad \tau_t = \frac{\mu \omega r}{s} \quad 2.14$$

Replacing the value of τ_t into the frictional torque equation 2.15 for the two sides of the disc (but not for the edges) the final equation 2.16 for a torque associated to windage losses is derived [147]. The torque coefficient C_m is given in equation 2.17 for a free disc in laminar flow and in equation 2.18 for a turbulent flow.

$$T = 2 \int_0^a 2\pi r^2 \tau_t dr \quad 2.15$$

$$T_w = \frac{1}{2} C_m \rho \omega^2 r^5 \quad 2.16$$

$$C_m = \frac{3.87}{Re^{0.5}} \quad 2.17$$

$$C_m = \frac{0.146}{Re^{0.2}} \quad 2.18$$

An important parameter to refer for windage losses is the Rotational Reynolds number (equation 2.19). Inside the disc boundary layer, the flow regime is governed by the ratio between the inertia to the viscous effect and this is strictly related to the disc geometry characteristic.

$$Re = \frac{\omega r^2}{\nu} \quad 2.19$$

Where angular speed ω in this case is expressed in rps. The flow near the axis z of rotation tends to be laminar, however, for large values of angular velocity the flow can become turbulent even at low radius and there is a transition between laminar flow and turbulent flow zones. The firsts to investigate better the phenomenon and develop the formula and the theory for the boundary layer

applied in windage losses based on Von Karman studies for a rotating disc were Nece et al. [148][149]. Their experimental study consisted in testing multiple configurations of an enclosed rotating disk rotating in air. Considering the radius r of the disc and the gap s between the disc and the enclosure, the results were obtained for a range of variables of the ratio s/r from 0.0127 to 0.217 and for values of Re in acceptable range for both laminar and turbulent flow regime. The minimum gap to consider valid the equation was 0.2mm. The authors identified overall 4 flow regimes between the area of the disc and the enclosure (Figure 2.16). As an example, study of Regime I was investigated firstly by [150] and [151]; it is assumed Couette flow (the flow of a viscous fluid in the space between two surfaces) one of which is moving tangentially relative to each other. The flow is driven by the drag force acting on the fluid and by a possible additional pressure gradient in the flow direction. No radial velocity component is present and the effect of the friction on the cylindrical walls is ignored. Table 2.3 reports the values of C_m for all regimes.

Further studies have been carried out on the phenomenon for this simplified geometry. Etemad [152] provided an equation for the power loss associated to an enclosed rotating disk with no net mass transfer between rotor and stator. Ketola and McGrew [153] carried out experiments on a partially wet enclosed rotating disc, providing new C_m values for the four regimes.

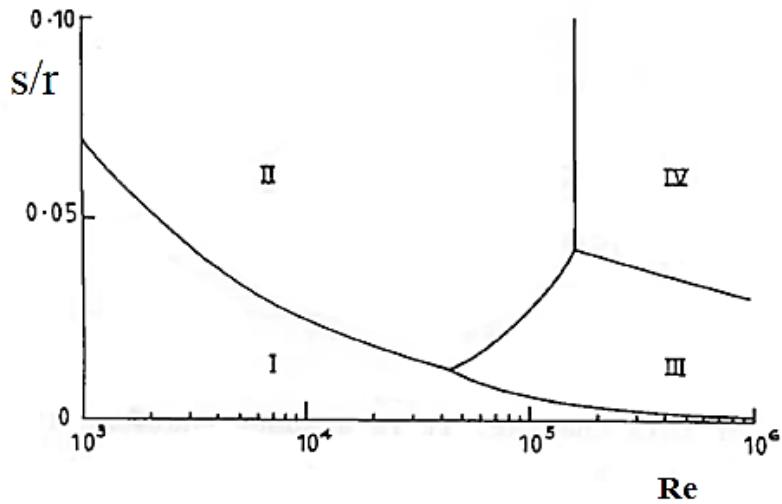


Figure 2.16 Type of regime expected from Daily and Nece studies [148]

given the Reynolds number and the ratio of the axial clearance s to the disc radius r [233]

Regime	Description	Theoretical equation for C_m		Reference
I	Laminar flow, close clearance. Merged boundary layers lead to a continuous variation of velocity across the axial gap *	$C_m = \frac{2\pi}{(\frac{s}{r})Re}$	1.19	[148]
II	Laminar flow, separate boundary layer, axial gap is greater than twice the boundary layer thickness and it contains the rotor and stator separate boundary layers. Between the boundary layers there is a core of rotating fluid.	$C_m = \frac{3.7(\frac{s}{r})^{1/10}}{Re^{1/2}}$	1.20	[148]
		$C_m = \frac{2.66}{Re^{1/2}}$	1.21	[154]
		$C_m = \frac{2.6}{Re^{1/2}}$	1.22	[150]
III	Turbulent flow, close clearance. A turbulent regime I with high Re. With a large value of axial clearance, this regime could never be observed.	$C_m = \frac{0.0622}{(\frac{s}{r})^{1/4} Re^{1/4}}$	1.23	[151] [148]
IV	Turbulent flow, separate boundary layer. A turbulent regime II with higher Re. The transition from regime III to this regime depends upon the ratio s/r ; as this ratio increases, Re can be lower in order for the change of regime to occur.	$C_m = \frac{0.0102(\frac{s}{r})^{1/10}}{Re^{1/5}}$	1.24	[148]
		$C_m = \frac{0.0622}{Re^{1/5}}$ [79]	1.25	[154]
		$C_m = \frac{0.0836}{Re^{1/5}}$ [76]	1.26	[150]

Table 2.3 C_m values for different flow regimes in windage losses study

2.5.2 Windage losses in gear/turbines geometry

The works discussed before highly contributed to the windage losses basic theoretical approach. When these findings are applied to a more realistic scenario, it becomes clear that specific geometrical aspects must be studied to fully comprehend the behaviour of windage losses in real systems such as turbine wheels. The windage friction loss in these systems can be attributed to two different parts, the turbine schematised disc and the turbine buckets sides. The buckets alone were found to be responsible for the majority of the windage loss, and the height of the buckets was found to have a significant impact on the loss. [155]. In this regard, the comparison of power loss at high and low face widths was investigated and led to the conclusion that the teeth draw air axially ejecting it radially and/or tangentially toward middle face. The theory of gear/wheel/turbines windage losses was investigated by different researchers. Lord [156] approached the single-phase flow in gear windage considering the gear geometry composed by two bodies: the main body, modelled as a smooth or rough disc and the circumferential face of the gear/turbine, modelled as a disc of finite thickness. The power losses experienced are assumed to occur due to: the pumping of air from the centre of a gear disk to the outer periphery, the pumping of air through the teeth, the passage of a tooth through the wake of the preceding tooth. Anderson and Loewenthal [157], Dawson [158] and Matsumoto et Al. [159] proposed other equations for windage power loss based on empirical experimental parameters, however it is specified that these cannot be considered reliable for more generic systems. Another interesting characteristic to consider in the theory of windage losses, is the impact of the disc/wheel surrounding. In single-phase flow, an enclosed turbine disc experiences less windage power loss compared to a turbine disc in free air with no restrictions [155]. Etemad et al. [152] in their research used an adapted coefficient from [148] investigating the effect of the air gap size and determining to be insignificant on its own. Furthermore, they showed that the losses predicted by the equations were larger than experimental results.

The impact of the shrouding, however, highly depends on the type and state of the fluid. Mann and Marston [160] carried out a study to investigate the friction drag on bladed disc in housing, extending Daily and Nece initial work [148]. In a disc, only viscous forces give energy to the circulating fluid, whereas in wheel,

momentum exchange in the centrifugal field caused by the blades also imparts energy to the fluid [160]. According to Mann, the drag effect increases proportionally to the density of the fluid and the torque coefficient takes the form of $C_m = K/Re^{\gamma_w}$; when $\gamma_w=1$ drag is independent of density and it only depends on viscosity, when γ_w goes to zero, density effect matters. This is however just a qualitative analysis given from the author. The experimental results showed that as the aspect ratio (blade height/disc thickness) increased, the moment coefficient increased as well. Further experimental studies from Lord [161] confirmed that windage losses are reduced reducing both axial and peripheral clearance having a single-phase gaseous flow. Lord also explored two-phase flow experimental investigations addressing the aspect of a single spur gear shrouded with oil injection. This can be considered a similar case to the Pelton in KERB system rotating in oil-air mixture in small-casing. The results showed that in the meshed spur gears, an increasing oil flow rate caused the power loss to be increased. Lord also investigated the effect of single gear shrouding and draining hole size in two-phase flow. The oil flow rate was constant and the drain hole was either 20mm or 40mm, varying the axial clearance while keeping constant the peripheral clearance at 1mm. Comparing single-phase and two-phase flow in same condition it was shown that the increased density and the misty condition created by the oil particle worsened windage losses. Having just two-phase flow experiments, increasing the axial clearances maintaining the peripheral clearances constant, results in increasing slightly the windage losses. It was also observed that above 3000 rpm the oil was unable to drain out of the chamber properly and it was pulsed out when the pressure increased. Varying the peripheral clearance and keeping constant the axial clearance at 1mm, resulted in a dramatical increase in windage losses introducing oil into the shrouding in same conditions of the single-flow. Windage losses were found to be proportional to the flow rate and the increase of the peripheral clearance reduced the windage power losses significantly. Keeping the shrouding constant at 1mm clearance and the flow rate at 2.5 l/min it was investigated the effect of the draining diameter variation at 20mm an 40mm on the windage losses. These were dramatically reduced when the 40mm draining hole was leading to the conclusion that an optimum draining system is essential to reduce windage losses in a two-phase flow. Lord also showed that the increasing temperature of

the oil has a beneficial effect on windage losses. Summarizing, having a spur gear, rotating in a two-phase flow mixture, the main results from Lord experiments were:

- The shrouding and increased density increased windage losses;
- The increase of axial clearance slightly increases windage losses, but results fluctuation does not give reliability to the outcome;
- The increase of peripheral clearance decreases windage losses;
- Windage losses increase increasing the flow rate;
- An optimised draining strategy is essential to keep windage losses low. A double-diameter size hole reduced windage losses of more than 55%.
- Windage losses increases as the speed increases.

The work of Lord was related strictly to spur gears however, the observations above can be taken as starting point to consider the effect of shrouding in the Pelton wheel in the HAT. These are of specific interest for the draining strategy to optimise the rotational speed of the assembly.

The accurate calculation of friction and windage losses in wheels/gears depend on the design of turbine casings and the specific density of wet air and they are usually determined by the so-called retardation test [162]. During this test, the time-dependent rotational speed of the Pelton wheel in the form $n = f(t)$; the calculations for the windage losses are manipulated using the moment of Inertia I of the rotating system and angular momentum. The final equation of the rotational speed is dependent from the moment of Inertia, and from some constants derived from a curve-fitting calculation using the nonlinear least-squares method.

2.6 Summary

The literature review chapter provides the theoretical background and the relevant tools to understand the value of the research project. The HAT is framed in the engine's improvement solutions panorama to mitigate Climate Change. The main takeaway messages from the chapter are listed as follows:

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- Emissions from ICE are among the main concerns for Climate Change and their reduction in traditional and hybrid vehicles can be addressed with rightsizing techniques.
- Rightsizing strategies include Superchargers and Turbochargers; the latter are preferred as they recover otherwise wasted energy. Coupled with a kinetic energy recovery system they can be part of the system to reuse up to 70% of useful motor energy [94]. Specifically, hydraulic recovery allows up to 25% of fuel economy improvement [99].
- The assisted turbocharger technology represents a valid solution for the turbo-lag issue; tested electric assist can lead to a 43%-70% decrease in response time [82][83] however, the required power to assist a larger turbocharger is more demanding [20]. The hydraulic assist is an attractive alternative for heavy-duty applications because of lower inertia cost and packaging; additional research is required on the potential benefits.
- Hybrid Electric vehicles are an almost established technology in passenger cars however, there are still significant challenges for the heavy-duty application, where the size of the batteries represents the main obstacle. Hydraulic hybrid vehicles are a feasible solution for their supremacy in power density capability and for the capacity of recovering up to 56% of the braking energy to drive wheels [23].
- The internal combustion engine will still be widely present in heavy-duty and off-road vehicles due to power/torque requirements, but mitigation strategy needs to be applied to meet the emissions reduction legislation. A hydraulic-assisted turbocharger coupled with a hydraulic hybrid vehicle is envisaged as a potential compromise for heavy-duty vehicle efficiency improvements.

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- The theory on windage losses for rotating wheels gives numerous but empirical formula which can result in significant variation in prediction of the final speed. Insufficient literature is available for windage losses of enclosed wheels in a two-phase fluid dynamic regime.

3 Experimental Setup and Methodology

3.1 Introduction

The Introduction and Literature review chapters introduced the research Subject of the thesis; the following chapter gives an overview of the experimental apparatus designed and built to replicate conditions when the assisted turbocharger is implemented in an HHV. The prototype was inherited from a previous project at Brunel University [49] where it was never tested; the starting point of the experimental investigation was then to assess the preliminary functioning. The chapter described the performance shakedown and identifies potential constraints and game-stoppers for the definitive experiments. No previous methodology details are available in literature regarding a Pelton turbine mounted on a standard turbocharger shaft except from the previous work mentioned on [49] which was however needed to be innovated and readapted for the HAT complete application.

The chapter section is divided in 3 parts. Firstly, an overview of the rig schematic is given focusing on the components and the data acquisition strategy.

Secondly, the Design of Experiment (DOE) strategy applied to the experimental campaign, to ensure reliability of results, is explained

Finally, the experimental shakedown procedure is discussed and main conclusions are summarised focusing on hydraulic lock investigations, sensor testing and dummy chamber testing.

3.2 Experimental Rig Design

3.2.1 Overview

The experimental rig aims to reproduce the working condition of the turbocharger prototype in a HHV or, generally speaking, in a vehicle in which hydraulic power source is available. Initially, the air source for the gas turbine has been neglected. The implementation of further components to replicate the

turbocharger functioning in engine-like condition is explained in Chapter 6. The complete rig schematic is given in Figure 1.1. A focus on the components is carried out in section 3.2.2. In the standard vehicle configuration, the hydraulic accumulator is kept pressurised through the regenerative brake of the vehicle; to simulate this, in the experimental rig, referring to Figure 1.1, the accumulator is pressurised with Nitrogen up to 280 bar (1). The hydraulic oil is first fed into the hydraulic accumulator (3) through a low-pressure pump (9); once the connection with the pump is closed, the oil is pressurised to the desired pressure with Nitrogen gas on the other side of the accumulator bladder. The oil feed from the accumulator then splits to go to two nozzles (19) on each side of the Pelton. Once the oil has been injected into the Pelton it then drains through the oil drain hole into a reservoir (15); if needed, a gear scavenge pump (14) is used. The oil in the reservoir is then pumped back up to the feed reservoir (4) to be reused. The oil for the bearing feed (if the bearings need to be externally fed) is pumped directly from the lower reservoir and returns the same way through the drain hole. In sections 3.2.3 and 3.2.4 the two main hydraulic circuit are explained.

3.2.2 Components

The experimental Rig components and their function are listed as per schematic in Figure 3.1:

1. *Nitrogen gas bottle*: the Nitrogen bottle is required to inflate the hydraulic accumulator bladder, providing potential pressure energy for the oil on the liquid side. Depending on the pressure desired a single 300 bar bottle or a bank of bottle of 15x300 bar Nitrogen are supplied by BOC.
2. *High-pressure regulator*: the regulator is mounted on top on the Nitrogen bottle and it is required for regulating the charging pressure. The cylinder pressure regulator used was a Hornung HD 400 FL rated 0-400 bar.

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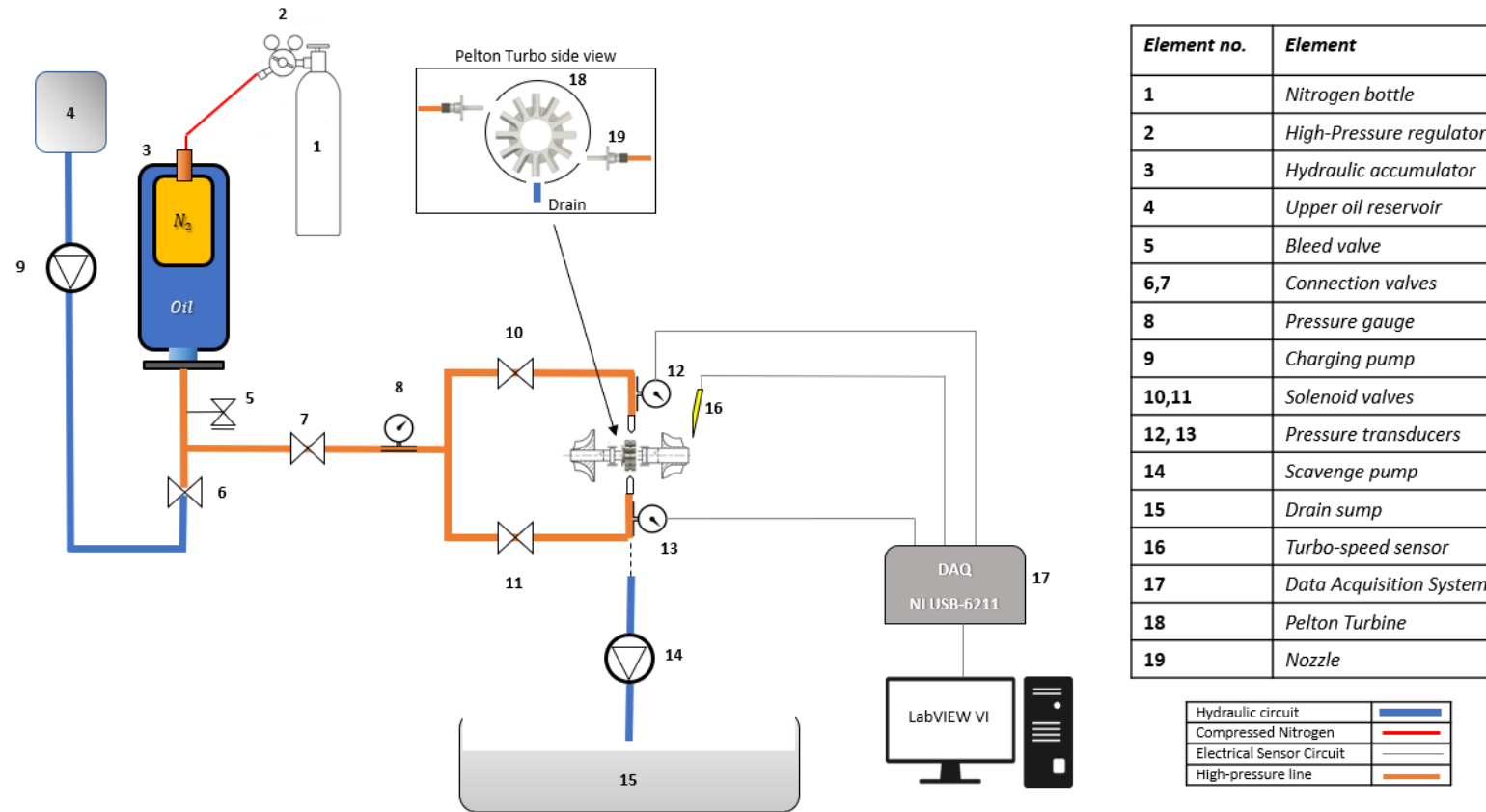


Figure 3.1 Hydraulic rig schematic of KERB-HAT testing

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3. *Hydraulic accumulator*: the hydraulic accumulator available for the rig was a 20 L oil volume capacity Parker-Hannifin EHV 20-330/90-A25GB with maximum working pressure 330 bar. The high-pressure system stores the high-pressurised oil which addresses the two nozzles; usually, hydraulic accumulator is pre-charged to a certain pressure and they receive oil from the hydraulic oil system which pressure rises thanks to the regenerative braking. In the proposed experimental rig the accumulator functions in the opposite way, hence it is first filled with oil and then charged with Nitrogen. For this purpose, a customised charging procedure has been identified. The accumulator is equipped with a *Gas Charging kit*; this is required for opening and closing the hydraulic accumulator gas side, for monitoring the accumulator pressure and for releasing the pressure from the accumulator; it is composed by an a3 universal adapter, a main body to attach to the a3 adapter, a gas relief valve, a main spindle, a pressure gauge and hoses connections.
4. *Upper oil reservoir*: it accommodates the charging oil for the hydraulic accumulator. It is made in house in the workshop of engine's laboratory and equipped with a transparent pipe on the side to check the oil level when charged.
5. *Bleed valve*: situated under the hydraulic accumulator, it acts as a control device used to purge the system from the oil.
6. *High-pressure all valve*: two high-pressure ball valves (500 bar rated). These valves address the oil in the right path depending on which stage the experiment is taking place (running or refilling upper tank).
7. *High-pressure ball valve*: this valve addresses the oil from the accumulator to the nozzles when the experiment runs.
8. *Pressure gauge*: it provides an instantaneous reading of the pressure to safely check the pressure in the hard high-pressure pipes that connect the hydraulic accumulator to the nozzles.

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9. *Charging pump*: to fill the accumulator with the desired quantity of oil and to investigate repeatability for the tests identifying the approximate amount of oil filled inside the accumulator.
10. *Solenoid valves 1st assembly*: the assembly is composed by a ball valve, a pneumatic actuator, and a solenoid valve. This configuration has been designed to remotely control the opening of the valves from the Labview software. The design is based on a prompt and safer technique to activate the valves instead of the manual alternative. The ball valve is normally closed and it is activated by a pneumatic actuator supplied with compressed air (6 bar max) and controlled by a solenoid valve supplied with 24V. This assembly open the oil flow from the accumulator; it also acts as emergency control device shutting off in case the set limits of turbo speed are exceeded. All the components are supplied by Swagelok.
11. *Solenoid valves 2nd* : same assembly as per item 10.
12. *Pressure transducer*: the Omega Engineering high performance pressure transducers model PXM5500MC6-400BARG5T provide the monitoring of the pressures next to the nozzles to give a reading of the exit pressure. They are located next to each nozzle to provide the pressure at the nozzle inlet, bypassing the pressure losses from the accumulator. They are piezoresistive pressure transducers in which the change in resistivity of the material (then the resistance of the Wheatstone bridge built inside) is caused by mechanical stress of semiconductor elements. The electrical variation is then directly related to a pressure variation through a calibration curve. The accuracy is $\pm 0.08\%$.
13. *Pressure transducer 2*: same as per item 12.
14. *Oil Scavenge pump*: The gear pump Tuthill 1012 provides oil drain from the turbocharger casing and recirculation to the top oil reservoir or to an additional reservoir to measure the flow rate.

15. *Lower Oil reservoir*: this reservoir receives the oil from the turbocharger drain hole or the scavenge pump. The low tank also provides the oil required for the lubrication of the turbocharger bearings when required.
16. *Turbo speed sensor*: it provides the monitoring of the rotational speed of the Pelton turbine through the detection of rotational speed from the compressor blades. The model used is an inductive proximity sensor from Microepsilon and it is a DZ140 DS05(15). The sensor is the AECS type (active eddy current sensors). The sensor probe has its own impedance; the probe is supplied with AC creating a primary time-varying magnetic field. When one of the aluminium blades of the compressor passes through this magnetic field, it creates “eddy currents” inducing a secondary magnetic field which modifies the probe impedance. This is then converted in rotational speed thanks to the calibration curve and the DAQ. The accuracy is 1%.
17. *DAQ*: the system is composed by the computer station, the National Instruments DAQ USB-6211 and the LabView VI programmed to control remotely the rig (opening and closing solenoid valves) and record data (pressures and turbo speed).
18. *Turbocharger prototype*: the turbocharger prototype has a Pelton turbine mounted in the same shaft of the compressor and the turbine; the Pelton is supposed to rotate thanks to the action of the high-pressurised oil coming from the nozzles. The prototype can be replaced by the dummy chamber or the free-running Pelton depending on which type of test want to be performed.



Figure 3.2 Hydraulic rig model design (left) and actual rig (right) of KERB-HAT

When testing the original prototype, an additional lubrication oil circuit required for plain bearings was included which included a low-pressure pump and a low-pressure regulator. The pump was injecting the necessary flow rate of oil at 2 bar as advised from standard ratings [163], [164].

3.2.3 High-pressure hydraulic circuit

The high-pressure hydraulic circuit is dedicated to the pressurised oil for the Pelton turbine. It is shown in orange in Figure 3.1 and Figure 3.3 and it comprises a nitrogen gas bottle, the hydraulic accumulator, the bleed valve, ball valves, the solenoid valves, the pressure gauge, the pressure transducers, the nozzles, and the turbocharger prototype. Once the accumulator has been charged with oil from the upper reservoir, the high-pressure circuit is filled with oil and less or no air is present in the circuit; a detailed charging procedure allows to pressurise the oil inside the hydraulic accumulator at the desired pressure. When solenoid valves are automatically opened by remote control, the oil is ejected through the two nozzles. To reach the solenoid valves, hence the nozzles, the line splits into two branches through a tee junction; the two branches have the same length, and this action, together with the synchronised time response of the

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two electronic solenoid valves, ensures to have equal pressure and flow rate along them.

All components in this circuit have been strictly selected and rated for the high-pressure requirement.

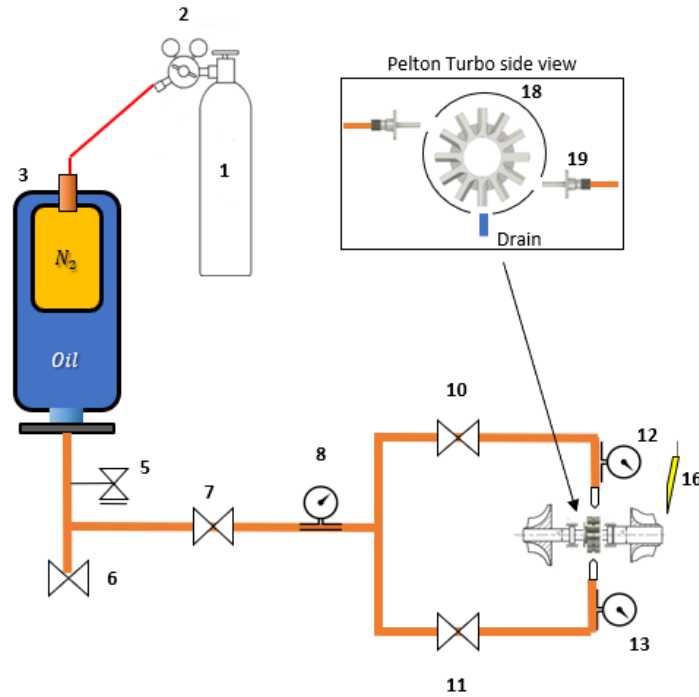


Figure 3.3 High-pressure circuit of KERB-HAT

3.2.4 Low-pressure Hydraulic circuit

This circuit involves the bearings lubrication line, the drain line, and the recirculation and feed line.

- Bearing lubrication line: this line comprises the lower oil reservoir, the oil-bearing pump and the oil pressure reducing valve. In standard turbocharger bearings lubrication plays a vital role in the performance and life of rolling elements in the shaft, indeed the lubricant has the important task to separate parts moving relative to one another avoiding friction and wear. To achieve the right amount of flow rate to delivery for the bearings lubrication, an electric bearing pump provides 2-4 bars of low-pressurised oil withdrawing it from the lower oil reservoir. An oil pressure reducing valve allows to change the delivered pressure because since the output power of the pump is constant.

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- Drain line: this line comprises the oil scavenge pump and the lower oil reservoir. The amount of oil entering the turbocharger from both the lubrication flow rate contribute and the high-pressurised oil flow rate from the nozzles needs to be drained through the drain hole. Because of the high flow rate that is reached corresponding to a high-pressure charging of the hydraulic accumulator, a scavenge pump is required in the OTP (original turbocharger prototype) to discharge the oil from the drain hole to the lower oil reservoir.
- Recirculation line: At the end of the tests to fill the upper tank to begin a new cycle of testing, the scavenge pump provides the necessary pressure to recirculate the oil from the lower to the upper tank.
- Feed line: it comprises upper oil reservoir, ball valves and electric pump. The electric charging pump provides the necessary pressure for filling the accumulator with oil.

3.2.5 Working fluids

The fluids used in the experimental rig are Nitrogen gas and Lubrication oil. The oil selected for the experimental testing was different from the engine oil traditionally used for turbochargers, This, would have had indeed a different viscosity due to the lower temperature of testing. The selected BP oil has a viscosity at ambient temperature which is in the range of the one expected from the oil 5W-20 when heated during engine operation cycles. Physical characteristics of both oils are available in Table 3.1 [165][166]. The computational model presented in Chapter 4 clarify, however, that the impact of the different viscosity of the oil on the final rotational speed is less than 2%.

	BP 2380	5W-20
<i>Density @ 15°C [Kg/ m³]</i>	0.9749	0.848
<i>Kinematic Viscosity @ 100°C [mm²/s]</i>	4.97	8.88
<i>Kinematic Viscosity @ 40°C [mm²/s]</i>	24.2	49.35
<i>Pour Point [°C]</i>	-57	-38
<i>Flash Point [°C]</i>	265	>200

Table 3.1 Testing Oil properties

Nitrogen is charged into the hydraulic accumulator to pressurise the oil. After the oil is discharged, the remaining Nitrogen in the accumulator needs to be discharged as well, to be able to perform new test. Nitrogen is an inert fluid but in a certain percentage can cause asphyxia; calculations on the amount of N₂ released in the air to discharge were carried out to ensure a safe environment for operating the system. This can be found in APPENDIX A: Calculation for safe Nitrogen discharge after testing.

3.2.6 Data Acquisition System

To collect data from the physical experimental rig, a DAQ from National Instruments was used, model USB-6211. A Labview VI was created to collect and manage data from sensors connected to the experimental rig to the DAQ. As a first approach a DAQ assistant block was used in the programme, however, shakedown tests revealed that this system was not suitable to accommodate the amount of data from the physical DAQ as the USB-6211 does not support hardware-timestamping. It does not have a real-time clock inside; therefore, when it acquires a measurement, it does not link that value to a point in time, this needs to be done later in software. A good workaround was implemented into the DAQmx driver: this tries to evaluate the moment the USB-6211 started the acquisition, and then uses the acquisition rate to advance in time. As the acquisition itself is hardware-timed (the USB-6211 uses an internal timer), that rate was shown to be very stable.

Moreover, all Express VIs (Virtual Instruments) like DAQ Assistant and "Write to Measurement File" are developed to allow quick development of very simple VIs, but the requirements of the research subject application were some orders of magnitude above that. A better approach was then developed using **modularity** strategy, that means breaking up the macro task in multiple mini tasks better manageable by the programme. The main VI comprises different states according to the different operations of the rig and uses numerous sub-VI for performing auxiliar tasks. In the main VI a producer/consumer structure is used: the first loop acquires the data and open the safety valve whereas the consumer loops are used to update all Front Panel graphs and to write in a .csv file. There are 5 states in the main VI:

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- "Init" sets up the DAQmx tasks and closes the solenoid valves. This state always transitions into the "Acquire" state.
- "Acquire" reads the raw data, converts the voltages into other data (pressure, speed, etc.), and checks if maximum values are exceeded. Normally, the next state is "Set Valve Position", except one of the stop conditions happened, in that case it goes into "Stop" next.
- "Set Valve Position" outputs the "Override" buttons state to the Digital Line. Goes back into "Acquire" state afterwards.
- "Stop" closes the valve and goes into "clean up".
- "Clean Up": Stops the DAQmx tasks, closes the queues as soon as they are empty, and stops the producer loop afterwards.

Labview code has been tested with a power supply to see the response of the sensors and after setting the proper limits and the data acquisition mode, the wave chart reflected the voltage supply. The programme, when started, automatically records and saves the data in a .csv file specified in the file path given. The default frequency of samples to read is 1 to 100Hz. The value in the box on the front can be changed setting the range limits and according to these limits the code stops registering data and close the solenoid valves. The solenoid valve button open and close the solenoid valves and the stop button arrests the program. If the program is stopped, the solenoid valves are automatically closed because the voltage is interrupted. There are three graphs in the front panel that show in real time the data recorded of the sensors connected. Once the program is stopped, a .csv file is automatically created and saved in the selected folder.

3.3 Design of Experiments (DOE)

In this section DOE is introduced the simplified procedure to determine statistical significance of experimental tests and parameters variation is explained.

An experiment aims to predict the outcome by introducing a change of the preconditions (independent variables) also referred to as "input variables" or "predictor variables." The change in one or more independent variables is generally hypothesised to result in a change in one or more dependent variables,

also referred to as "output variables" or "response variables." The experimental design identifies control variables that must be held constant to prevent external factors from affecting the results. It involves the selection of suitable independent, dependent, and control variables and planning the delivery of the experiment under statistically optimal conditions given the constraints of available resources. Reliability of results can be partially addressed by carefully choosing the independent variable and reducing the risk of measurement error. Design of experiments (DOE) applies principles and techniques at the data collection stage to ensure the generation of valid engineering conclusions. There are different areas in which DOE is applied and all of them were used during the experimental campaign to test and optimise the turbocharger prototype:

- *Screening Characterization*: "understanding" the process as a whole in order to, after design and analysis, rank a list of important factors that affect the process.
- *Comparison*: assessing whether a change in a single factor has in fact resulted in a change/improvement to the process as a whole.
- *Modeling*: functionally modeling the process with the output being a good-fitting (high predictive power) mathematical function, and to have maximal accuracy in estimating the coefficients in that function.
- *Optimizing*: determining optimal settings of the process factors; that is, to determine for each factor the level of the factor that optimises the process response.

The first step is understanding if the collected data are reliable, and the number of tests required to ensure a good representation of the phenomenon; this is done firstly checking that the data follows a normal-gaussian distribution using the q-q plot and then building a Control Chart that verifies the random behaviour of the tests. The q-q plot is a quantile-quantile plot and it is a graphic method that provides an indication of the normality of the dataset; it differs from a probability plot because it shows observed and expected values instead of percentages on X and Y axes. The data are sorted in ascending order, and then plot versus quantiles calculated from a theoretical distribution. If the data is normally distributed, the

points will fall on the 45-degree reference line. If the data is not normally distributed, the points will deviate from the reference line.

For each configuration at least 3 tests are carried out in the same operating condition; this is done to taking into account the normal fluctuations of the system response and perform an investigation that gives statistically significant results.

Factorial design was chosen as approach to carry out the screening and the comparison in HAT-DOE; this was to understand which factors to prioritise in the process of screening and the level of influence of each factor in the phase of comparison. The selected parameters can be found in Chapter 5 in section 5.2 and the expanded test matrix for free running Pelton (FRP) are available in APPENDIX B: Test matrix free running Pelton experimental campaign. Using statistical tools, comparing the pressure and volume of oil as main factors, the half-normal analysis identified the pressure as main factor influencing the Pelton rotational speed (resulting in a $p\text{-value} < 0.05$), where the oil charging volume of the accumulator was two order of magnitude less important.

3.4 Shakedown

Before operating at full capacity, a shakedown was carried out. This involved:

- Bearing oil line testing
- Hydraulic lock investigation
- Labview sensor testing

A focus is given on the hydraulic lock as it was identified as critical issue for the correct working of the original prototype and one of the main reason of improvement of the original casing.

3.4.1 Bearing line shakedown

A bearing oil line was needed to test the original prototype which was supplied with plain bearings. Bearing line was tested during the shakedown phase to ensure the right amount of oil was delivered to the plain bearings ensuring lubrication and reducing friction losses. Two different pumps were used

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delivering pressurised oil needs at 2-5.5 bar. Assuming the flow of lubricant from a pressure-fed bearing having a central groove, journal bearing flowrate Q_s is calculated in equation 3.1 [167]:

$$Q_s = \frac{\pi p_s r_b c^3}{3\mu l'} (1 + 1.5\epsilon_b^2) \quad \left[\frac{m^3}{s}\right] \quad 3.1$$

Parameter	Description	Value/considerations
p_s	Fed-pressure	4 bar
c	Radial clearance at neutral position	37-45 μm (*)
μ_{oil}	Oil dynamic viscosity	0.1 kg/m*s (**)
r_b	Bearing radius	8.75-9.75 mm (***)
l'	Length of the half bearing	3.5 mm
ϵ_b	Eccentricity	0

Table 3.2 journal bearings parameters

(*) Typically, minimum oil film thicknesses in the load zone during operation ranges from 1.0 to 300 microns, but values of 5 to 75 microns are more common in midsized industrial equipment. The film thickness will be greater in equipment which has a larger diameter shaft [168].

(**) Oil dynamic viscosity depends on the temperature according to Vogel equation [169].

(***) Geometrical parameter are deducted from oil datasheet and from turbocharger drawings (HTA-003).

There are some limitations to equation 3.1 for the bearing flow rate Q_s :

1. Shaft speed: speed variation could result in a different flow rate because of variation of eccentricity ϵ_b [170] which is instead considered at regime speed;
2. Clearance: c was not available from the drawings and it was assumed 37 μm for the compressor bearing and 45 μm for the turbine bearing considering the basic guidelines used for journal bearing diametral clearance. Especially values for journal bearings are usually between 5 and 75 μm , or in other words clearance is 1.5 mm-per-m of journal diameter [170].
3. Type of bearings: in the configuration there is also a thrust bearing which are not considered, thus the real flow rate is expected to be slightly higher.

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The tests were carried out using 5W20 oil, the assembly rotation was provided using a compressed air injection in the turbine inlet. An electric pump was chosen as bearing pump because after firsts preliminary tests gear bearing pump was leaking and not suitable. The oil pressure reducing valve worked correctly and at maximum speed pump delivered 4 bar. Preliminary tests revealed a leakage, components were then further sealed. The experimental flow rate (0.102 L/min) was in the same order of magnitude but slightly higher than the theoretical one (0.089 L/min). This is justified considering that the flow rate is provided also for the thrust bearing which is not considered in the equation.

3.4.2 Hydraulic lock considerations

Scavenging tests and theoretical investigations were carried out to assess the correct working of the scavenge pump and the possible difficult scenarios in which the gear pump on the rig needed to ensure all the oil was drained from Pelton casing. The reason for these studies and for these preliminary investigations was to identify and avoid the phenomenon of cavitation in pumps.

Cavitation is defined as the formation of vapour bubbles in the region of low pressure within the flow field of a liquid; this happens when the minimum pressure in the flow reaches the vapor pressure [171]. It can cause damage to material surfaces close to the area where bubbles collapse when they are convected into regions of higher pressure, but the main disadvantage caused in pumps is the degradation of their performance. The dynamic response of the flow is compromised as well as the potential flow rate that the pump is capable to drive at a certain speed. To study and characterise the phenomenon of cavitation some parameters are required. It is possible to define a pressure parameter C_p shown in equation 3.2

$$C_p = \frac{(p_{st} - p_{1p})}{\frac{1}{2} \rho_{oil} U_p^2} \quad 3.2$$

Where p_{st} is the static pressure [Pa], p_{1p} is a reference static pressure of the pump (pump inlet) [Pa] and U_p is a reference velocity (inlet tip speed) [m/s]

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For an incompressible liquid with rigid boundaries, C_p is only a function of the geometry of the boundaries and of number of Reynolds Re . Furthermore, in absence of cavitation the fluid velocities and C_p are independent of the level of the pressure, this means that if the inlet pressure p_{1p} changes, all the other pressures change and the pressure coefficient would not be affected. As a consequence, there is a particular location at which the pressure p_{stmin} is at its minimum p_{stmin} and where a correspondent value of C_{pmin} can be found. If C_{pmin} can be determined experimentally or theoretically, it is possible to establish the value $p_{1p(cavitation)}$ at which the cavitation appears, assuming cavitation occurring at $p_v = p_{stmin}$. In equation 3.3 the value of C_{pmin} is intrinsically negative, thus a negative sign is applied.

$$p_{1p(cavitation)} = p_v + \frac{1}{2}\rho_{oil}U_p^2(-C_{pmin}) \quad 3.3$$

For a given pump and fluid temperature $p_{1p(cavitation)}$ is only function of U_p . The cavitation number σ is a dimensionless parameter used for evaluating the potential for cavitation and it is defined in equation 3.4. Every flow has a value σ_i called cavitation inception number (equation 3.5) at which cavitation occurs which corresponds to a particular value of inlet pressure p_{1s} .

$$\sigma = \frac{(p_{1p} - p_v)}{\frac{1}{2}\rho_{oil}U_p^2} \quad 3.4$$

$$\sigma_i = \frac{(p_{1p(cavitation)} - p_v)}{\frac{1}{2}\rho_{oil}U_p^2} \quad 3.5$$

If effectively cavitation occurs at $p_{min} = p_v$, combining equations 3.3 and 3.5 it results $\sigma_i = -C_{pmin}$; however often $p_{min} \neq p_v$.

3 Experimental Setup and Methodology

Regarding the practical aspect in pumps, a procedure is carried out to identify if there is cavitation occurs and the parameter which is used from manufacturers is the NPSH (Net Positive Suction Head) which is defined in equation 3.6

$$NPSH = \frac{(p_1^T - p_v)}{\rho_{oil}g} \quad 3.6$$

Where p_1^T is the inlet total pressure that is defined in equation 3.7 using the properties of a reference section.

$$p_1^T = p_1 + \frac{1}{2}\rho_{oil}v_1^2 \quad 3.7$$

For every pump the performance curves are provided and they give an idea of the flow delivered against the speed. An example is shown in Figure 3.4 where the coloured lines identify the Discharge pressure DP. The differential pressure DIP can be defined as per equation 3.8:

$$DIP = \text{discharge pressure (DP)} - \text{suction pressure (NPSH)} \quad 3.8$$

$$DP = \text{static discharge pressure (SDP)} + \text{friction losses} \quad 3.9$$

$$SDP = \rho g H_t \quad 3.10$$

H_t is the head between the outlet of the pump and the inlet of the delivery tank. Discharge pressure generally is greater than suction pressure thus differential pressure is positive. However, if pressure losses in discharge line are negligible respect to suction side and to pumping power, differential pressure could be 0 or even negative, this means that the pump does not see discharge line as a problematic in pumping the oil.

3 Experimental Setup and Methodology

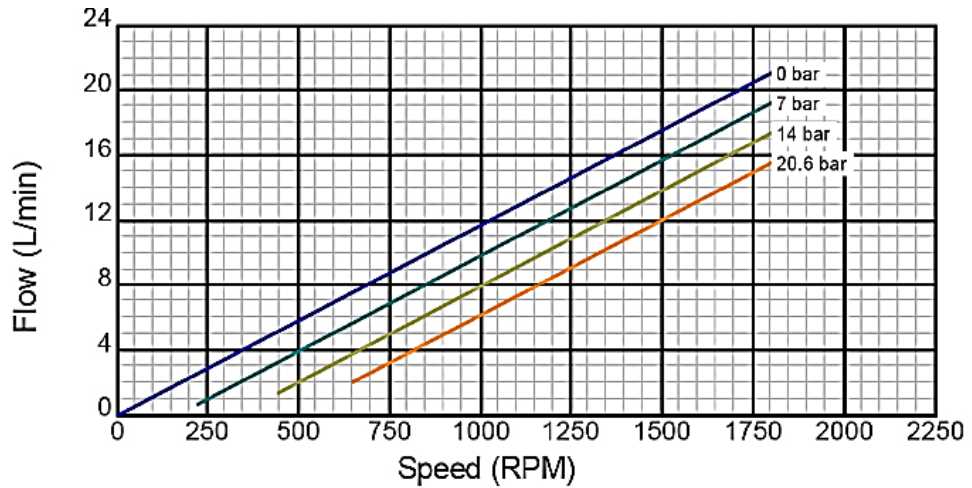


Figure 3.4 Typical Performance curve of a gear scavenge pump (Tuthill catalog)

A study was carried out to determine which flow rate the draining pump would have been able to accommodate without causing a hydraulic lock [172]. An Internal gear pump can create a vacuum to capture the oil due to the expanding volumes at the suction, however every pump requires a minimum amount of pressure energy in the liquid supplied to provide adequate performance thus avoiding cavitation. This pressure energy refers to Net Positive Suction Head (NPSH) and it solely a function of the system design on the suction side of the pump. To avoid cavitation NPSH available from the system must be greater than the NPSH required by the pump 3.11.

$$NPSH_{system} > NPSH_{pump} \quad 3.11$$

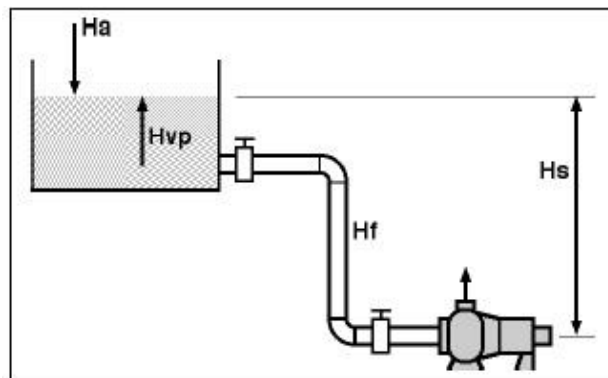


Figure 3.5 Graphic description of parameters involved in the hydraulic lock study

Considering Figure 3.5 NPSH available from the system is:

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$$NPSH_{system} = H_a + H_s - H_{vp} - H_f \quad 3.12$$

Where:

H_a = head on the surface of the liquid in the tank. In an open system like this, it would be atmospheric pressure (barometric pressure).

H_s = vertical distance of the free surface of the liquid above the centre-line of the pump impeller. If the liquid is below the pump, this becomes a negative value. (static suction pressure).

H_{vp} = vapour pressure of the liquid at the pumping temperature.

H_f = total friction losses in the suction piping.

The procedure to calculate if the pump can operate in the system without causing cavitation is the following:

1. Calculate $NPSH_{system}$.
2. Determine the DIP to identify the operating performance curve.
3. Select the flowrate and DP of the system and the corresponding speed.
4. Compare $NPSH_{system}$ and $NPSH_{pump}$ at that speed. If $NPSH_{system} < NPSH_{pump}$ cavitation occurs.
5. Repeat procedure changing system characteristics (i.e. pipes dimension) or flow rate.

Different scenarios were investigated; a total maximum flowrate was initially considered with 2x1.5mm nozzles at 300 bar with no loss and with additional flow rate required from journal bearing. In Table 3.3 relevant data on working condition are summarised. In these working conditions, NPSH system become negative, this means pressure losses are too high with the chosen flowrate. If suction pressure is too low or even negative in this case, it means the flowrate considered is too high for the system and particularly in this case the critical major pressure loss was represented by the drain hole and in its fitting (contraction 13-9 mm+ expansion 9 mm-pipes). The flowrate is such that the pump would create excessive vacuum in the suction side preventing the fluid flowing from the hole to the pipes; this vacuum from hole to pipes stops the flow

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and the results would be vacuum in the pipes and flooding of the turbocharger chamber. Decreasing flowrate, NPSH of system can be increased: through an optimization process, it was found out that a maximum of 39 L/min can be afforded by the pump avoiding suction in this condition; new parameters are shown in Table 3.4. The pressure losses in the contraction and expansion due to the drain hole and the fitting are always the major losses sum together.

Flowrate [L/min]	D pipes [inch]	ρ [kg/m ³]	H_a	H_s	H_{vp}	H_f	NPSH _{system} [FT]
62	1/2	975	1 bar	670 mm (0.06 bar)	0	1.51 bar	<0

Table 3.3 NPSH original scenario parameters Tuthill 1012 pump for 1/2" pipes

Flowrate [L/min]	D pipes [inch]	ρ [kg/m ³]	H_a	H_s	H_{vp}	NPSH _{system} [FT]
39	1/2	975	1 bar	670 mm (0.06 bar)	0	6.63

Table 3.4 NPSH optimised parameters Tuthill 1012 pump for 1/2" pipes

Considering a Tuthill 1012, $NPSH_{pump} - NPSH_{system} > 0$ thus the pump can work in this condition safely avoiding cavitation. Changing pipes diameter to 5/8" is another strategy to avoid cavitation. The optimal flow rate and correspondent parameters are shown in Table 3.5. Increasing pipes diameter to 1" does not influence significantly the final results.

Flowrate [L/min]	D pipes [inch]	ρ [kg/m ³]	H_a	H_s	H_{vp}	H_f	NPSH _{system} [FT]
48	5/8	975	1 bar	670 mm (0.06 bar)	0	0.87 bar	6.2

Table 3.5 NPSH optimised parameters for Tuthill 1012 pump with 5/8" pipes

In conclusion a Tuthill 1012 with 5/8" pipes in the suction line was identified as the best solution to afford a maximum flowrate of 48 L/min allowed according

to $NPSH_{system}$. Summarizing the conclusions for the hydraulic lock study were the following:

- Considering a 62 L/min as maximum overall flowrate to drain, due to nature of the system in the OTP, NPSH of the system would be always less than zero meaning that pressure losses experienced on the pipeline are highly detrimental in terms of hydraulic lock.
- The selected pump Tuthill 1012 can afford maximum 39 L/min if ½” pipes are used in the suction side and 48 L/min if 5/8” pipes are used.
- Major local pressure losses occur in the ball valves and in the flow through the fitting for the drain until entering the pipes. Changing the design of the oil drain might help in larger flowrate to afford.

3.4.3 Labview sensor test

A Labview VI was configured to acquire and monitor physical data from the rig as pressure and turbine speed. The main sensors were tested to verify the DAQ response. All the sensors were pre-calibrated, however, to verify that the calibration curves were correctly read from the Labview program, the sensors calibration was tested. Pressure transducers calibration was verified through a Deadweight hydraulic test and the calibration process confirmed the linear relation between voltage and pressure. A Deadweight tester (Figure 3.6) uses dead weights in the determination of pressures operating in a closed and compressed fluid system. For each value of pressure to be exerted in the closed hydraulic system, a predetermined amount of weights is used to exert the force to counterbalance the buoyancy forces. Each weight is stamped with the equivalent amount of force that will be exerted on a predetermined area of piston cylinder arrangement by using that weight. The system then, generates a force over a specific area, weights are placed on a piston and exerts a force on a fluid in the cylinder, in this case, an inert oil. The fluid is pressurised under the piston and at equilibrium, the pressure of the fluid is equal to the force divided by the area of the piston. Force is however affected by gravity and buoyancy and area is affected by the temperature of the piston and the pressure surrounded by the piston.

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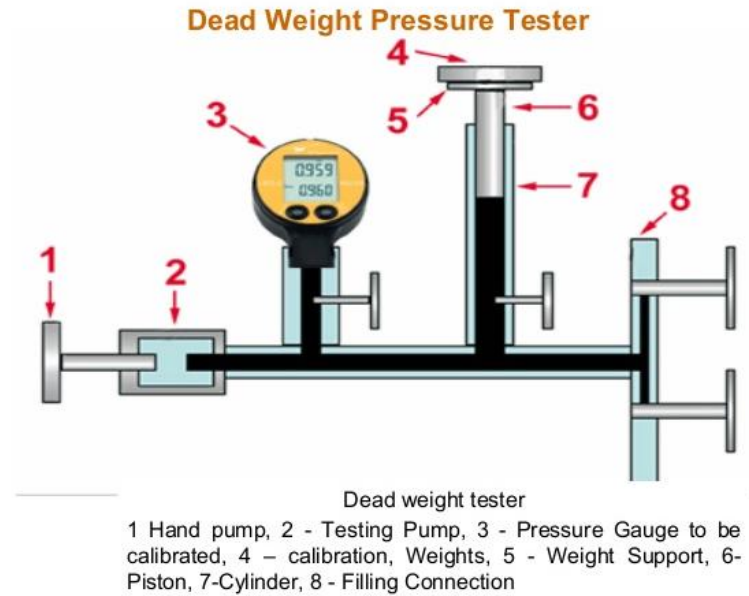


Figure 3.6 Deadweight test schematic [173]

After bleeding the system from air, the pressure transducer was screwed into the adaptor on the right of the pictures. The pressure transducer was then connected to DAQ USB-6211 to acquire the signal, and to the power supply which provided the 24 V required for the sensor excitation. Weights were located on the top of the weight support on the piston; these weights can provide two different pressures depending on the level which the hand pump is pushed in (i.e. blue line 10 bar, red line 100 bar). When the desired pressure in term of weight was reached, the hand pump was rotated until the lower (blue) or upper (red) level was reached. Several pressures have been recorded and their relation with the voltage shown in the DAQ are plotted in Figure 3.7 and the calibration curve was confirmed to be trustful.

The speed sensor signal was tested using a demo mode (mode 8) that was used for verifying the correct speed reading. The speed was confirmed to be matched with the voltage value expected reading 100,000 rpm with a given voltage of 2.5V.

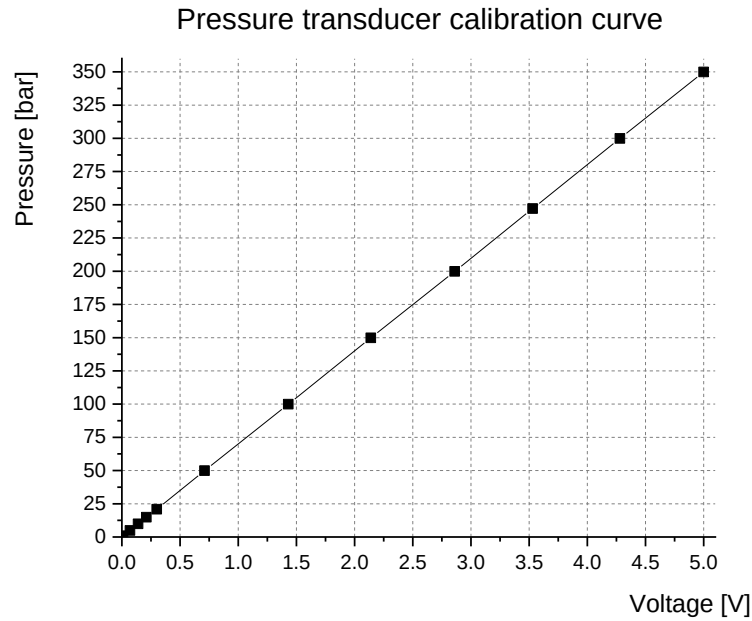


Figure 3.7 Calibration curve pressure transducer. The standard deviation for the pressure was ± 0.11 bar on average

3.5 Summary

The methodology chapter aims to show the experimental rig design logic and to assess potential constraints of the experiments. The main outcomes are listed below.

- The chapter presents an innovative methodology to test the hydraulic assist of a HAT. The detailed features are presented for the first time in literature.
- A Hydraulic-Assisted turbocharger prototype can be tested using an oil hydraulic accumulator energy source with 280 bar maximum output pressure, Nitrogen and a system of manual and solenoid valves monitoring oil pressure with piezoelectric sensors and assembly rotational speed with eddy-current sensors.
- In LabView software the modularity approach with 100Hz acquisition rate is the most reliable strategy to acquire and process experimental data compared to using the Express Virtual Instruments tools.

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- The Design of Experiments is a powerful tool to ensure reliability of experimental results. Each test-configuration needs to be tested at least 3 times to guarantee that 99.7% of the results lie inside a statistic normal distribution.
- Statical analysis reveals that when high-pressurised oil is injected from an accumulator into the buckets of a micro-hydraulic Pelton the oil-charging volume, compared to oil charging pressure, does not cause significant variation in maximum rotational speed.
- Hydraulic lock is a constraint in an encased-micro-Pelton design for high-head applications; theoretical investigations show that in a chamber volume of $8.4\text{E-}5 \text{ m}^3$ with a drain hole of diameter 13mm, the maximum accepted flow rate using a scavenge pump of 1kW is 48 L/min is 5/8" pipes are used, however effects on the wheel rotation are not taken in account.

4 Computational Model

4.1 Introduction

The computational model used to emulate KERB experimental setup in Chapter 3 is described in this chapter. This model is bespoke for the application as in literature no model was available linking a hydraulic accumulator discharge with the variation of rotational speed in a Pelton turbine. MatLab and Simulink were used to design customised virtual components rather than the ready-to-use blocks in the Simulink library, tailoring the programme for the specific application.

The mathematical description of fluid and mechanical systems is briefly discussed, followed by an introduction to ordinary differential equations and numerical methods for solving them.

The differential equation of the pressure trend, which includes the typical friction and localised pressure losses throughout the pipeline, is then determined using the accumulator modelling.

The Pelton dynamic modelling is shown including bearing friction and windage losses. The final three equations used in the modelling for the windage losses are chosen based on the literature review and the suitability for the configurations experimentally investigated.

Finally, the Simulink simulation strategy is explained. The simulation's inputs and outputs are defined, as well as the timestep and simulation type.

Experiments at various pressures and speeds were used to validate the model; the findings are available in Chapter 5 and Chapter 7.

4.2 Mathematical Representation of Fluid and Mechanical Systems

Fluid and mechanical systems are described in terms of the evolution of state variables for defined system properties and given initial conditions using mathematical equations. The vector of state variables $\mathbf{x}(t)$ is defined in equation 4.1 and it describes n properties of interest in the system, evolving for times $t \geq 0$ and starting from a given initial state $\mathbf{x}(0) = \mathbf{x}_0$; the governing mathematical problem can be written in the form of equation 4.2 [174]:

$$\mathbf{x}(t) = (x_1(t), x_2(t), \dots, x_n(t)) \in R^n \quad 4.1$$

$$\frac{d\mathbf{x}}{dt} = f(\mathbf{x}, t), \quad \mathbf{x}(0) = \mathbf{x}_0 \quad 4.2$$

From a mathematical point of view considering the vector $\mathbf{x}$ as a function $\mathbf{x}$ and considering its function $f(\mathbf{x})$ in an interval of time $I: a < t < b$, equation 4.2 is defined as an Ordinary Differential Equation (ODE) and involves the variable $\mathbf{x}$ the function $f(\mathbf{x})$ and one or more of its derivatives [175]. The order of an ODE is defined as the maximum number of derivatives of the function. To solve an ODE of the 1st order the initial condition needs to be specified and this is called the initial value problem (IVP). The numerical methods to solve an IVP [176] are divided in *Explicit methods* in which the current approximation depends on the previously determined approximate values and *Implicit methods* where the current approximation depends on both sides of the equation. The Euler method is the most fundamental explicit approach for numerical integration of ODEs in the Runge-Kutta method family, which includes both implicit and explicit methods [176]. Having the function y this method allows to obtain an approximation to the solution y_{n+1} at various values in a chosen interval; having a step size h , setting $t_n = t_0 + nh$, one step in the Euler method from t_n to $t_{n+1} = t_n + h$ is defined in equation 4.3 where y_n is an approximation of the solution of the ODE at the time t_n : $y_n \approx y(t_n)$.

$$y_{n+1} = y_n + hf(t_n, y_n) \quad 4.3$$

4.3 Accumulator modelling

4.3.1 Accumulator Discharge

In the experimental rig for testing the HAT the hydraulic accumulator is one of the Pelton oil jet's potential energy sources. When the accumulator is charged, the pressure of Nitrogen inside the bladder accumulator equalises the oil's starting pressure. The study of the pressure trend enables the identification of the dynamic change in velocity of the nozzle jet. Bladder accumulators are usually

pre-charged and subsequently the oil is accumulated from the hydraulic circuit; the temporal variation of the accumulator pressure has been modelled by readapting the set of equations given in [177]. The process in [177] is described for the gas discharge of the accumulator whereas in KERB application the investigation is focused on the oil discharge relating then the differential equation to its properties. Furthermore in the final equation the discharge area is the nozzles area, instead of the gas port. In KERB application indeed, the oil is charged first in the hydraulic circuit, then the hydraulic circuit is isolated and the bladder is charged with Nitrogen. Although the accumulator initially has the highest charging pressure, the pressure of the oil ejected drops immediately after the nozzles are opened, as the expansion process occurs inside the accumulator bladder. The expansion process was assumed to be reversible and adiabatic with an isentropic index $\gamma=1.4$. The charging and discharging steps of the accumulator are shown in Figure 4.1. The stages are:

- *Stage 1 - Accumulator empty*: the bladder is empty at ambient pressure and the oil poppet valve at the bottom is open.
- *Stage 2 – Oil charge*: accumulator is charged with oil from the poppet valve, the bladder is at his minimum volume while all or part of the available volume is occupied by oil.
- *Stage 3 - N_2 charging*: the hydraulic system is isolated, the oil flowrate $\dot{v}=0$ and all valves in the hydraulic circuit are closed. The accumulator is charged with N_2 and the initial charging pressure is set.
- *Stage 4 – Accumulator discharging*: the nozzles are opened, the oil flowrate $\dot{v}>0$ and the N_2 gas undergoes an adiabatic expansion.

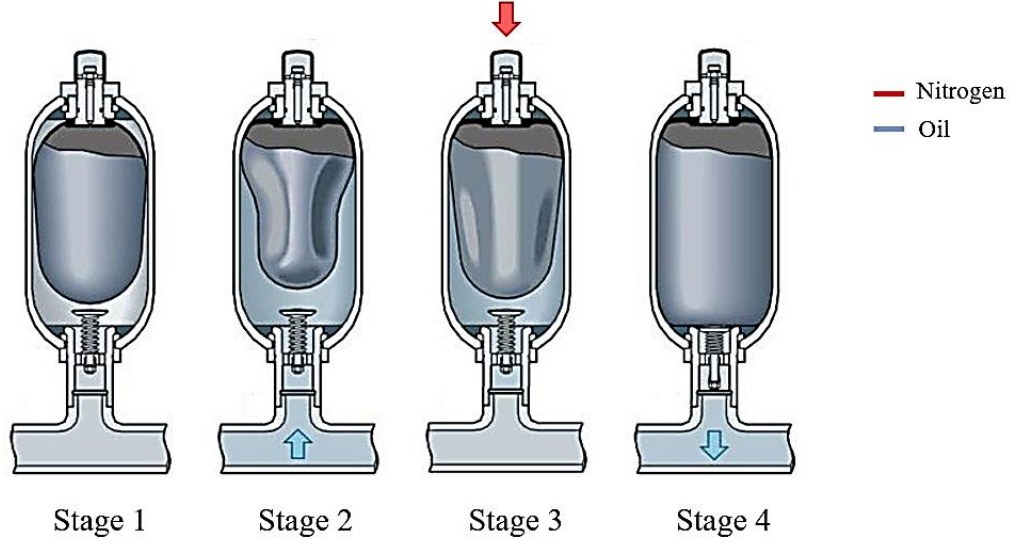


Figure 4.1 Stages of charging of a hydraulic accumulator in KERB (modified from [178])

Equation 4.4 describes the adiabatic process the N_2 undergoes inside the bladder during stage 4 considering two consecutive states of the process.

$$p_1 V_1^\gamma = p_2 V_2^\gamma \quad 4.4$$

Where p_1 [Pa] is the initial pressure, V_1 [m³] is the initial volume, p_2 [Pa] is the final pressure, V_2 [m³] is the final volume of N_2 in the bladder and γ is the adiabatic expansion coefficient. Considering two consecutive states of the expansion, to calculate the final pressure, the final volume is required; this is equal to the initial volume plus the product of the volumetric oil flow rate $\dot{v}$ at the nozzles and the selected timestep Δt of the expansion (equation 4.5 and 4.6).

$$V_2 = V_1 + V_t \quad 4.5$$

$$V_t = \dot{v} * \Delta t \quad 4.6$$

The flow rate is computed at each timestep; the initial flow rate depends on the oil jet velocity v_{jet} at the nozzle. The oil inside the accumulator has the same initial charging pressure p_1 of N_2 and Bernoulli's equation in 4.7 governs conservation of mass and energy and the sum of energy in section 1 (the accumulator) is equal to the sum of energy in section 2 (the nozzles outlet).

4 Computational Model

$$p_1 + \frac{1}{2}\rho_{oil}u_{1acc}^2 + \rho_{oil}gz_1 = p_{2amb} + \frac{1}{2}\rho v_{jet}^2 + \rho_{oil}gz_2 + p_{loss} \quad 4.7$$

Where ρ_{oil} [kg/m³] is the oil density at test temperature, u_{1acc} is the oil velocity at the accumulator [m/s], g [m/s²] is the gravity acceleration, z_1 [m] is the elevation of the accumulator above a reference plane with the direction opposite to g , p_{2amb} [Pa] is the ambient pressure, v_{jet} [m/s] is the oil jet velocity at the nozzle, z_2 [m] is the elevation of the nozzles above a reference plane with the direction opposite to g and p_{loss} [Pa] is the sum of all the pressure losses. Assuming $u_{1acc} = 0$, and $z_1 = z_2$ equation 4.8 describes v_{jet} . Assuming the fluid is incompressible, the oil flow rate $\dot{v}$ is influenced by the ratio between the theoretical mass flow rate and the actual mass flow rate which is called the discharge coefficient C_D . The discharge coefficient is defined in equation 4.9 and 4.10 where C_c is the contraction coefficient (the ratio between the jet area and the orifice area) and C_v is the velocity coefficient (the ratio between the actual velocity and the theoretical velocity). For a straight tip nozzle; $C_c = 1.0$ and C_v depend on the nozzle finishing quality [179]. Oil flow rate $\dot{v}$ is given in equation 4.11.

$$v_{jet} = \sqrt{\frac{2 \times [p_1 - p_{2amb} - p_{loss}]}{\rho_{oil}}} \quad 4.8$$

$$C_D = \frac{Q_{Actual}}{Q_{Theoretical}} \quad 4.9$$

$$\therefore C_D = C_c C_v \quad 4.10$$

$$\dot{v} = N_n C_D v_{jet} A \quad 4.11$$

Where D_n [m] is the nozzle diameter, N_n is the number of nozzles, $A = \pi \left(\frac{D_n}{2}\right)^2$ [m²] is the exit area of the nozzle. Combining equation 4.6, 4.8, 4.9, 4.11, V_2 can be summarised in equation 4.12 and p_2 in equation 4.13.

$$V_2 = V_1 + \sqrt{\frac{2 \times [p_1 - p_{2amb} - p_{loss}]}{\rho_{oil}}} * N_n * A * C_D * \Delta t \quad 4.12$$

$$p_2 = \frac{p_1 V_1^\gamma}{\left(V_1 + \sqrt{\frac{2 \times [p_1 - p_{2amb} - p_{loss}]}{\rho_{oil}}} * N_n * A * C_D * \Delta t \right)^\gamma} \quad 4.13$$

The pressure p_i at the conclusion of each timestep can be determined iteratively by approaching the adiabatic expansion as a continuous mathematical problem. The flowrate $\dot{v}_i$ varies depending on the pressure, as the velocity at the nozzles; Since the initial and final volumes change, equation 4.14 describes the pressure at the conclusion of each timestep. The differential equation that regulates the problem and serves as the calculation base for the Simulink model is developed and generalised. Considering the adiabatic expansion equation 4.15 And defining $C_{1p} = C^\frac{1}{\gamma}$ we obtain 4.16.

$$p_{1+i} = p_i \left(\frac{V_i}{V_{1+i}} \right)^\gamma = p_i \left(\frac{V_i}{V_i + \dot{v} dt} \right)^\gamma \quad 4.14$$

$$pV^\gamma = C \quad 4.15$$

$$V = \frac{C_{1p}}{p^\frac{1}{\gamma}} \quad 4.16$$

Combining the two definition for the volumetric flow rate in equation system 4.17, the derived volume V in equation 4.16 and Bernoulli equation in 4.8, highlighting v_{jet} , equation 4.18 is obtained. Defining $C_{2p} = N_n A C_D \sqrt{\frac{2}{\rho_{oil}}}$, the differential equation 4.19 is derived. C_{1p} and C_{2p} groups constant values to simplify the written final equation.

4 Computational Model

$$\dot{v} = \begin{cases} N_n C_D v_{jet} A \Rightarrow N_n A C_D \sqrt{\frac{2\Delta p}{\rho_{oil}}} \\ \frac{dV}{dt} \Rightarrow -\frac{C_{1p}}{\gamma p^{\frac{1}{\gamma}+1}} \frac{dp}{dt} \end{cases} \quad 4.17$$

$$-\frac{C_{1p}}{\gamma p^{\left(\frac{1}{\gamma}+1\right)}} \frac{dp}{dt} = C_{2p} \sqrt{p} \quad 4.18$$

$$\frac{dp(t)}{dt} + \frac{C_{2p}}{C_{1p}\gamma} p(t)^{\frac{2+3\gamma}{2\gamma}} = 0 \quad 4.19$$

Considering the analytical resolution, equation 4.19 is rearranged and integrated over time obtaining 4.22. Constant C_3 is derived setting the initial condition in equation 4.23 and obtaining 4.24.

$$\int -\frac{dp}{p^{\left(\frac{1}{\gamma}+\frac{3}{2}\right)}} = \int \frac{\gamma C_{2p}}{C_1} dt \quad 4.20$$

$$-\frac{1}{-\left(\frac{1}{\gamma}+\frac{1}{2}\right)} p^{-\left(\frac{1}{\gamma}+\frac{1}{2}\right)} = \frac{\gamma C_{2p}}{C_{1p}} t + C_3 \quad 4.21$$

$$\frac{2\gamma}{2+\gamma} p^{-\frac{2+\gamma}{2\gamma}} = \frac{\gamma C_{2p}}{C_{1p}} t + C_3 \quad 4.22$$

$$p(0) = p_0 \quad 4.23$$

$$C_3 = \left(\frac{2\gamma}{2+\gamma}\right) p_0^{-\frac{2+\gamma}{2\gamma}} \quad 4.24$$

Replacing 4.24 in 4.22 and rearranging we obtain equation 4.25. Defining $C_{pt} =$

$\left[\left(\frac{2+\gamma}{2}\right) \frac{C_{2p}}{C_{1p}}\right]^{-\frac{2+\gamma}{2\gamma}}$ equation 4.28, which express $p(t)$, can be derived.

$$\frac{2\gamma}{2+\gamma} p^{-\frac{2+\gamma}{2\gamma}} = \frac{\gamma C_{2p}}{C_{1p}} t + \left(\frac{2\gamma}{2+\gamma} \right) p_0^{-\frac{2+\gamma}{2\gamma}} \quad 4.25$$

$$p^{-\frac{2+\gamma}{2\gamma}} = \left(\frac{2+\gamma}{2} \right) \frac{C_{2p}}{C_{1p}} t + p_0^{-\frac{2+\gamma}{2\gamma}} \quad 4.26$$

$$p = \left[\left(\frac{2+\gamma}{2} \right) \frac{C_{2p}}{C_{1p}} t \right]^{-\frac{2+\gamma}{2\gamma}} + p_0 \quad 4.27$$

$$p(t) = C_{pt} t^{-\frac{2+\gamma}{2\gamma}} + p_0 \quad 4.28$$

Equation 4.28 includes pressure losses terms listed in section 4.3.2. The Simulink model includes a MatLab script that computes the pressure losses in each timestep. The detailed approach of Simulink solving method is described in section 4.7.1.

4.3.2 Pressure losses

In KERB experimental rig the oil pumped through the fluid power system is subject to an energy loss due to the frictions along the pipe. The analysis in section 4.3.1 includes these pressure losses p_{loss} along the hydraulic circuit from the accumulator outlet to the nozzle exit. Pressure losses in the pipeline influence the adiabatic expansion, thus the final oil jet speed at the nozzles. Pressure losses can be classified in *Friction pressure losses* Δp_f that are caused by the friction of the oil that flows through the pipes and *Local pressure losses* due to obstacles in the hydraulic circuit such as expansions, contractions, valves, elbow etc. Friction pressure losses can be calculated through equation 4.29 [180] [181]:

$$\Delta p_f = \rho_{oil} \frac{fL}{D} \frac{u^2}{2} \quad [Pa] \quad 4.29$$

4 Computational Model

Where L is the length of the pipeline [m], D is the diameter of the pipe [m], u is the velocity in the pipe [m/s] and f is the non-dimensional friction factor calculated depending on Reynolds number Re . The velocity in the pipe u is calculated in equation 4.30 and Reynolds number is calculated in equation 4.31.

$$u = \frac{\dot{v}}{A_p} \quad 4.30$$

$$Re = \frac{u * D}{\nu} \quad 4.31$$

Where A_p is the pipes cross section [m²] and $\nu = \frac{\mu_{oil}}{\rho_{oil}}$ is the kinematic viscosity of oil [m²/s]. Equations 4.32 and 4.33 can be used to calculate the frictional factor f according to the flow regime inside the pipe respectively for laminar and turbulent flow. The thresholds for the regimes are: Laminar flow for $Re < 2300$, transition region for $2300 < Re < 3500$ and turbulent flow for $Re > 3500$

$$f = \frac{64}{Re} \quad 4.32$$

$$f = \frac{0.316}{Re^{1/4}} \quad 4.33$$

Local pressure losses vary depending on the type of obstacle [180][182]. In Table 4.1 the types of obstacles in KERB are summarised; pressure losses occur in the high-pressure section and they are focused in 1/2" pipeline, 1/4" pipes and inside the nozzle. In Table 4.2 the equations used for the local pressure losses are listed with the correspondent reference.

4 Computational Model

Section	Obstacles
½” pipeline	Sudden contraction accumulator - ½” pipes 2x Slow radius bend 90° 2x Elbow 90° Ball valve Tee junction
¼” pipes	Ball valve Reduction ½” - ¼” pipes Slow bend
Nozzles	Gradual contraction ¼” -3.5mm Friction pressure losses in nozzle length

Table 4.1 Local pressure losses in KERB rig

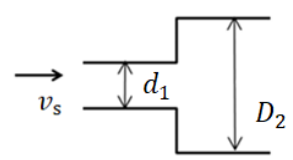
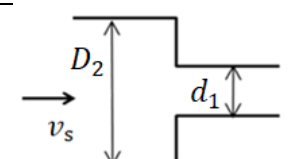
Density and viscosity are strongly dependent on temperature. To include this effect in the pressure losses, the equation 4.34 that governs the changing of viscosity according to temperature was used where T_1 is test temperature, ρ_{1oil} is the density at T_1 , β is the volumetric expansion coefficient [0.0007 1/K], ρ_{0oil} is the density at 15 °C [kgm⁻³], T_0 is reference temperature [15° C].

$$\rho_{1oil} = \frac{\rho_{0oil}}{1 + \beta(T_1 - T_0)} \quad 4.34$$

To include the changing of viscosity with temperature Vogel equation 4.35 [169] has been used where μ_{oil} is dynamic viscosity at temperature T, k , θ_1 and θ_2 are Vogel coefficient and in this case are $k=0.078$ mPas, $\theta_1=1180$, $\theta_2=133$.

$$\mu_{oil} = k * e^{\left(\frac{\theta_1}{\theta_2 + T}\right)} \quad 4.35$$

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Pressure loss type	Equation	Reference	Notes/schematic
Valves	$\Delta p_v = \frac{(3600\dot{v})^2}{K_v^2} [bar] \quad 4.36$ $K_v = 0.864C_{vv} \quad 4.37$	[183]	K_v or C_{vv} are coefficients given by the supplier.
Elbow, slow bend, tee	$\Delta p_l = K \rho_{oil} \frac{v_s^2}{2} [Pa] \quad 4.38$	[183]	Elbow $K=0.9$ [181] Slow bend $K=0.14$ Tee $K=1.8$
Sudden Expansion	$\Delta p_{se} = \rho_{oil} \frac{v_s^2}{2} \left(1 - \frac{A_1}{A_2}\right)^2 \quad 4.39$	[184]	
Sudden Contraction	$\Delta p_{sc} = \rho_{oil} \frac{v_s^2}{2} K_{sc} \quad 4.40$	If $\frac{A_1}{A_2} > 0.76$ equation for sudden expansion can be used, otherwise equation should be used [181]	

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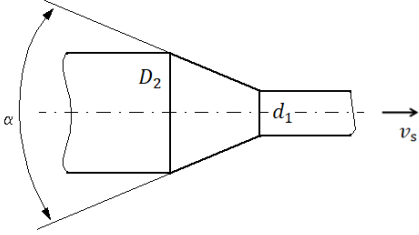
Gradual contraction	$\Delta p_{gc} = \rho_{oil} \frac{K_{GC} v_s^2}{2} \quad 4.41$ $K_{GC} = \begin{cases} K_{cor} * 0.5 * \left(1 - \frac{A_1}{A_2}\right)^{0.75} * 1.6 * \sin \frac{\alpha}{2} & \text{for } 0 < \alpha \leq 45^\circ \\ K_{cor} * 0.5 * \left(1 - \frac{A_1}{A_2}\right)^{0.75} * \sin \frac{\alpha}{2} & \text{for } 45^\circ < \alpha \leq 180^\circ \end{cases}$	[184]	
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Table 4.2 Local pressure losses equations for different obstacles

Coefficient K_{sc} in equation for sudden contraction can be calculated from tables available in the literature, depending on the area or diameters ratio.

K_{cor} in the coefficient formula for the gradual contraction is a correction factor (0.45) and α is the angle of tapered reduction in radiant.

d_1/D_2	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9
K_{sc}	0.5	0.48	0.46	0.42	0.38	0.32	0.26	0.18	0.1

Table 4.3 Coefficient K in local pressure losses equations depending on diameter ratio of the contraction[181]

An empirical formula for K_{sc} is also proposed in equation 4.42 [185]

$$K_{sc} = 0.42 * \left(1 - \frac{d_1^2}{D_2^2}\right) \quad 4.42$$

4.4 Pelton wheel dynamic modelling

The equation of the force exerted on the Pelton wheel by an oil jet is derived performing the conservation of momentum to a control volume around a Pelton bucket [162]. The terminology in Figure 4.2 follows Zhang [162] and it is then adapted for KERB application. The oil jet impact into the Pelton bucket with velocity C_0 which represents the jet speed calculated through Bernoulli formula. The bucket is assumed to have a constant translational speed U , the interaction between oil jet and the bucket is considered in the relative moving system. Therefore, for the oil flow at the bucket entry $C_1=C_0$ and the relative velocity between the jet and the moving bucket is $W_1=C_1-U$.

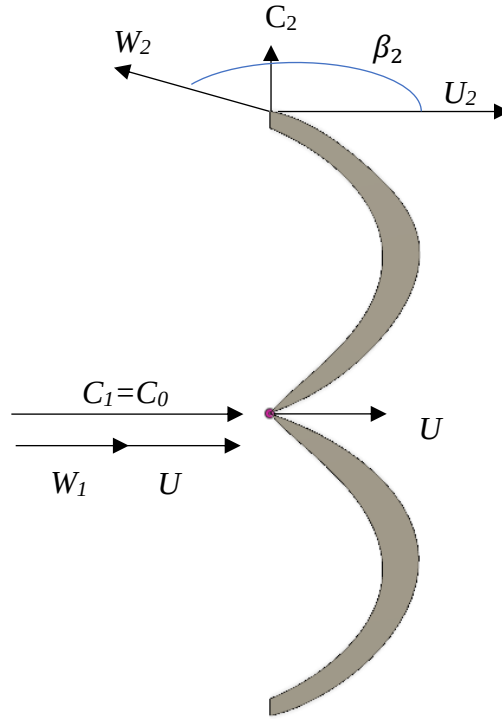


Figure 4.2 Flow interaction between oil jet and Pelton bucket

The jet velocity enters the vein at velocity W_1 and leaves the bucket with a velocity $W_2 = W_1 = W$ at an angle β_2 assuming the process ideal. According to exit triangle, the absolute velocity C_2 is in equation 4.43.

$$C_2^2 = U^2 + W^2 + 2UW\cos\beta_2 \quad 4.43$$

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The momentum flux difference between entry and exit of the moving bucket is evaluated in equation 4.44, to find the impulsive force acting on the bucket, where $\dot{m}_w$ [kg/s] is considered in the relative frame and to describe the flow rate $\dot{m}_c$ in the absolute frame equation 4.46 and 4.47 can be used.

$$F_p = \dot{m}_w W_1 - \dot{m}_w W_2 \cos \beta_2 \quad 4.44$$

$$F_p = \dot{m}_w W (1 - \cos \beta_2) \quad 4.45$$

$$\dot{m}_w = \rho W A \quad 4.46$$

$$\dot{m}_c = \rho C_1 A \quad 4.47$$

Equalizing equations for ρA 4.46 and 4.47 above we obtain 4.48:

$$\dot{m}_w = \frac{W}{C_1} \dot{m}_c \quad 4.48$$

Replacing 4.48 in 4.45 we obtain 4.49.

$$F_p = \dot{m}_c \frac{W^2}{C_1} (1 - \cos \beta_2) \quad 4.49$$

$$F_p = \frac{\rho_{oil} \dot{v} (C_1 - U)^2}{C_1} (1 - \cos \beta_2) \quad 4.50$$

Using terminology for KERB for which $C_1 = v_{jet}$ [m/s] and the relationship between the oil flow rate and the jet speed $\dot{v} = v_{jet} A C_D$ then equation 4.51 is obtained.

$$F_p = \rho_{oil} A C_D (v_{jet} - U)^2 (1 - \cos \beta_2) \quad 4.51$$

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The relationship between the ideal torque T_p and force F_p acting on the Pelton is shown in equation 4.52 and combining this with equation 4.51 we obtain the final equation 4.53 for T_p .

$$T_p = N_n F_p r \quad [Nm] \quad 4.52$$

$$T_p = N_n r \rho_{oil} A C_D (v_{jet} - U)^2 (1 - \cos \beta_2) \quad 4.53$$

Where r is the pitch circle radius of Pelton wheel [m] and N_n is the number of nozzles. T_p is also related to the angular acceleration through the equation 4.54

$$T_p = I \alpha_p \quad 4.54$$

Where I is the turbocharger inertia [kgm^2] and α_p is angular acceleration [rad/s^2]. The 3D model of the assembly is used to calculate Inertia values. Angular acceleration can then be calculated combining equations 4.53, 4.54 highlighting α_p obtaining equation 4.55. Angular speed can then be calculated integrating α_p in the performance time in equation 4.57.

$$\alpha_p = \frac{N_n r}{I} \rho_{oil} A C_D (v_{jet} - U)^2 (1 - \cos \beta_2) \quad 4.55$$

$$\therefore \alpha_p = \frac{d\omega}{dt} \quad 4.56$$

$$\omega = \int \alpha_p \delta t \quad 4.57$$

The differential equation of Pelton dynamic for calculating the ideal angular speed is therefore given in equation 4.58 combining the equations for the accumulator discharge modelling and the Pelton dynamic.

$$\frac{d\omega}{dt} = C_{pm} \left[\sqrt{\frac{2}{\rho_{oil}} \left(C_{pt} t^{-\frac{2\gamma}{2+\gamma}} + p_0 \right)} - \omega r \right]^2 \quad 4.58$$

Where $C_{pm} = \frac{N_{nr}}{I} \rho_{oil} A C_D (1 - \cos\beta_2)$. The pressure losses are computed in each timestep along the hydraulic circuit. The friction losses due to bearings and windage losses are separately computed and included in the model.

4.5 Bearings friction losses modelling

4.5.1 Bearing friction losses overview

Bearings are used to prevent metal to metal contact between two elements in relative motion reducing friction between the parts while supporting the rotational movement. The bearings can be classified according to the load they can support in *Thrust load bearings* where the axial load is supported, *Radial load bearings* where the radial load is supported and *Combined load bearings* where both radial and axial load are supported. The different geometries reflect their friction reduction strategy and the two most common type of bearings used in turbocharger technology are rolling element bearings and plain bearings.

- Rolling element bearings: these bearings have an inner and outer race that contain rolling elements inside such as cylinders or balls kept in place by a cage. The main load is transferred to the rolling elements, the friction is then reduced as the rotational movement faces a rotational resistance rather than a sliding resistance. The two main categories are *ball bearings* (radial, angular contact, etc.) and *roller bearings* (cylindrical, spherical, gear, tapered, needle). The advantages are a good wear resistance, low lubrication needed, low friction, long service life, easy replacement, relatively economic and that can handle both radial and thrust load. Disadvantages are the possibility of breaking due to shock, noise and that they cannot support large weights. These bearings were used in FRP configuration and in the HAT assembly.

- Plain/fluid bearings: this type of bearings does not have any additional element other than a sleeve mounted on the shaft and fit into the bore. The fluid types rely on pressurised gas or liquid to carry the load and reduce friction in machines that work at high speeds and loads. In *hydrostatic bearings* the lubricant is forced between the moving parts forming a wedge that keeps them apart, whereas in *hydrodynamic bearing* the journal motion sucks the fluid between the shaft and the housing. These bearings are almost inexpensive, compact, lightweight and able to support a high load. They however require a specific amount of continuous lubrication and they are subject to erosion. These bearings were mounted in the inherited unoptimised prototype.

Petroff equation 4.59 describes the basic lubrication theory which is then particularised for each bearing geometry [186]. The shear stress in a lubricant τ is equal to the product of the dynamic viscosity μ and the velocity gradient $\frac{du}{dy}$.

$$\tau = \mu \left(\frac{du}{dy} \right) \quad 4.59$$

Various regimes of lubrication can occur (Stribeck curve in Figure 4.3); the friction in fluid-lubricated contact (as in journal bearing) is function of a friction coefficient f (equation 4.60) and a dimensionless duty parameter called Hersey number (equation 4.61) which depends on dynamic viscosity μ of the lubricant, the rotational speed N and the normal load per length of the tribological contact σ_h .

$$f = \alpha_b f_s + (1 - \alpha_b) f_l \quad 4.60$$

$$\text{Hersey number} = \frac{\mu N}{\sigma_h} \quad 4.61$$

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α_b is the metal-to-metal contact constant (varying between 0 and 1), f_s is the metal-to-metal coefficient friction and f_l is the hydrodynamic coefficient of friction. The extreme regimes are:

- *Boundary (solid) friction*: $\alpha_b \rightarrow 1$ and $f \rightarrow f_s$
- *Hydrodynamic (viscous or thick film) friction*: $\alpha_b \rightarrow 0$ and $f \rightarrow f_l$

The transition between these two regimes is called *mixed lubrication regime*.

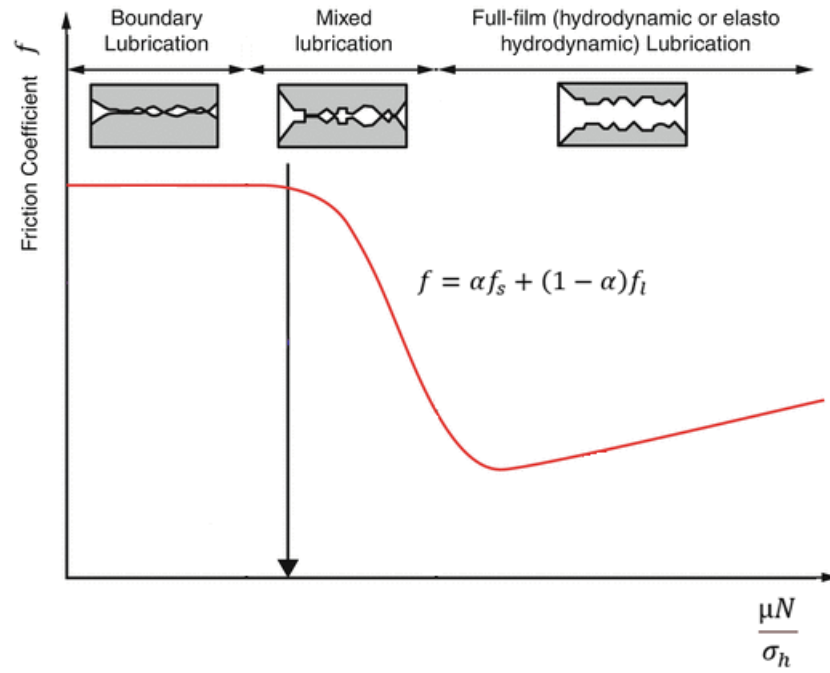


Figure 4.3 Stribeck diagram for fluid lubricated contacts (modified from [187])

4.5.2 Bearing friction losses modelling in KERB-HAT

During Pelton rotation, friction losses occur due to ball bearings. These can be calculated starting from Figure 4.4; the schematic of the shaft shows the *weights force* of the different components and the *oil jet force* acting on the assembly. The origin O is conventionally set on the bearing on the compressor side. Considering the forces applied to the mass centre of each body the final reaction forces F_{rb} , and F_{rs} are obtained solving the system of equations for the static equilibrium at the shaft in the system (equation 4.62). In the weight forces $F_{i=t,p,c}$ of different components, subscripts t,p,c are used respectively for “turbine”, “Pelton” and “compressor”. The force F_j of the oil jet depends on the

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oil mass flow rate $\dot{m}$ [$\frac{kg}{s}$] and jet speed v_{jet} [$\frac{m}{s}$]. M_i and M_j are the moments of the forces.

$$\begin{cases} \sum_{i=t,p,c} (F_i + F_j) = m_i g + \dot{m} v_{jet} = 0 \\ \sum_{i=t,p,c} (M_i + M_j) = 0 \end{cases} \quad 4.62$$

Characterizing the system for the turbocharger prototype shaft, we obtain 4.63 and 4.64 gives the solution for F_{rb} , and F_{rs} .

$$\begin{cases} F_{rb} + F_{rs} - F_t - F_p - F_j - F_c = 0 \\ F_{rb} * a_b - F_t * a_t - (F_p + F_j) * a_p - F_c * a_c = 0 \end{cases} \quad 4.63$$

$$\begin{cases} F_{rs} = F_t + F_p + F_j + F_c - \frac{F_t * a_t + (F_p + F_j) * a_p + F_c * a_c}{a_b} \\ F_{rb} = \frac{F_t * a_t + (F_p + F_j) * a_p + F_c * a_c}{a_b} \end{cases} \quad 4.64$$

Once calculated the acting forces of the system, the empirical approach of Palmgren [188] adapted by Harris [189] is used to evaluate rolling bearing friction torque T_{ft} [Nmm] in equation 4.65 which splits in two components T_l and T_v .

T_l in equation 4.66 is a resistant torque due to the bearing load which depends on the radial load of the specific bearing F_r , on the bearing medium diameter $d_m = \frac{d_{out} + d_{in}}{2}$ and on a factor f_1 described in 4.67 and function of the bearing design.

$$T_{ft} = T_l + T_v \quad 4.65$$

$$T_l = f_1 F_r d_m \quad 4.66$$

$$f_1 = z \left(\frac{F_s}{C_s} \right)^y \quad 4.67$$

C_s is the basic static load given from manufacturer's catalog along with data to enable calculation of the force F_s . The latter is the equivalent static bearing load required to compare the load on a non-rotating bearing with the basic static capacity.

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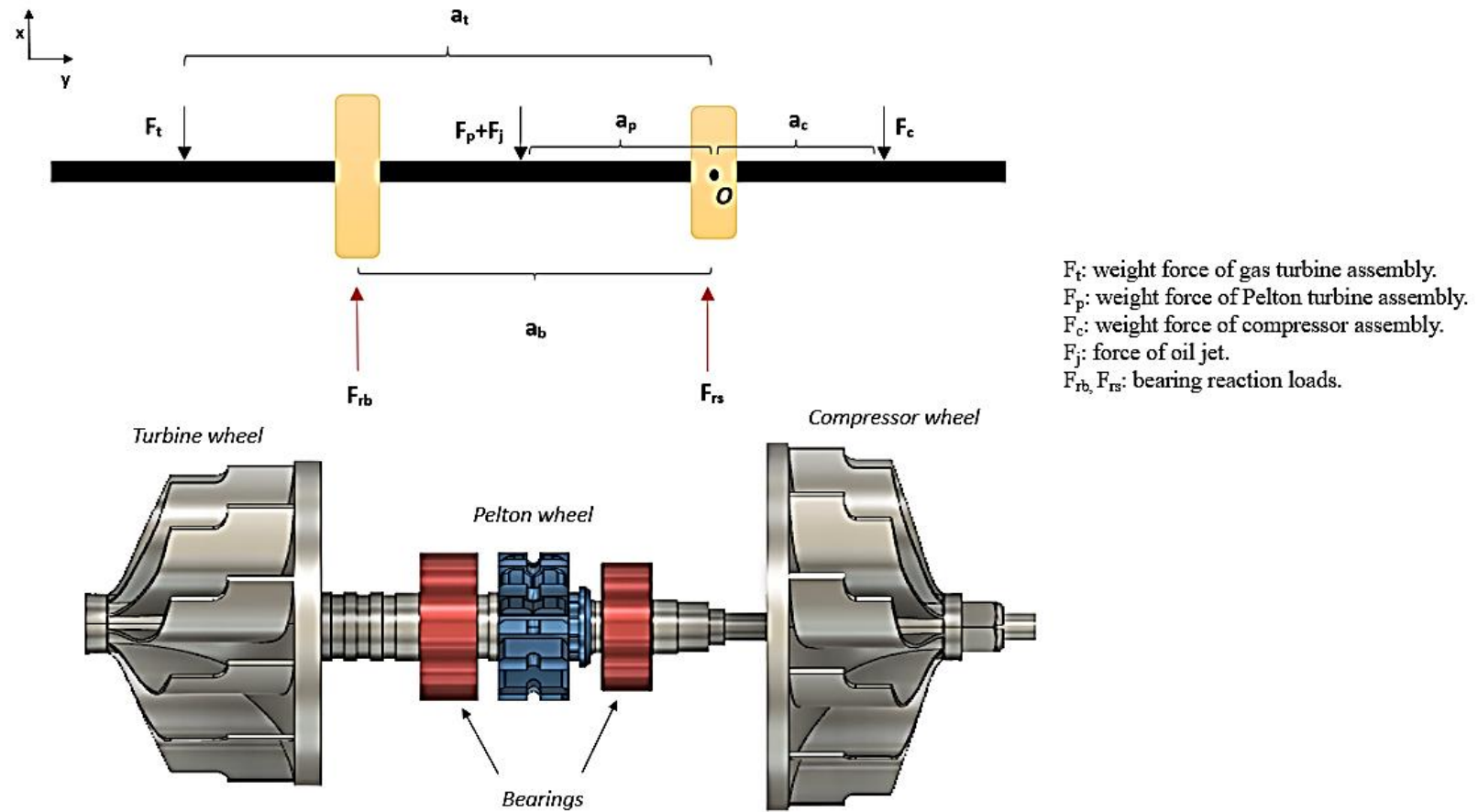


Figure 4.4 Schematic for bearing load calculation and turbocharger prototype reference.

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C_s is defined as the load applied to a non-rotating bearing that will result in a permanent deformation of $0.0001D$ at the weaker of the inner or outer raceways contacts occurring at the position of the maximum loaded rolling element [189] (D is defined as the balls diameter of the rolling elements between the two races of the bearings). F_s is given in equation 4.68 and it depends on the radial force F_r and on the axial force F_{ax}

$$F_s = X_s F_r + Y_s F_{ax} \quad 4.68$$

In KERB-FRP application $F_{ax} = 0$ obtaining then 4.69. According to [189] having $X_s = 0.5$ leads to have $F_s < F_r$; in this condition $F_s = F_r$. Equation 4.67 then becomes 4.70.

$$F_s = X_s F_r \quad 4.69$$

$$f_1 = z \left(\frac{F_r}{C_s} \right)^y \quad 4.70$$

Values for z and y are given depending on the bearing design type [189].

The other resistant torque T_v is due to the fluid friction of bearing element in contact with the lubricant and it depends on the viscous properties of the lubricant, the amount of oil employed and the bearing speed (equations 4.71 and 4.72).

$$T_v = 10^{-7} f_0 (vn)^{\frac{2}{3}} d_m^3 \text{ [Nmm]} \text{ if } vn \geq 2000 \quad 4.71$$

$$T_v = 160 \times 10^{-7} f_0 d_m^3 \text{ [Nmm]} \text{ if } vn \leq 2000 \quad 4.72$$

ν is the oil viscosity [cst], n is the rotational speed [rpm] and f_0 is a factor depending on the type of bearing and the method of lubrication. An oil mist is considered as lubrication method for selecting f_0 value because of the oil distribution when the nozzles injects it into the chamber. An example of geometrical and coefficient values for calculating T_l and T_v are selected from [189] and [190] and they are shown in table Table 4.4.

	$d_{out} [mm]$	$d_{in} [mm]$	z	y	C_s	f_0
12mm bearing	28	12	0.0004	0.55	2402	1.7
15mm bearing	32	15	0.0004	0.55	2824.6	1.7

Table 4.4 Values to calculate resistant torque components T_l and T_v in bearings friction

4.6 Windage losses modelling

The computational model calculates different configurations of the windage losses according to which rig is used:

- FRP (free running Pelton): in this configuration the Pelton can be considered without casing or encased.
- TPSR (Turbo Pelton Stand Rig): the Pelton is considered encased and there are additional windage losses from the compressor and the turbine wheel which are dragged by the Pelton rotation.

The different configuration can be simulated with the equations in sections 4.6.1, 4.6.2, 4.6.3.

4.6.1 Free running Pelton windage losses modelling

The FRP can have two configurations: not encased or encased. The first case will be discussed here, while the second configuration will be discussed in section 4.6.2. The approach described by Diab [144] is used to calculate the windage losses of FRP. Diab results show that the windage loss is directly proportional with the diameter of the gear and the width of its teeth. Diab explored windage losses for gear with ranges of pitch circle diameter 144-288mm with width of the teeth 30-60; Pelton wheel pitch circle diameter in the HAT was 28mm and the teeth were 10 mm long. The windage loss is considered as the sum of two contribute: a free rotating disc and its side teeth/buckets. The total windage loss resistance torque $T_{w,fp}$ in equation 4.73 is then represented as the sum of the resisting torque of a free running disc T_{wd} in equation 4.75 [147]

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using the torque coefficient C_m for a free disc in laminar flow adapted from equation 2.16 and the resisting torque of the Pelton buckets T_{wb} . T_{wb} is the windage torque exerted by gear teeth (Equation 4.74).

$$T_{w,fp} = T_{wd} + T_{wb} \quad 4.73$$

$$T_{wb} = Z_b \rho x b \omega^2 \left(R_a - \frac{x}{2} \right)^3 \quad 4.74$$

$$T_{wd} = \frac{1}{2} C_m \rho \omega^2 r^5 \quad 4.75$$

Z_b is the number of buckets, x is the bucket active length [m] as represented in Figure 4.5, b is the bucket width [m], ω is the rotational speed [rad/s], ρ is the density of the fluid in which the wheel is rotating [kg/m³] and R_a is the Pelton wheel radius. The term x should be calculated as Figure 4.5 and this length is higher than the half-bucket length; however, x is approximated as half of bucket length after verifying that no significant variation occurs using higher values.

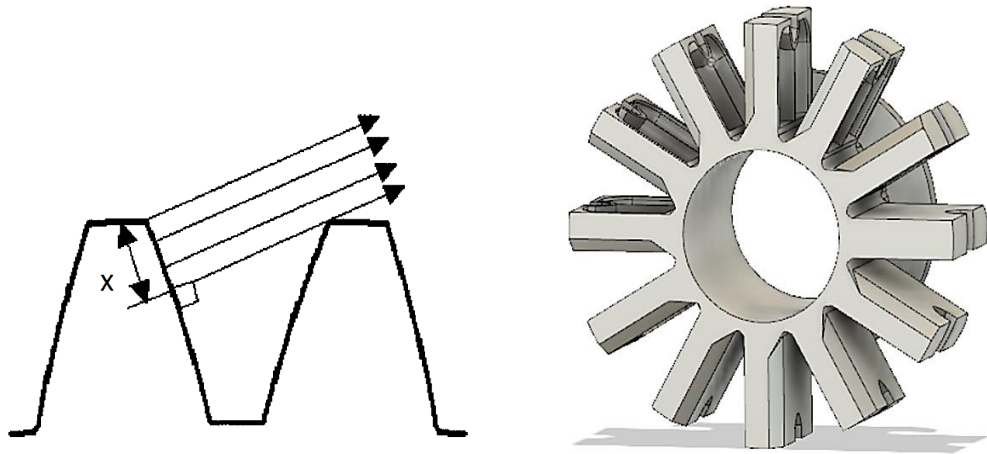


Figure 4.5 Geometrical representation of gear/bucket active length x for equation 4.72 (modified from [191])

4.6.2 Encased Pelton windage losses modeling

In FRP and TPSR, Pelton can be encased in the first configuration and it is always encased in the last configuration. Equation 4.76 [192] can be used, the geometrical parameters are referred to Figure 4.6 and ρ_w is a density parameter

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to represent the specific experimental case. This parameter in the original guidelines for windage losses in Pelton [192] is 15 but the authors specify that this value is valid only for a specific mix of water/air. In this thesis the value of ρ_w is usually considered just air, however increasing it leads to higher windage losses; this type of configuration and the role of this value is assumed to be significant in the original prototype (OTP) as the small draining hole create an oil mist surrounding in which the Pelton rotates.

$$T_{w,p} = \rho_w N^2 D_b^5 \left(\frac{B_a}{D_b}\right)^{1/4} \left(\frac{B_{io}}{D_b}\right)^{3/4} \left(\frac{B_{iu}}{D_b}\right)^{5/4} \left(\frac{R_{io}}{D_b}\right)^{7/4} \quad 4.76$$

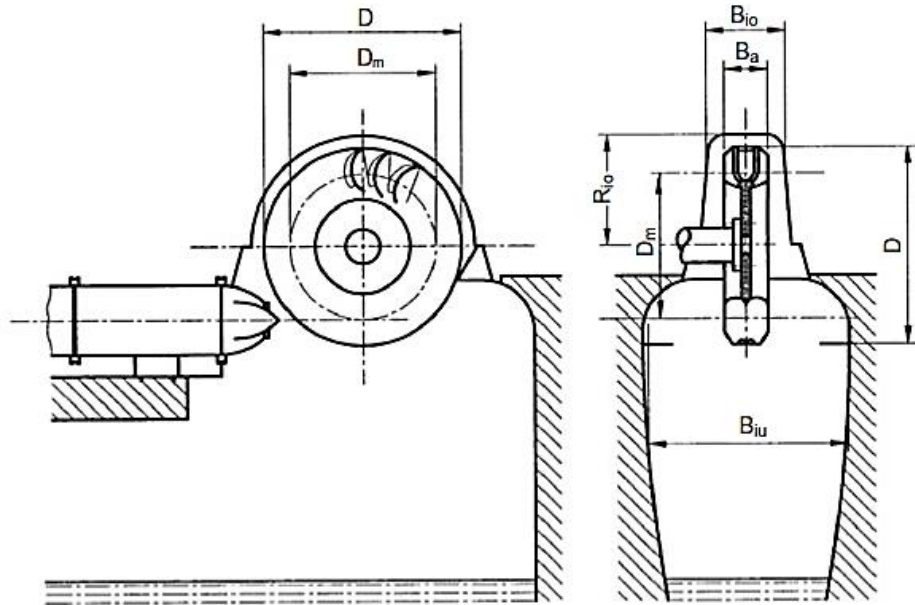


Figure 4.6 Parameter definition for calculation of windage losses in horizontal Pelton turbine according to IEC60041[192]

Comparing a free disc rotating in air and a Pelton, Zhang [162] revealed that a Pelton turbine in given condition can have windage losses about 16 times greater than at a comparable disk in open air. This is due to the presence of the buckets and a comparable high specific density of the air-water mixture. Another approach was considered to extrapolate windage losses contribute from experimental data; equation 4.77 from Tondell [193] can be used to determine empirical coefficient k_t to match experimental results. Once k_t is found it can

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be compared to other experimental results to see if it's suitable. It was chosen to proceed with the most reliable formula for Pelton windage and adjust it to experimental results if needed.

$$T_{w,p} = \omega^2 R_a^4 (R_a + k_t h) \quad 4.77$$

Where h is the buckets height [m] and k_t is an empirical coefficient.

4.6.3 Compressor/turbine windage losses modelling

In TPSR configuration, turbine and compressor wheels are considered additional losses according to the different inputs from the rig. When just oil is injected, both wheels represent a loss, whereas when air input is injected as well, just the compressor is ideally considered source of additional windage losses. Two models were used and compared; Buckingham formula [194] for a free gas turbine rotating in air is shown in 4.78 whereas Etemad et Al. formula is particularised for a turbocharger study [152], includes the casing effect and it is shown in 4.79.

$$T_{w,c/t} = \left(1 + 590 \left(\frac{L_{c/t}}{D_{c/t}}\right)^2\right) \left(\frac{60}{2\pi} \omega\right)^2 2D_c^5 \rho_{air} 10^{-6} \quad 4.78$$

$$T_{w,c/t} = 0.0311 * \left(\frac{s}{R_{c/t}}\right)^{-0.25} \mu_a^{0.25} \rho_a^{0.75} \left(\frac{60}{2\pi} \omega\right)^{7/4} R_{c/t}^{9/2} \quad 4.79$$

Where $L_{c/t}$ is the compressor/turbine blade length, $D_{c/t}$ and $R_{c/t}$ are respectively the compressor/turbine diameter and radius, ρ_a is the density of the air, s is the air gap between the wheel and its casing and ω is the rotational speed in rad/s.

4.7 Overview of Simulink Simulation Approach

4.7.1 Computational model simulation reasoning

In section 4.3.1 hydraulic accumulator discharge mathematical approach was explained. To investigate the problem dynamically, a 0D Simulink model was built. The pressure at the nozzles determines the oil speed and consequently the flow rate ejected which depends as well from the nozzles diameter. The complete physical system in KERB is modelled in Simulink and the resolution and simulation strategy can be summarised through the block scheme in Figure 4.7. Below each block is briefly described, whereas the detailed blocks of the model are shown in Figure 4.8 . The model is more extended, however the logic is the same for every output, therefore just the example for Rotational speed is shown.

- P is the output block of the hydraulic accumulator model sub-system, and it represents the Input block for the Pelton turbine system, including some variable parameters.
- ω is the Pelton angular velocity (state variable) (rad/s).
- $\dot{\omega}$ is the the Pelton angular acceleration (the rate variable) (rad/s²).
- Z-block sub-system is an Interpreted Matlab Function which calculates the variable ω in each iteration.
- The Integrator block convert the rate variable Pelton acceleration $\dot{\omega}$ to the state variable angular velocity ω .

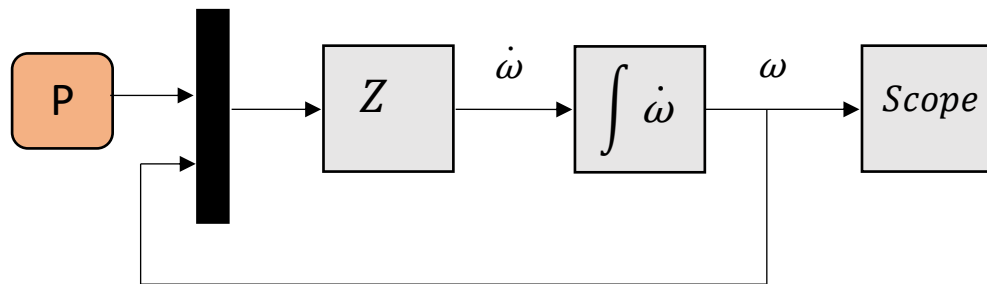


Figure 4.7 Simulink model resolution scheme block

The final computation is described in equation 4.80 where T is the ideal torque minus the friction and windage losses torques. For the angular velocity to reach convergence, a closed loop is used; this is set to a maximum value for which the

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Pelton speed must not exceed the jet velocity. The closed loop allows to take in account the system feedback to back control the result.

$$\dot{\omega} = \frac{d\omega}{dt} = \frac{T}{I} \quad 4.80$$

The Simulink model follows a series of steps to reproduce the equations provided in section 4.3, 4.4, 4.5 and 4.6. The parameters mask in Figure 4.8 the main system characteristic can be changed; these are listed in Table 4.6. In the sub-mask, the MatLab script “Acc_Script” calculates the pressure losses, the pressure at the nozzles and the flow rate. The important assumption considered is the continuity equation for which the flow rate is the same over the different sections. To comply with such statement, iterations are coded into the MatLab script so that the script accounts for the pressure losses since initially run. The script outputs the main variables and the demux block separates the volumetric flow rate, pressure at transducers and pressure at nozzle signal for further development. The flow rate $\dot{v}$ is used to calculate the pressure p_2 at the end of each timestep and from which the new flow rate is obtained. This is then fed into the new timestep to calculate the new pressure and volume in the gas chamber derived through the iterative process described in equation 4.14; this is achieved using two memory blocks to have the previous output of V_2 and p_2 . The gas chamber effective volume is 0.0178m^3 , which is its maximum volume.

Constructing a logic circuit by using a switch block and a stop block, the simulation stops when the gas chamber has reached its maximum volume. The stop block ends the simulation once the input is a nonzero value. When $V_2 < 0.0178\text{m}^3$, the input from the switch is 0, but when V_2 reaches 0.0178m^3 , the switch inputs 1 to the stop block and it terminates the simulation.

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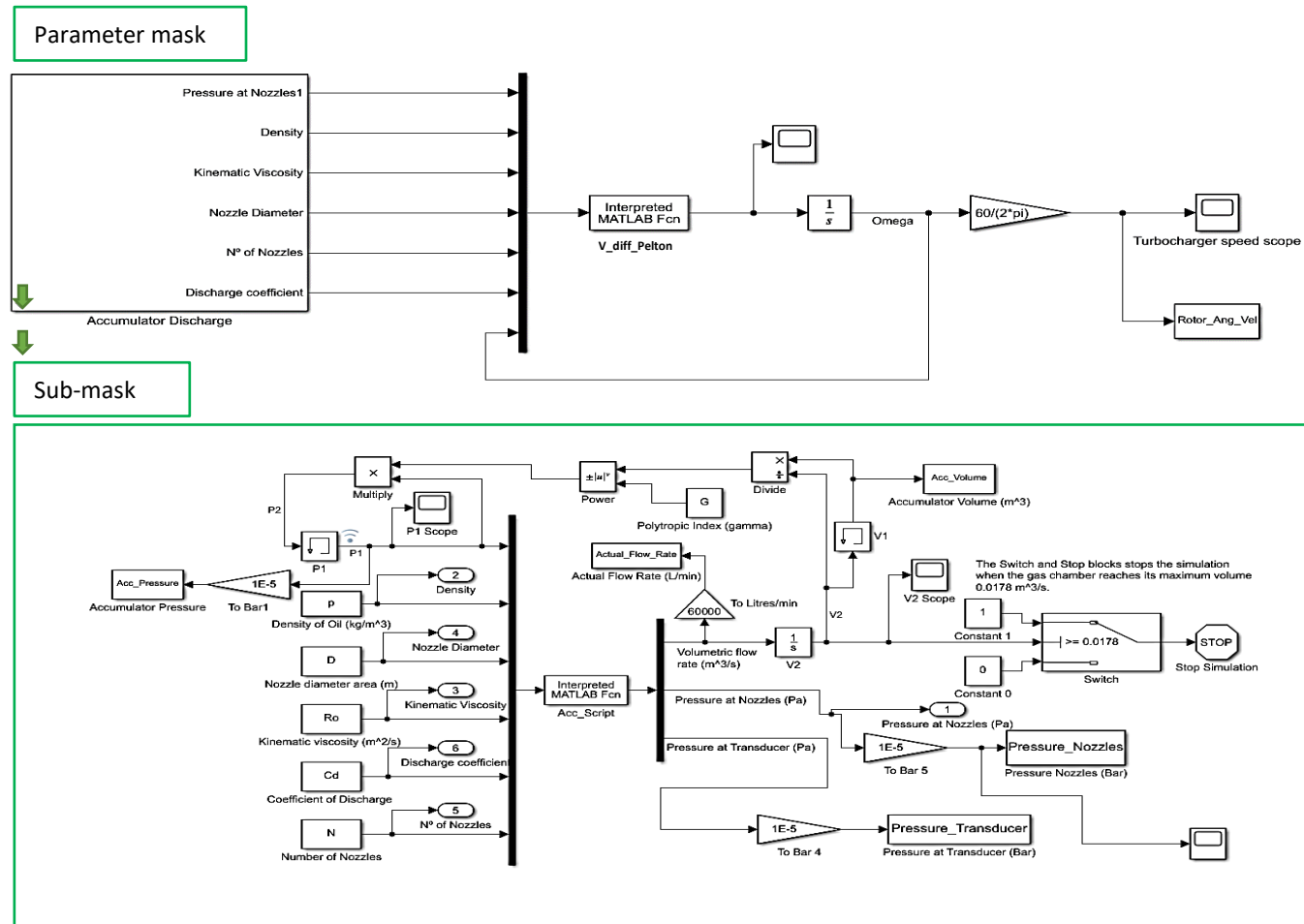


Figure 4.8 Complete Simulink model and Accumulator discharge Simulink sub-model

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The simulation is launched through an external script, which after running it, it saves the output for further investigation in Matlab. The other interpreted functions in the Matlab script concerns with Pelton dynamic including the calculation of the windage losses. The function *V_diff_Pelton* includes all the equation shown in the previous sections of this chapter and it has as output the rotational speed. The windage losses power are calculated similarly with same inputs as the function *V_diff_Pelton*. The static equilibrium equations are solved for each timestep giving punctual values for reaction forces at the bearings thus the friction loss in each timestep. The friction losses are subtracted in each timestep to the ideal torque computed by the model. A summary of inputs and outputs are shown in Table 4.5. The geometrical characteristics of the investigated assembly are part of the inputs. The nozzle diameters as well as number of nozzles indicated are the main ones investigated, however different values are used for parametric studies.

<i>Inputs</i>	<i>Range Values</i>	<i>Outputs</i>	<i>Range Values</i>
Charging pressure	0-300 bar	Rotational speed <i>N</i>	0-120k rpm
Oil density	971 kg/m ³	Pressure at the nozzles	0-280 bar
Oil kinematic viscosity	55.2E-6 m ² /s	Pressure at the transducers	0-280 bar
Number of nozzles	1-2	Oil Mass Flow rate	0-62 L/min
Nozzle Diameter	1.1-1.5mm	Ideal Power	0-11kW
Discharge Coefficient	0.87	Bearing Power loss	0-500W
Moment of Inertia	5-10E-5 kgm ²	Overall Windage losses	0-7.6kW
Initial Gas chamber volume	1-17.8 L		

Table 4.5 List of inputs and outputs of the computational model for the HAT simulation

4.7.2 Modelling solver and timestep approach

To simulate a dynamic system, Simulink software [195] computes its states at successive time steps over a specified time span using information provided by

a model of the system; this process is defined as “solving” the model. The simulation time t_{sim} is the total time simulated by the software and the time step $t(k + 1) = t(k) + \Delta t$ is the time interval when the computation happens; its size Δt is called the step size. Simulink uses different approaches to solve a model the so-called “solvers”. A solver applies a numerical method to solve the set of differential equations that describe the model. The suitable solver for simulating a physical system depends on characteristics such as system dynamics, solution stability, computation speed and solver robustness. The numerical solvers in Simulink can be classified according to two properties:

1. Computation step size

- *Fixed-step solvers* solve the model using the same step size throughout all the simulation. Decreasing the step size increases the accuracy of the result but also the time required to simulate the system.
- *Variable-step solvers* vary the step size during the simulation. The step size is reduced at certain events during the simulation to increase accuracy or increased in other events to avoid taking unnecessary steps when the states of a model changes slowly.

2. Model States

- *Continuous solvers* use numerical integration to compute continuous states of a model at the current time step based on the states at the previous time steps and the state derivatives. These solvers rely on individual blocks to compute the values of the discrete states of the model at each time step.
- *Discrete solvers* are used for solving discrete models in which the state variables change only at a countable number of points in time. They compute only the next simulation time step for the model and they rely on each block in the model to update its individual discrete state without computing continuous states.

In Simulink it is also possible to check and improve simulation accuracy varying the relative tolerance in the model when a variable-step method is selected. The

4 Computational Model

relative tolerance specifies the largest acceptable solver error, relative to the size of each state during each time step. If the relative error exceeds this tolerance, the solver reduces the step size. A value of relative tolerance of 1E-3 means that the computed state is accurate to within 0.1%. The steps to check simulation reliability and accuracy when a variable step method is used are the following:

- Simulate the model over a reasonable time span.
- Reduce either the relative tolerance to 1e-4 (the default is 1e-3) or the absolute tolerance.
- Simulate the model again.
- Compare the results from both simulations. If the results are not significantly different, the solution has converged.

A Fixed time step means that in $T(k + 1) = T(k) + \Delta t$, the step size Δt is constant. The choice of the right time step leads to a result that is closer to the actual solution. A fixed step continuous explicit solver computes the value of a state at the next time step as an explicit function of the current values of both the state and the state derivate (equation 4.81).

$$x(n + 1) = x(n) + \Delta t * Dx(n) \quad 4.81$$

Where x is the state, Dx is a solver-dependent function that estimates the state derivative and n indicates the current time-step.

The different solvers provided for a fixed step continuous model are based on numerical integration technique and are listed in Table 4.6 from the least complex (ode1) to the most complex (ode8). The number next to “ode” identifies the order of accuracy of the method and it quantifies the rate of convergence of the numerical approximation of a differential equation to the exact solution.

<i>Solver</i>	<i>Integration technique</i>
ode1	Euler's method
ode2	Heun's method
ode3	Bogacki-Shampine formula
ode4	Fourth-order Runge-Kutta (RK4) formula
ode5	Dormand-Prince (RK5) formula
ode8	Dormand-Prince RK8(7) formula

Table 4.6 Solvers and integration techniques used by Simulink for solving fixed step continuous models

In Pelton dynamic complete model, a fixed-step continuous solver is used. The Pelton model is indeed firstly considered as a continuous states model because the state variable ω change in a continuous way and not abruptly from one state to another. Secondly, the model is also considered solvable with a fixed-step method as there are no special events that require an increased accuracy during the simulation. A suitable timestep size is chosen according to the following convergency criteria method:

1. The model is simulated for 5 different timestep at same initial condition: 1s, 10^{-1} s, 10^{-2} s, 10^{-3} s, 10^{-4} s;
2. The Pelton speed value at different t_{sim} is identified for each timestep;
3. The speed % difference $\Delta\omega$ is calculated between each consecutive timestep at the identified t_{sim} ;
4. When $\Delta\omega\% < 0.02\%$ for the identified t_{sim} , the timestep is considered sufficiently accurate and no further reduction is needed.

In table Table 4.7 is shown the timestep comparison and the resulting $\Delta\omega$ at different timestep. According to these results, reducing Δt from 10^{-2} s to 10^{-3} s would result in $\Delta\omega\%=0.01$ rpm which satisfy the set criteria. Simulation timestep is then chose $\Delta t = 10^{-2}$ s .

The automatic solver selection selects the most suitable solver to the model selected; in the simulation the auto-solver suggests ode3 Bogacki-Shampine.

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This method is a Runge-Kutta method which uses three function evaluations per step.

Timestep Δt comparison [rpm]				
	1s vs 10^{-1} s	10^{-1} s vs 10^{-2} s	10^{-2} s vs 10^{-3} s	10^{-3} s vs 10^{-4} s
$\Delta\omega$ 1s	5.03%	0.07%	0.01%	0.0007%
$\Delta\omega$ 3s	1.75%	0.06%	0.01%	0.0006%
$\Delta\omega$ 5s	1.07%	0.06%	0.01%	0.0006%
$\Delta\omega$ 7s	0.82%	0.05%	0.01%	0.0005%
$\Delta\omega$ 9s	0.69%	0.05%	0.005%	0.0005%

Table 4.7 $\Delta\omega$ speed value according to different simulation timestep

4.8 Summary

This summary highlights the main outcomes from the computational chapter to identify the essential information of the complete numerical modelling to simulate KERB experimental rig.

- An Accumulator discharge can be modelled using 0D MatLab and Simulink software with bespoke differential equations providing the pressure trend decrease during oil ejection from one or multiple nozzles in high-pressure applications.
- Pelton dynamic and friction losses can be modelled through manipulations of momentum equation obtaining ideal torque. Knowing the characteristic inertia of the system, the angular acceleration can be derived, thus the final rotational speed integrating the acceleration over time. The final differential equation is then solved through the 0D simulation.
- Three different configurations of windage losses can be outlined bases on the different experimental configurations tested for a hydraulic-assisted turbocharger.

4 Computational Model

- To simulate the angular velocity of a Pelton wheel and reach convergence of results, a closed loop must be used; this must be set to the maximum value for which the Pelton linear speed must not exceed the jet velocity speed.
- The Ordinary Differential equation for hydraulic Pelton wheel injected with high-pressurised oil in a nozzle from a hydraulic accumulator can be simulated and solved with a timestep of 0.01s which respect the convergence criteria of a speed percentage difference $<0.02\%$ when decreasing the timestep magnitude.
- The innovation relating the computational model presented in the thesis is the ability of correlating dynamically the hydraulic discharge of an accumulator to the variation of the rotational speed of a Pelton turbine.

5 Hydraulic Assist

Experimental Investigation

5.1 Introduction

This chapter includes the experimental investigations carried out on the following configurations:

1. Original turbocharger prototype (OTP) with journal bearings;
2. Original turbocharger prototype retrofitted with ball bearings;
3. Free running Pelton with multiple casings configurations (FRP).

Experiments aims and data analysis are initially described and preliminary tests on the OTP including the modification carried out on the bearing casing are shown.

The FRP was tested to identify the parameters to optimise the new casing for the HAT prototype. At the end of each section, a discussion of results is presented; it includes the main outcome of the experimental campaign, the constraints and, when possible, the comparison with the computational model. In the summary, the main outcomes are highlighted, defining the criteria for the design of the new prototype in Chapter 6. Further discussion is available in Chapter 7.

5.2 Experiments aims and Data analysis

The aims of the experimental campaigns in this chapter were to:

- Preliminary test the OTP;
- Identify potential power losses from bearings and windage;
- Validate the computational model;
- Identify the optimum geometrical parameters to design a bespoke bearing housing for the new assisted-turbocharger prototype;

The study of the windage losses is further discussed in the experimental campaign and discussion on the HAT is in Chapter 6 and Chapter 7. The combination of parameters that influenced speed trend were:

- Number of nozzles: 1 or 2 nozzles;
- Diameter of nozzles: 1.1mm, 1.5mm;
- Types of casings (described in section 5.4.1);
- Charging pressure at the accumulator: 40 to 280 bar.

The main test matrix used for free running Pelton (FRP) experimental campaign is shown in APPENDIX B: Test matrix free running Pelton experimental campaign. Each run with same combination of parameters was carried out at least 3 times. Some casings were not tested further and not included in the matrix as they showed at low pressure overflowing, thus they were discarded.

Every investigation is characterised specifying the charging pressure, the number of nozzles and the nozzle diameter. For example, “40 bar, 1x1.1mm” define the average results of a series of test carried out charging at 40 bar the hydraulic accumulator and using one nozzle of 1.1mm diameter. If not specified otherwise, the accumulator, after being charged with Nitrogen, was closed. Occasionally the communication between the Nitrogen bottle and the accumulator gas-side was left open to test maximum performances of the rig. The pressure of the oil at the nozzles, does not correspond to the initial oil charging pressure because of the pressure losses along the hydraulic line. As a rule of thumb, for the rig used in this research project, the maximum pressure at the nozzle inlet can be considered as 86-90% of the charging pressure at the accumulator.

The tests were performed with BP Turbine Oil 2380 as the viscosity value of this oil at temperature of 20°C (58.2 cst) is similar to viscosity of standard engine oil at higher temperature in real operation. The viscosity of the oil decreases with the increase of the temperature, however this effect is neglectable for the study of this chapter. The computational model showed that the influence on the final rotational speed is 2-3%; this range is within the error accepted according to the

DOE analysis shown at the end of the section. The tests available in this chapter were all performed without introducing the exhaust gas input.

The reported graphs of the speed are represented with line+symbol. The data acquisition frequency was 100 Hz meaning that every 0.01s a measure is taken. For representation purposes the symbols show the data collected every 0.2s but the curve is built using all the collected points. The modelled data are instead represented with a continuous line. The data are filtered through Adjacent Average method that allows wide smoothing as the speed trend does not present any peak behaviour. Given $\{f_i | i = 1, 2, \dots, N\}$ as the input data point and $\{g_i | i = 1, 2, \dots, N\}$ the output data point, each output g_i is the average of the data points within the moving window. Specifically, each smoothed data point is computed from data points within a moving window of 50 units; this window was chosen comparing lower and higher number of points and evaluating which optimised value maintained the reliability of the curve in terms of maximum peak and time response. A window with fewer points would have not removed the noise of the trend, on the other hand, a window too wide would have distorted the realistic values of the rotational speed losing experimental information on time response and maximum speed. Summarizing, the adjacent averaging takes the average of a user-specified number of data points around each point in the data, and replaces that point with the new average value.

For each configuration at least 3 tests are carried out in the same operating condition; this is done to take into account the normal fluctuations of the system response and perform an investigation that gives statistically significant results.

In the comparisons and results presentation, the “Max speed time response” is defined as the maximum speed reached by the rotating assembly. The “Absolute time response” is instead calculated using speed difference in consecutive timesteps to give realistic data to compare for future studies avoiding the potential errors introduced by data acquisition rate. The starting moment of this absolute time response is identified when the speed difference between a consecutive timestep and the other is higher than 50 for 3 consecutive timesteps in the chosen acquisition rate, and similarly the end is identified when the same difference goes below 50 rpm for 3 consecutive timesteps. This last time response is important to identify when the regime speed is reached; this last

definition will be more used in chapter 6 however some results in this chapter are presented similarly to give a tool for comparison in the conclusions chapter.

Regarding the error analysis a DOE analysis was carried out to identify the variables that played a significant role in speed trend. The preliminary shakedown on FRP highlighted that charging pressure and nozzle diameter played an important role on final rotational speed, significantly varying the maximum velocity and the time response. This was confirmed by multi-factorial analysis using experimental results. A parameter to focus on was the oil charging volume. The accumulator has a maximum volume of 20L; to optimise the execution of the tests of all the experimental campaigns, the accumulator was filled at approximately 10L and the tests were run until the oil was finished. This allowed 3-7 tests for a set condition. The maximum expected variation due to only the oil charging volume (keeping the other parameters constant) was identified to understand its impact on the final rotational speed. As an example, FRP configuration was tested at 80 bar with 1x1.5mm nozzle and the speed was compared at 1s; this value was chosen as it is representative of the turbocharger time response range. Seven tests were performed varying the initial oil charging volume from the highest level 14L to the lowest level 2L. The value of the rotational speed was averaged across the tests and it was 33,460 rpm with a maximum variation of 1523 rpm (4.5%) between maximum and minimum values (33846-32323 rpm) and a standard deviation of ± 508 rpm. It was then deduced, with a conservative approach, that the oil charging pressure could cause a 5% max speed difference when comparing experimental tests. This was verified for several tested configurations at different time conditions.

During these tests in which it was also verified that nozzle discharge coefficient was 0.87.

Summarizing, the potential error in the experimental results is given by two contribute: the variation of the data of 5% shown in this section caused by the variation of the oil charging pressure and the intrinsic instrumentation errors given from pressure transducers reading and speed sensor reading shown in paragraph 3.2.2.

The interest of the experimental campaign is to look at significant improvement in rotational speed varying the geometrical and hydraulic

parameters of the assembly and for this reason and for representation purposes the experimental trends are presented without error bars.

5.3 OTP: Original Turbocharger prototype experiments

5.3.1 Preliminary tests

The original prototype in Figure 5.1 was inherited from a previous project, it was then tested to investigate its performances as no shakedown was carried out after its manufacturing. The OTP included the typical components of a traditional turbocharger, but the journal bearing casing was modified to accommodate the 32mm Pelton turbine selected from the previous PhD [49]. The aims of this specific experimental campaign were to:

- Test hydraulic rotational speed in the OTP assembly;
- Identify potential friction losses causes;
- Identify constraints of the design.



Figure 5.1 Original HAT Prototype on its support

Before testing the OTP, a dummy chamber was used to investigate nozzle finishing, discharge coefficient and hydraulic lock risk. The results showed a possible overflowing even at low accumulator pressures. The hydraulic lock was expected to create stress on the mechanical components and to significantly decrease Pelton speed. The preliminary tests for the OTP were then carried out initially at low pressures to verify the prototype behaviour. Despite the prototype

was put in motion when the oil was injected, the leakages, the noise and the speed curve trend suggested that vibrations and hydraulic lock between the components were occurring as expected. The conclusions was that the original casing was not able to withstand the hydraulic demand. Higher pressures were tested to have a term of comparison with previous results from Justus [49], who carried out speed tests on a free running Pelton-inertia disc system supported by two ceramic ball bearings (FRPJ). In OTP 1x1.1mm nozzle was used to mitigate the risk of overflowing and to have a good term of comparison with Justus experimental campaign with 1x1.2mm nozzle. The real rotational speed in OTP is significantly lower than expectation projected from previous work on the prototype. With a 1.2mm nozzle and 100 bar charging pressure, the peak speed recorded in FRPJ was around 50,000 rpm, whereas at the same pressure with 1x1.1mm nozzle, the OTP showed around 12,500 rpm (Figure 5.2). This speed reduction occurred because of the assumed hydraulic lock experienced in the OTP; this had just 13mm of available draining surface compare to the ideally unlimited draining in FRPJ.

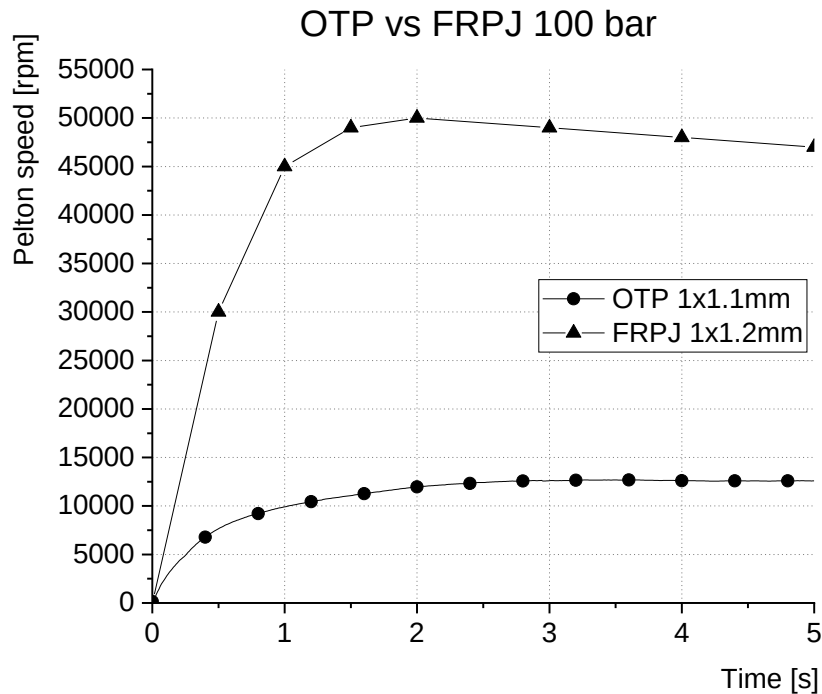


Figure 5.2 Comparison results Original HAT prototype vs free running Pelton in [49]

*in OTP the marker point skips 20pt for graphic representation reasons, the data acquisition frequency is always 100Hz

5.3.2 OTP modifications and ball bearing retrofit

Because of the OTP limitations in terms of speed performances given from design faults, it was decided to retrofit and modify the prototype to explore improvement solutions. The turbo assembly was taken apart (Figure 5.3) marking the shaft to ensure the balance when assembled again. Signs of wear were evident on the plain bearing placement. This suggested that although the correct amount of lubrication was provided [163], [164], the plain bearings were among the causes of the low rotational speed. Another possible cause was identified in the incorrect position of the nozzle in respect to the Pelton wheel. The nozzle is supposed to be positioned 90° in respect to the Pelton buckets and in correspondence of the pitch circle diameter of the wheel [162]. Finally, the other main cause for a low rotational speed was assumed to be hydraulic lock due to the small dimensions of the Pelton chamber.

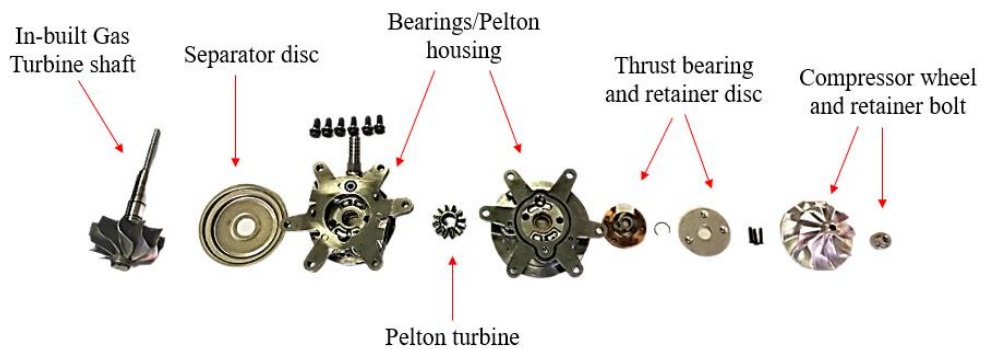


Figure 5.3 Original hydraulic-assisted prototype central casing disassembled

The main modifications to the assembly consisted in:

1. Use the upper nozzle hole as additional draining;
2. Modify the nozzle slot and the support to allow vertical movement;
3. Retrofit with ball bearings.

The vertical movement accuracy was ensured through a bespoke support for the turbocharger which kept the high-pressure hard pipe and the nozzles rigid. The tests carried after the main modifications focused on identifying if an additional

draining and the further vertical movement of the nozzle could improve the dynamic performance of the assembly. The tests were carried out taking a reference position “0mm” (Figure 5.4). The additional draining and the vertical movement improved the rotational speed from 14.5% to 34% (Figure 5.4). The benefit of the optimised position is confirmed later in section 5.4.2 in FRP rig. Additionally, ball bearings were retrofitted in the plain bearing casings. Ball bearings do not require continuous lubrication and they have the capacity of spool up faster during an acceleration event in the engine [163]. The external covers on the bearing side were drilled as the external bearing diameter was bigger than the plain bearing shoulder; a new compressor disc was created and adapted with a gasket to prevent leakages (Figure 5.5).

5 Hydraulic Assist Experimental investigation

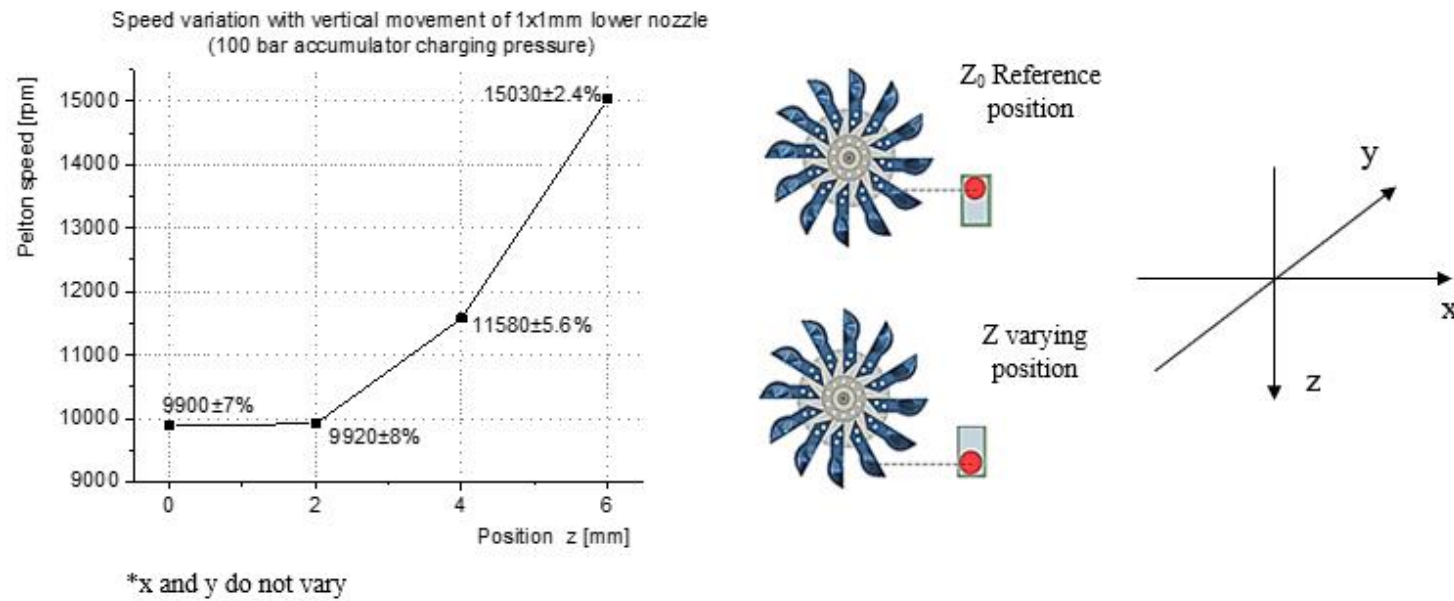


Figure 5.4 Nozzle Vertical position impact on final rotational speed in OTP.

(DTI=Dial Test Indicator)



Figure 5.5 Ball bearing retrofit in OTP

The necessary press fit force K_{ap} and removal force K_{ar} for the bearings were calculated through equations 5.1 and 5.2 [196]; they depended on the finish of the shafts and how much interference the bearings allow.

$$K_{ap} = 9.8 f_{kp} \Delta_{def} B \left(1 - \frac{d^2}{D_i^2} \right) 10^3 \quad 5.1$$

$$K_{ar} = 9.8 f_{kr} \Delta_{def} B \left(1 - \frac{d^2}{D_i^2} \right) * 10^3 \quad 5.2$$

Where f_k is a resistance coefficient and it depends on the conditions, Δ_{def} is the effective interference [mm], B is the nominal inner ring width [mm], d is the nominal inner ring bore diameter [mm] and D_i is the average outside diameter of the inner ring [mm]. Geometrical and coefficient values are available from Schaeffler bearing catalogue [197].

Tests carried out at 40 and 60 bar accumulator charging pressure showed an improvement in peak speed of 42-47% using the new ball bearings compared to

the plain bearing configuration. Despite the experimental campaign proved that rolling bearings were more performant than plain bearing the maximum final speed was always lower than 20,000 rpm. To accommodate the ball bearings, the compressor casing was removed (Figure 5.6) increasing consequently the windage losses in addition to the hydraulic lock visually experienced during testing. The overall modifications to the prototype were temporary and after 25-30 tests, the reliability of the experimental assembly, due the leakages and to the unbalanced shaft for the addition of the bearing, was compromised. This was evident from the speed trend which did not resemble anymore the typical one from a Pelton speed dynamic, presenting significant oscillations even after noise filtering. The max speed difference with the same input conditions reached a variation of $\pm 25\%$.

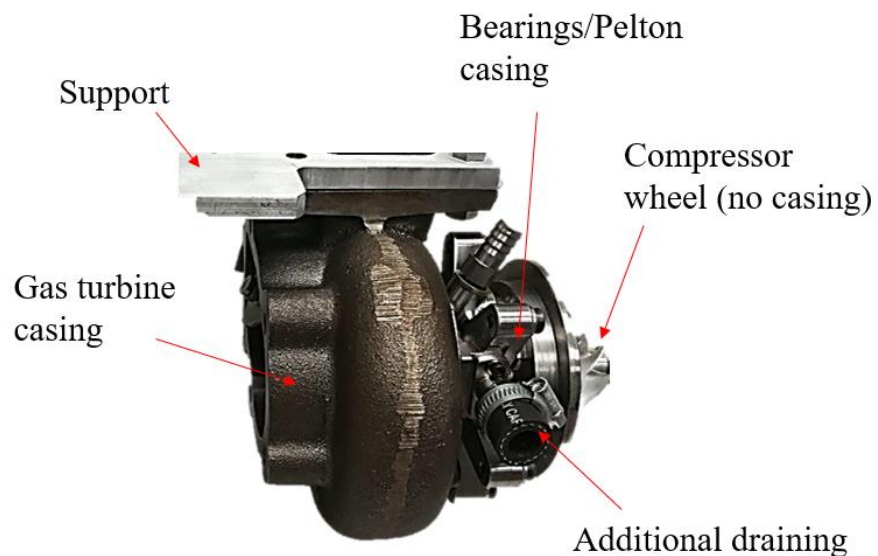


Figure 5.6 Modified OTP configuration

5.3.3 Discussion of the results

The results of the experimental campaign on the OTP showed critical constraints related to hydraulic lock and friction torque from the bearings. Some attempts to replicate the behaviour through the computational model were carried out however they were not successful. The assumption is that the variability of the geometrical characteristics of the bearings necessary to calculate friction torque in Equation 5.6 impacts the final predictions. The journal and thrust bearings

originally mounted on the OTP were hydrodynamic fluid film bearings. The stationary surface is separated from the rotating surface by a thin film of lubricant; the film pressure is created by the relative rotation of the surfaces as the lubricant is pulled into a converging geometry between them. Thrust bearings counteract axial loads, while journal bearings support radial loads perpendicular to the shaft axis of rotation. To calculate the correspondent friction torques T_j and T_{th} , linear radial velocity gradient is assumed in the shear stress τ (equation 5.3). Friction force F_b is derived in 5.4 and Petroff's equation [186] expresses T_j for journal bearings (equation 5.5), and T_{Th} for thrust bearings in expressed in equation 5.6 [198]:

$$\tau = \mu_{oil} \left(\frac{du}{dy} \right) = \mu_{oil} \left(\frac{r_b \omega}{c} \right) \quad 5.3$$

$$F_b = \tau \times 2\pi r_b L_b = \frac{2\pi r_b^2 L_b \mu_{oil} \omega}{c} \quad 5.4$$

$$T_j = F_b \times r_b = \frac{2\pi r_b^3 L_b \mu_{oil} \omega}{c} \quad 5.5$$

$$T_{Th} = \frac{\pi \mu_{oil} \omega (r_o^4 - r_i^4)}{\varepsilon} \quad 5.6$$

Where $\mu_{oil} \left[\frac{kg}{m \cdot s} \right]$ is the oil dynamic viscosity, u is the linear velocity [m/s], c is the radial clearance [m], ω is the rotational speed [rpm], r_b is the journal radius [m], L_b is the bearing length [m], ε is the axial clearance [m], r_o is the outer radius [m] and r_i the inner radius [m].

From the available drawings of the OTP, radial clearance c and axial clearance ε were not available. Radial clearances are usually given by the manufacturer, but a range value of 5 to 75 μm is considered common in midsize industrial equipment [168]. A parametric study with the Simulink computational model was carried out varying the radial clearance in the range 5-75 μm to investigate how the variation of radial clearance could influence the speed trend.

5 Hydraulic Assist Experimental investigation

Assumptions were to have $\varepsilon = 63.5 \mu\text{m}$ and only bearing losses $T_j + T_{th}$ as power losses detracted from the ideal Pelton torque T_p . A step of $\Delta c = 10 \mu\text{m}$ was chosen for the parametric study and results are shown in Figure 5.7.

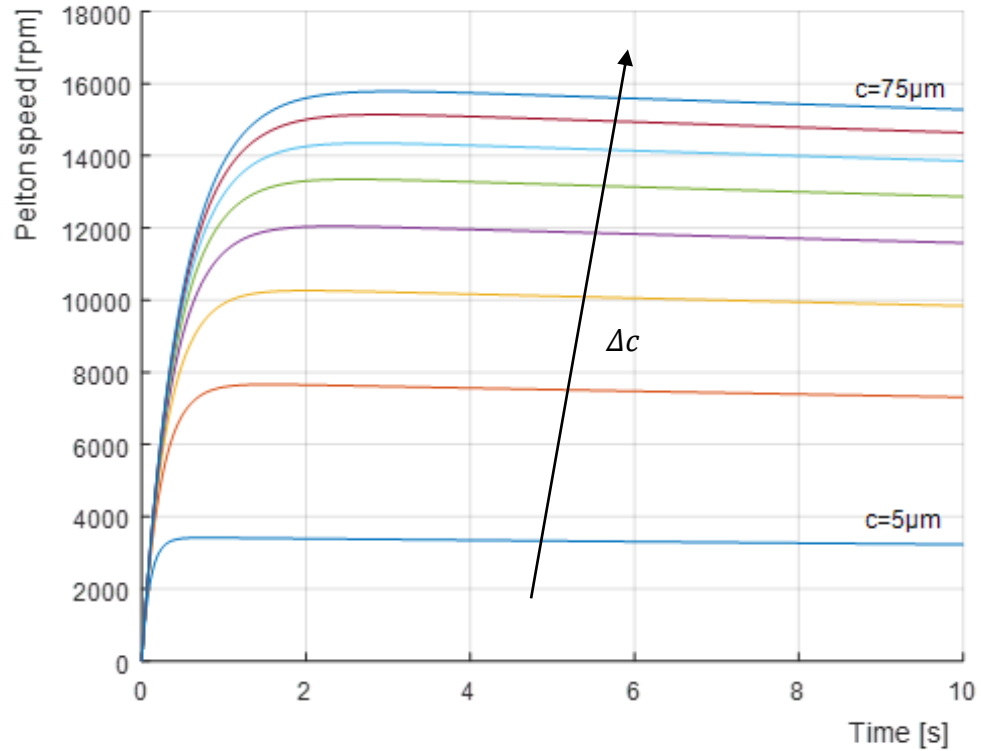


Figure 5.7 Sensitivity study on Pelton rotational speed varying radial clearance c of journal bearing at steps of $\Delta c = 10 \mu\text{m}$

The study shows that the maximum average difference in speed is 76%. The difference among speeds for each clearance step varies from 4% to 56%. In conclusion, the radial clearance value impacts massively on journal bearing friction losses torque T_j . This means that assuming an average value of $35 \mu\text{m}$ for the radial clearance could lead to a possible incorrect evaluation of Pelton speed up to 70%. For this reason, it was not possible to validate the model with the OTP.

The next step in the experimental campaign was then to build a free running Pelton (FRP) configuration to investigate the optimised parameters to design a new turbocharger prototype.

5.4 FRP: Free running Pelton experiments

5.4.1 FRP Design

A new configuration, namely FRP (free running Pelton), was designed to investigate the parameters influencing the hydraulic assist performances on a turbocharger prototype and their impact. The original version was the FRPJ used in [49] and it included a platform with a bearing housing mounted, a shaft with ball bearings, the Pelton turbine on one side of the shaft and an inertia disc on the other side of the shaft. The FRP took steps from it modifying it to allow further investigation on possible Pelton casings; afterwards a new version of the bearing housing was designed and manufactured (Figure 5.8) giving the final configuration of the FRP. Overall, the improvements were:

- New Inertia disc: the original inertia disc was in cast steel, aluminium or titanium were needed to generate the eddy currents detected by the speed sensor to compute the disc revolutions. The new disc was designed to have the moment of inertia of the disc in the same order of magnitude of a rotating assembly of a Garrett GTX-2867r (10^{-5} kgm²).
- Speed sensor seat: a support was designed to accommodate the speed sensor probe and adjust it according to the different configurations.
- Nozzle seat: this was needed to accommodate new designed nozzles for the prototype ensuring vertical and horizontal movement when needed.
- Platform modifications: to accommodate legs and draining hole.
- Bearing housing: this was redesigned to adapt new angular contact ball bearing (ACBB) and the several casings for the Pelton turbine.
- Pelton covers/casing: Covers for the Pelton to investigate hydraulic lock constraints. When there was no cover, a containment tank was mounted to prevent flooding of the oil through the rig.

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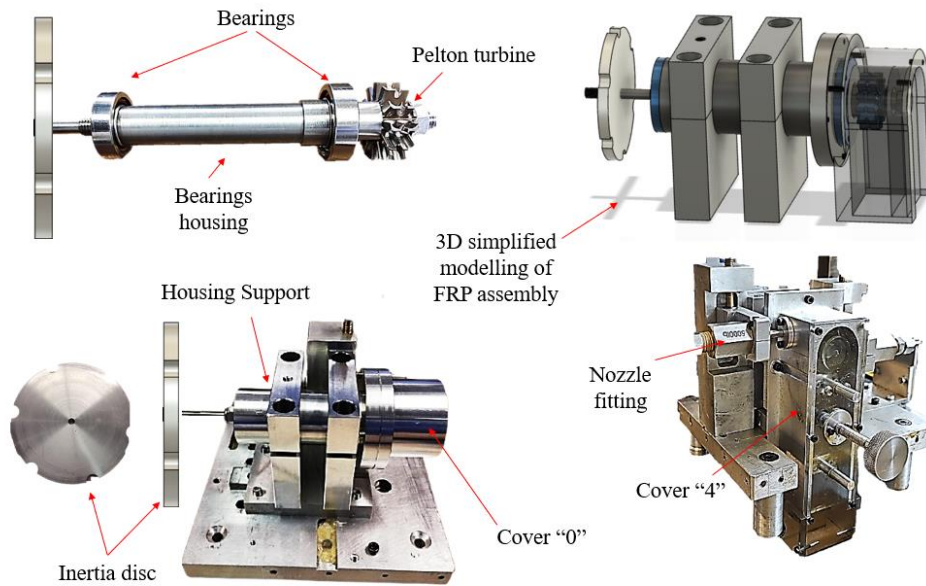


Figure 5.8 FRP components, bearing housing and details of the shaft (covers referred in details in Figure 5.10)

All the components were designed in Autodesk Fusion 360 and manufactured in L4 lab or L2 labs workshop. The shaft and the assembly were balanced by the Power Electronic and Machine Centre at the University of Nottingham removing some material from the inertia disc. This was done to avoid vibrations and potential cause of low performances and catastrophic failures. Different covers were manufactured and adapted to the FRP bearing housing to investigate an optimised casing for the new prototype (Figure 5.10). The “No cover” configuration included the Pelton without any casing restriction. Each cover configuration was an evolution from the previous one; the aim was to investigate how much the volume could be reduced without incurring on hydraulic lock. The configurations are explained below and the experimental results are available in section 5.4.4. For reference when axial and radial dimensions are cited Figure 5.9 is available.

0. Cover “0” has half-moon shape, it has the same inner diameter of the OTP (40mm), the axial dimension of 36mm and the nozzle is positioned in the optimised vertical line correspondent to the pitch circle diameter. A front layer can be attached to cover the front-side of the Pelton as shown in cover “1”.

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1. “Cover 1” is “Cover 0” plus a polycarbonate panel extension of the front-side and lateral-sides.
2. “Cover 2” is “Cover 1” with the front and side panels elongated until the metal platform of FRP support. The available draining is restricted.
3. “Cover 3” is “Cover 2” with the front-side panel moved to modify the axial dimension (Figure 5.9) available from the original 27mm to 18mm with still the draining restriction.
4. “Cover 4” is the final configuration and it combines different aspects of the previous covers. The peripheral diameter (radial dimension in Figure 5.9) is 40mm, it has elongated lateral and front panels and the latter can move to change the axial dimension (Figure 5.9) from 18-27mm; the draining hole changes accordingly to the axial direction varying its dimension then from 40x18mm to 40x27mm.

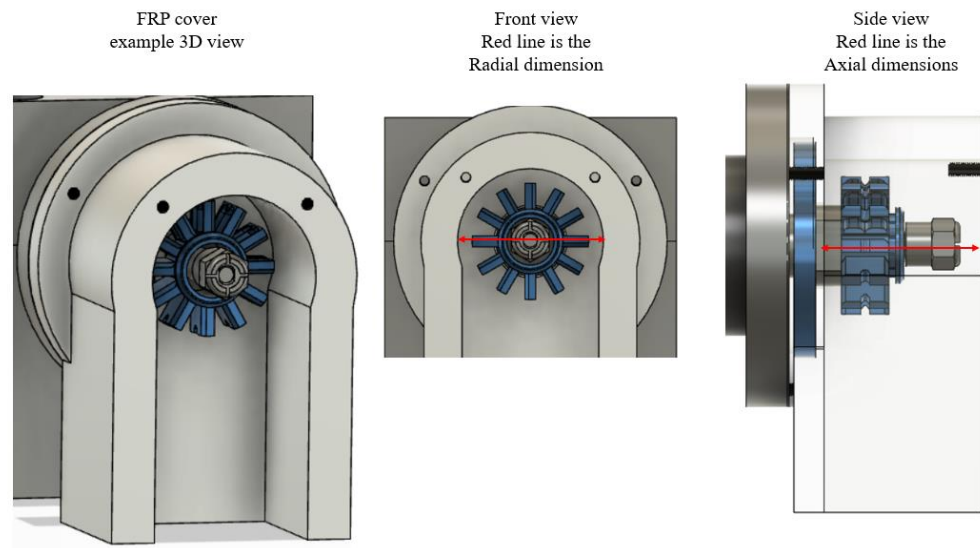


Figure 5.9 Geometry references for radial and axial dimensions in FRP covers

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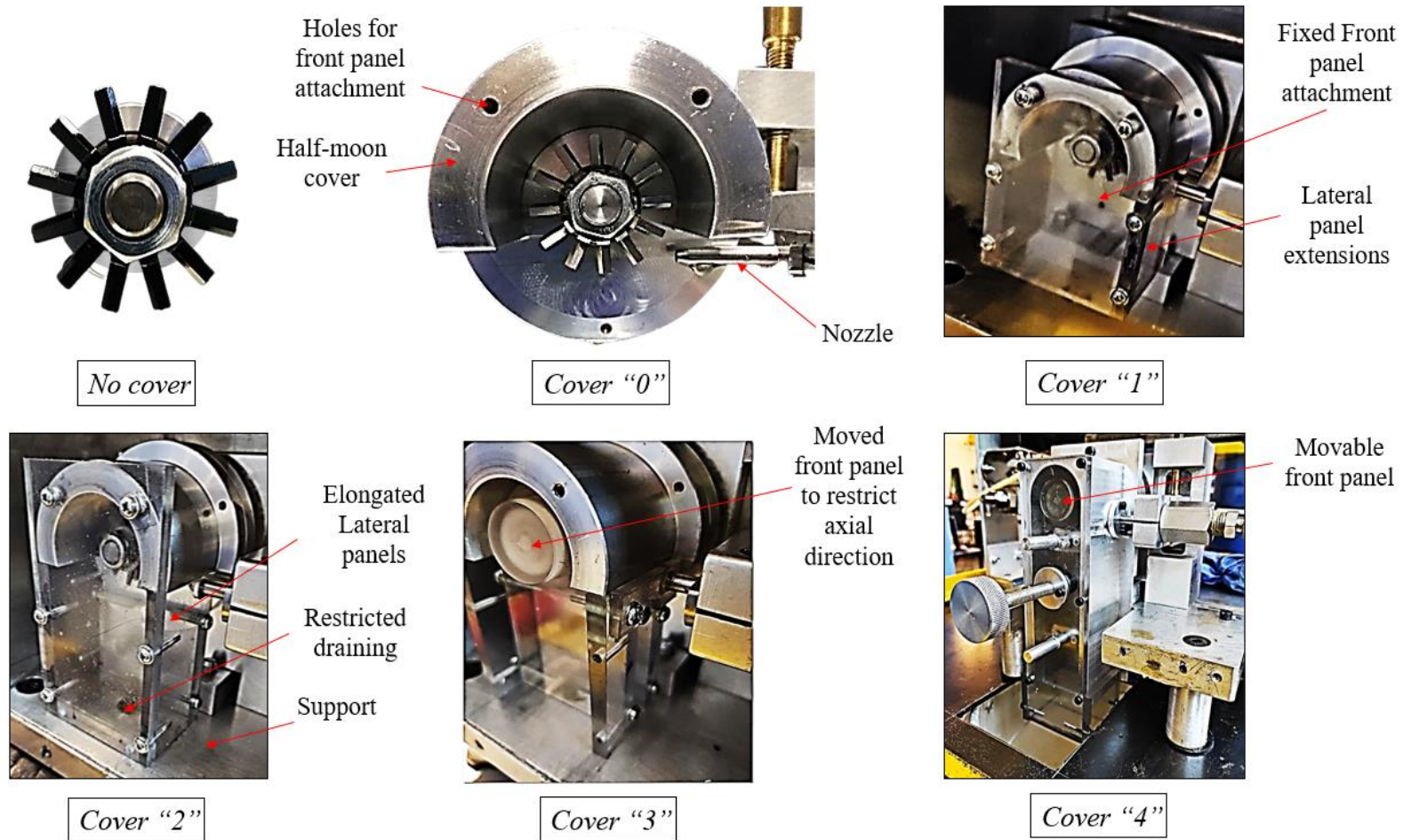


Figure 5.10 Covers for FRP configurations

5.4.2 Jet input influence

Nozzles characteristics were investigated to acquire information for optimizing the design for the new prototype. The parameters analysed in this chapter are: number of nozzles, nozzle diameter and nozzle vertical and horizontal position.

Firstly, the double vs single nozzle configurations were studied using “Cover 4”. Going from one to two nozzles increases the top speed by a maximum of 3.6% at the highest tested pressure of 160 bar; however the main advantage is the gain in time response which is up to 63%. To quantify this gain, the top speed for the 1x nozzle case is measured, then the time to achieve the same speed in the 2x nozzle case is identified. Results are summarised for different charging pressure in Table 5.1.

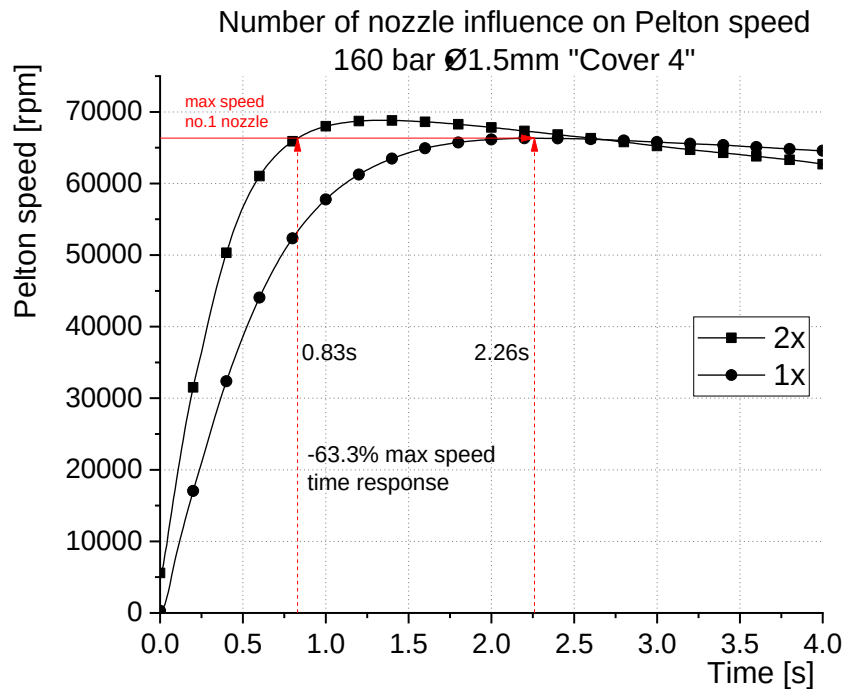


Figure5.11 Number of nozzles influence on Pelton speed (160 bar Ø1.5mm)

<i>Pressure</i>	<i>Top speed increase</i>	<i>1x Max speed Time response</i>	<i>Max speed Time response reduction using 2 nozzles</i>
80bar	+2.1%	3.66s	-63.4%
120bar	+3.4%	2.67s	-62.5%
160bar	+3.6%	2.26s	-63.3%

Table 5.1 Comparison of speed and time response using 1 vs 2 nozzle Ø1.5mm

Secondly, nozzle diameter was investigated to determine the benefit in increasing the nozzle diameter related to the flow rate to manage. The two diameters investigated were 1.1mm and 1.5mm. The results in Figure 5.12 show that, increasing the nozzle diameter, both top speed and max speed time response are improved respectively of +20% and -68% in the case studied. The increase of nozzle diameter from 1.1mm to 1.5mm results lead to an increase of +46% in the flow rate.

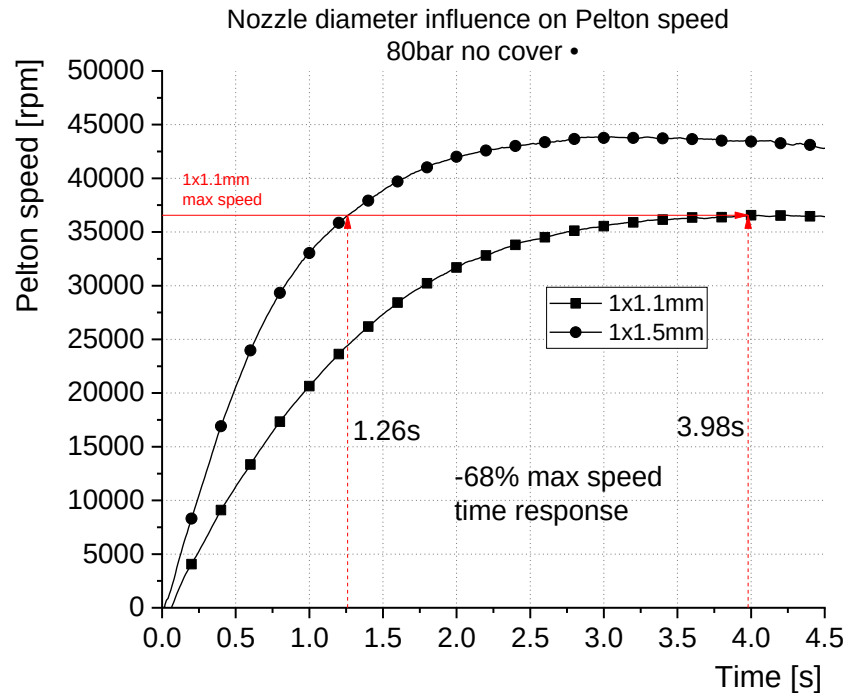


Figure 5.12 Nozzle diameter influence on Pelton speed 1.1mm vs 1.5mm

Finally, the position of the nozzles was investigated. The Pelton wheel is a tangential flow impulse turbine; the bucket is designed as a hemispherical cup to be struck by the fluid along the tangent of the runner in correspondence of the splitter. The splitter divides the jet in two equal parts deflecting the fluid at the exit of the bucket usually at 170° . As mentioned and tested in section 5.3.1, the vertical position of the nozzle influenced indeed the speed performances. The same type of test was performed with a higher level of precision in FRP rig. Horizontal movement impact of final rotational speed was tested; because of the symmetry of the geometry, just one side was tested; the Pelton bucket width is 11mm and compared to the centre the jet was moved 3mm on the horizontal plan using the sliding tracks in the FRP platform and a Dial Test Indication (DTI). The results in Figure 5.13 showed a maximum speed difference of 20.3% as, as expected, the deflection of the nozzle jet from the splitter significantly worsen the speed performances. The vertical movement influence was confirmed to be an important parameter that influences the speed performances; the tests were preliminarily carried out in OTP configuration and the results were confirmed in FRP configuration. In Figure 5.14 the results are available, together with the reference distance used. The starting arbitrary point was named as 0mm, and the nozzle was moved up at steps of 2mm until reaching 6mm. The best performance was obtained at 2mm with a maximum increase of 28%; this point most probably matched the pitch circle diameter of the Pelton. It is important to vertically position correctly the nozzle compared to the wheel, as the maximum performance is obtained when at least 2 buckets at the same time benefit from the jet flow [162].

5 Hydraulic Assist Experimental investigation

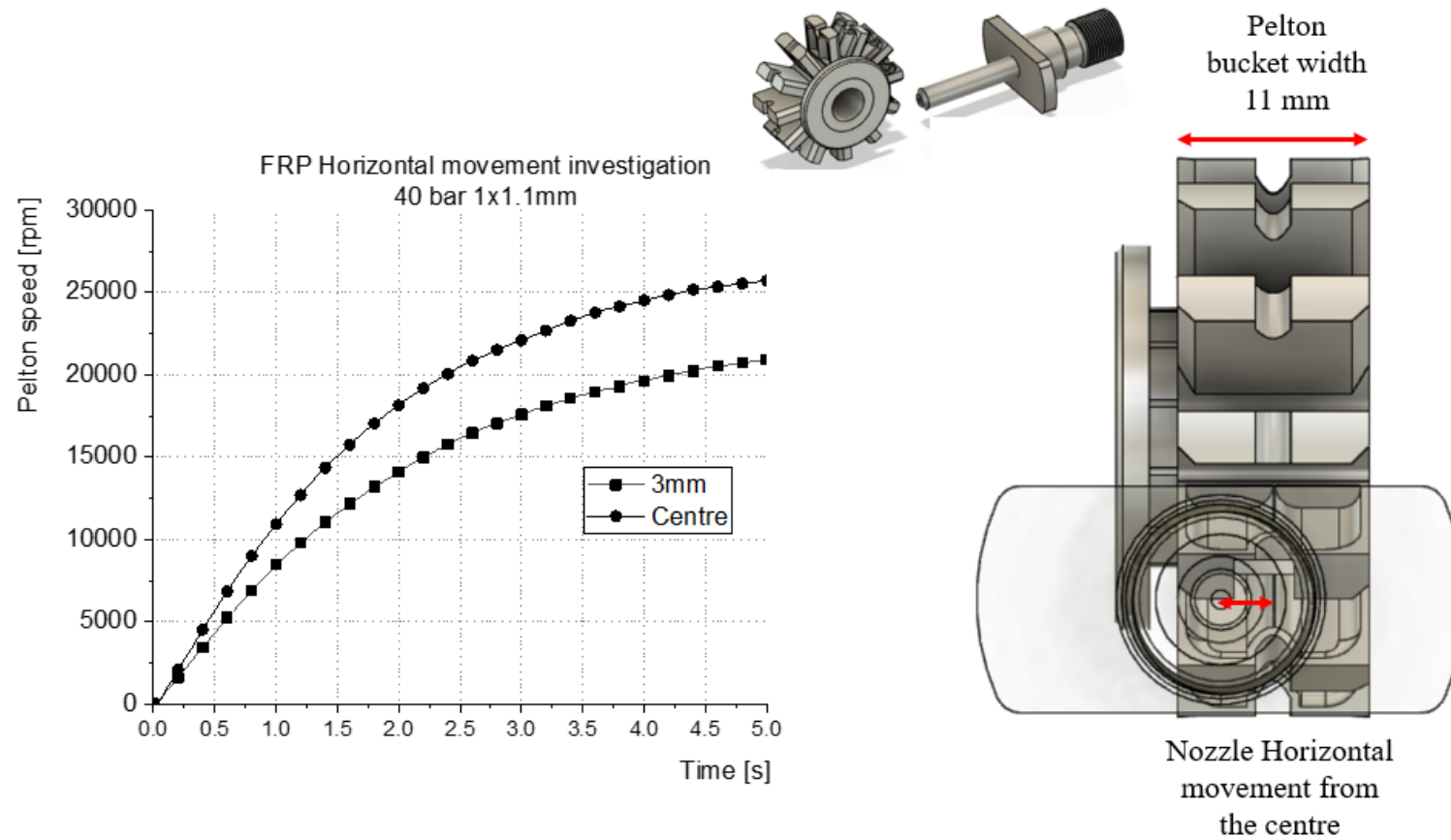


Figure 5.13 FRP Horizontal movement investigation on rotational speed at 40 bar and 1x1mm

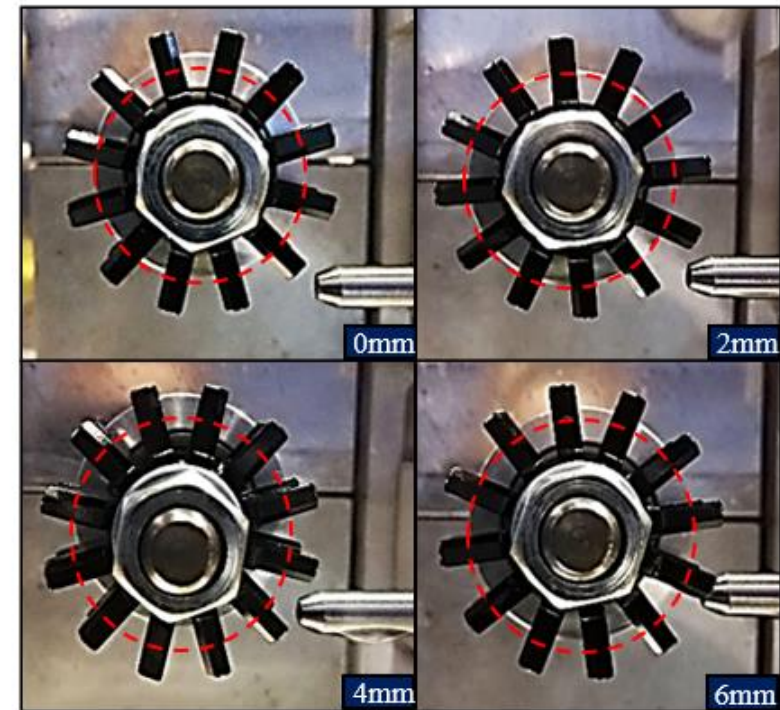
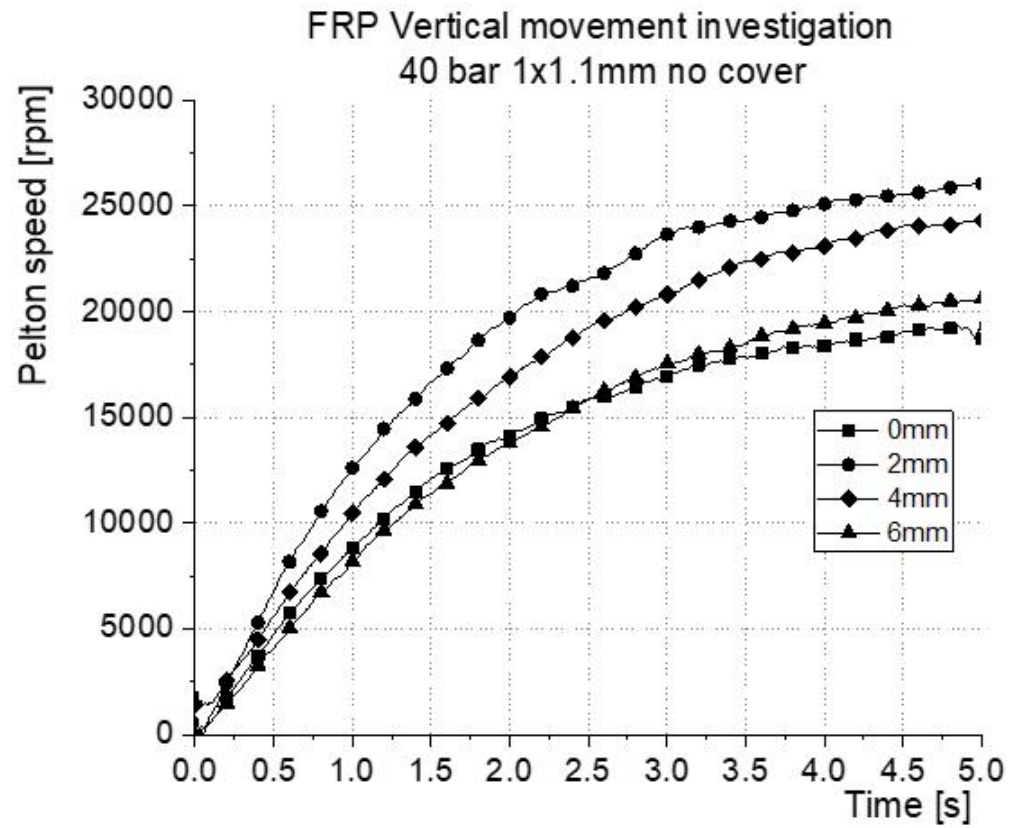


Figure 5.14 FRP vertical movement investigation on rotational speed at 40 bar and 1x1mm nozzle. Pitch circle diameter is shown in red dotted line.

5.4.3 Charging pressure influence

Numerous tests were carried out varying the charging pressure of the Nitrogen in the hydraulic accumulator. In the first instance it was possible to reach just 180 bar using 1 bottle of 9.8 m³ of Nitrogen at 230 bar, the limit was increased to 220 bar using a GENIE bottle of 20L at 300 bar and finally it was possible to reach 280 bar using a bank of 15x50L bottle of Nitrogen at 300 bar..

Several other tests with different combination of nozzle number and nozzle diameter were carried out varying the charging pressure however the main comparisons at different pressure are done using casing 4 and 2 xØ1.5mm nozzles; other configurations showed the same pattern in terms of results. The complete test matrix is available in APPENDIX B: Test matrix free running Pelton experimental campaign. The significant results are shown in Figure 5.15. Increasing the charging pressure results in decreasing the time response and increasing maximum speed. From 40 bar to 160 bar the max speed time response decreases up to -58%. With same pressures analyzed for 1x1.5mm nozzle the max speed time response decrease -55%. The maximum speed, almost 100,000 rpm in 1.2s, was reached with 2x1.5mm nozzle at the maximum pressure reachable with the bank of bottles left open (Figure 5.15). This test however was performed uniquely as it was not possible to guarantee the repeatability due to the maximum operating conditions. The same test carried out with 1x1.5mm nozzle although reaching a similar speed, has a max speed time response of 1.8s. Comparing the two curves then, the advantage is clear in both the absolute and max speed time response having the double nozzle configuration with a -30% of rapidness, and the compared relative time response.

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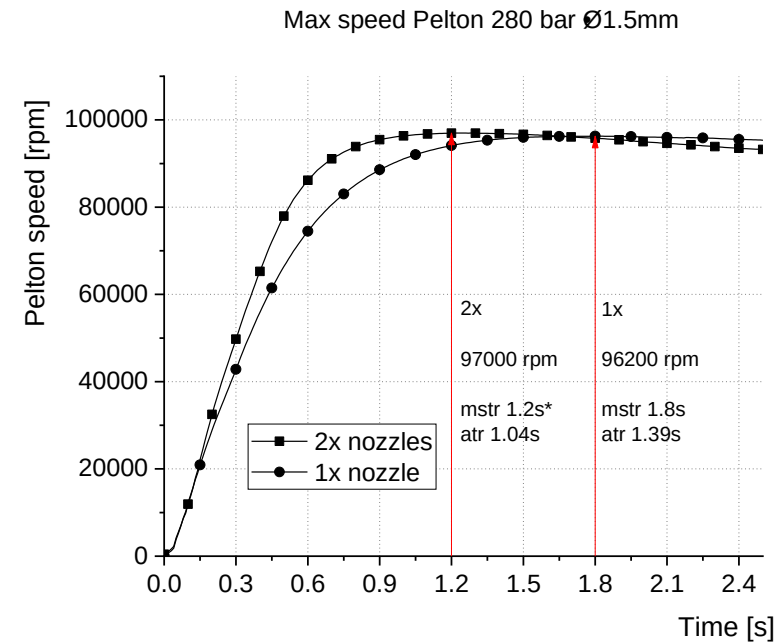
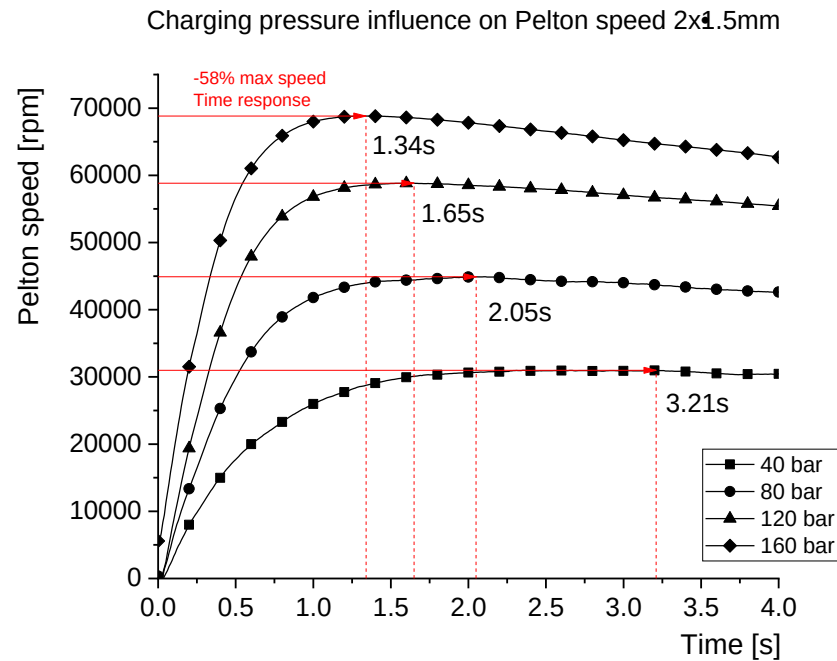


Figure 5.15 Charging pressure influence on Pelton speed/max speed time response using 2x1.5mm nozzles (left) and Maximum Pelton speed achieved at 280 bar charging pressure using 1.5mm nozzles (right).

*mstr: max speed time response, atr: absolute time response

5.4.4 Casing optimization

An essential step in the experimental campaign of FRP was to test different casing/cover configurations. The aim was to identify the best approach to maintain three main conditions: high rotational speed, avoiding overflowing and producing a compact design to implement in the HAT prototype. In section 5.4.1 the difference designs for the Pelton cover were shown and the configurations are available in Figure 5.10. All the tests were performed with 1 or 2 nozzles of Ø1.5mm. Cover 2 and Cover 3 did not produce reliable results as the first showed oil leakage through the nozzle hole and in the second it was not possible to ensure a perfect sealing between the end of the casing and the metal platform. The final comparison is carried out between no cover configuration and Cover 0, 1 and 4. The final configuration “Cover 4” has a variable axial distance and it was able to replicate then the smallest axial direction achievable and the radial dimension of original turbocharger prototype. The results in Figure 5.16 show that the best configuration is indeed represented by “Cover 4”. In section 5.2 it was assumed that a difference in speed trends below the 5% could be attributed just to an oil charging volume variation. For both 120 bar and 160 bar charging pressure, the max speed difference %S with casing configuration showed that between no cover, cover 0 and 1 the parameter %S was <1%, however casing 4 has %S>5% compared to the other configurations indicating the speed increase is due to a significant parameter such as the casing modification. The reason of being Cover 4 the best performing configuration lies in the smaller dimension of the casing. This was optimised to avoid hydraulic lock but mitigating excessive windage losses due to the two-phase oil-air rotating flow. Windage losses are widely explained in the Computational chapter; they are enhanced when more space is available for the fluid to rebound from the walls into the wheel instead of being addressed out of the casing.

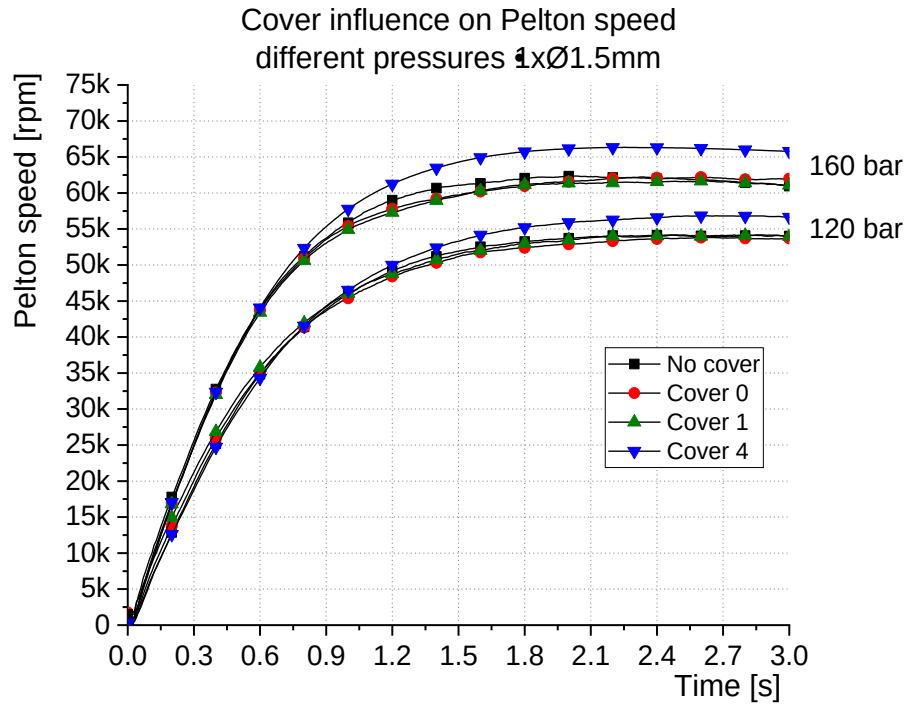


Figure 5.16 Cover influence on Pelton speed at 120 and 160 charging pressure with 1x1.5mm nozzle (referring to covers in Figure 5.10)

5.4.5 Discussion of Results

An overall amount of over 200 tests were carried out in FRP with different configurations. Among the several casings configurations, Cover 4 represented the optimised solution to implement in the HAT. It includes all the beneficial aspect required for the encased Pelton wheel in the turbocharger casing. The inner movable panel showed that it was possible to reduce as much as needed the radial space without impacting the final rotational speed. The open bottom, that follows the dimensions of the inner chamber diameter, allows to avoid the use of a scavenge pump. Compared to the original turbocharger prototype in same conditions, the advantage in maximum speed with the optimised configuration in FRP is on average more than +80% confirming the initial criticalities caused by the hydraulic lock issue. The time response investigation showed that the double nozzle injection was the best option. The benefit is however reached to the detriment of the higher oil flow rate. Envisaging an open bottom in the new HAT as Cover 4 optimised configuration, it does not represent a constraint to take in account. For the same reason, a greater nozzle diameter is considered the best option (1.5mm).

5.4.6 Computational Model Validation and Predictions

The experimental campaign was complemented with a computational analysis using the Simulink model shown in Chapter 4. This was validated with experimental data, however, some considerations must be taken in account. Oil charging volume and accumulator charging pressure measurements had intrinsic uncertainties due to the nature of the system. Details on accuracy of the sensors are shown in section 3.2.2. The oil charging volume was shown as well in section 5.2 to be the main cause of potential 5% difference in the expected values.

The model validation was carried out comparing pressure and speed trends with the experimental results. The preliminary comparisons with the computational model showed that the pressure trend did not accurately match the average pressure trend of the experimental data. It was assumed an unknown accumulator pressure loss P_{acc} due to the oil poppet valve at the end of the accumulator. A coefficient F_a was then derived to be used in 5.7; this was implemented into the modelled equations set for the hydraulic circuit pressure losses.

$$P_{acc} = p_1 * F_a \quad 5.7$$

For the parametric study to determine F_a , a step of comparison for F_a was chosen to be 0.01; the reason for that is shown in Figure 5.17. Steps of $\Delta F_a = 0.01$ keeps average speed difference $\Delta N\%$ of Pelton rotational speed within 1% which is below the Turbospeed sensor sensitivity. Steps of 0.01 are then considered to be enough accurate for the comparison to the pressure evaluation (Figure 5.17).

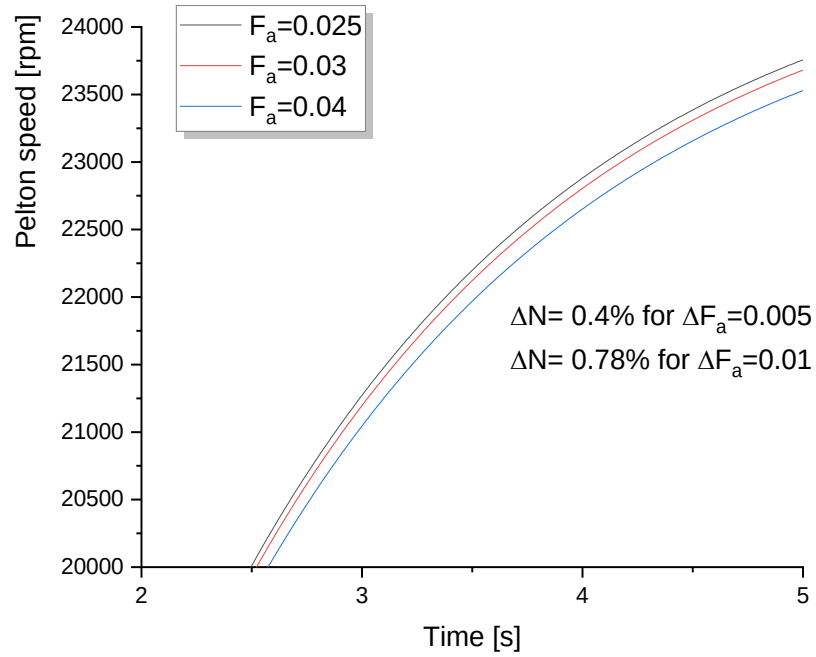


Figure 5.17 Sensitivity computational study of Pelton speed varying pressure coefficient F_a (Equation 5.7) steps

For the validation several pressure trends were analyzed to ensure F_a was valid for a wide range of experimental conditions. For 1.1mm nozzles, an example is available in Figure 5.18 (left). All the tests confirmed the best pressure trend match was verified for $F_a=0.03$. This value ensured that Difference % exp – model was the minimum, and example is shown in Figure 5.18 (right). The difference % between experimented and modelled value of the pressure is calculated as per equation 5.8 where n are the data acquired according to the selected timestep.

$$\text{Difference \% exp – model} = \frac{\sum \left| \left(\frac{p_{exp} - p_{model}}{p_{exp}} \right) * 100 \right|}{n} \quad 5.8$$

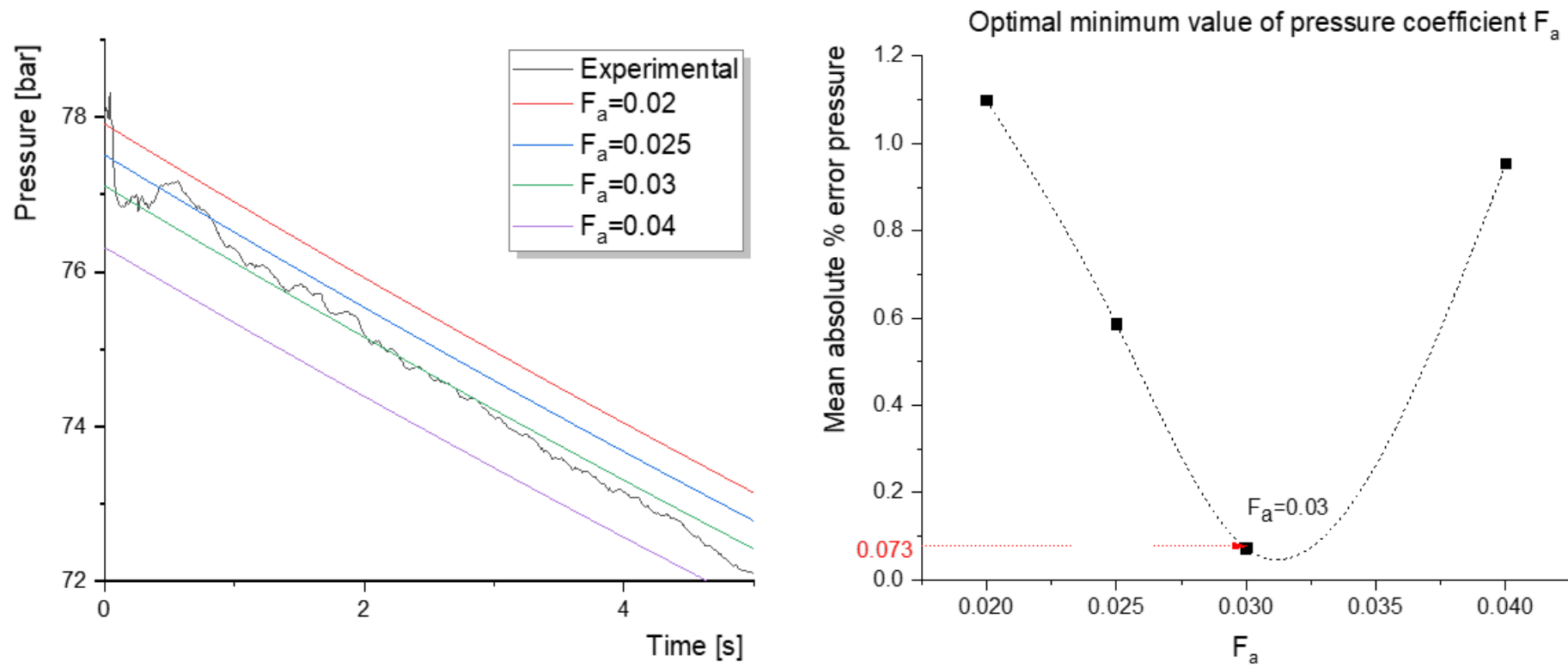


Figure 5.18 Pressure trend experiments modeled varying F_a (Equation 5.7) (left) and its optimal value for 1.1mm nozzle(right)

The same procedure was carried out for 1.5mm nozzle and it was found that the coefficient F_a has a similar trend to a polynomial curve dependent on the charging pressure (Figure 5.19). Although the fit was not highly accurate, it was used in the model to provide some repetitive behaviour to attribute to the pressure coefficient.

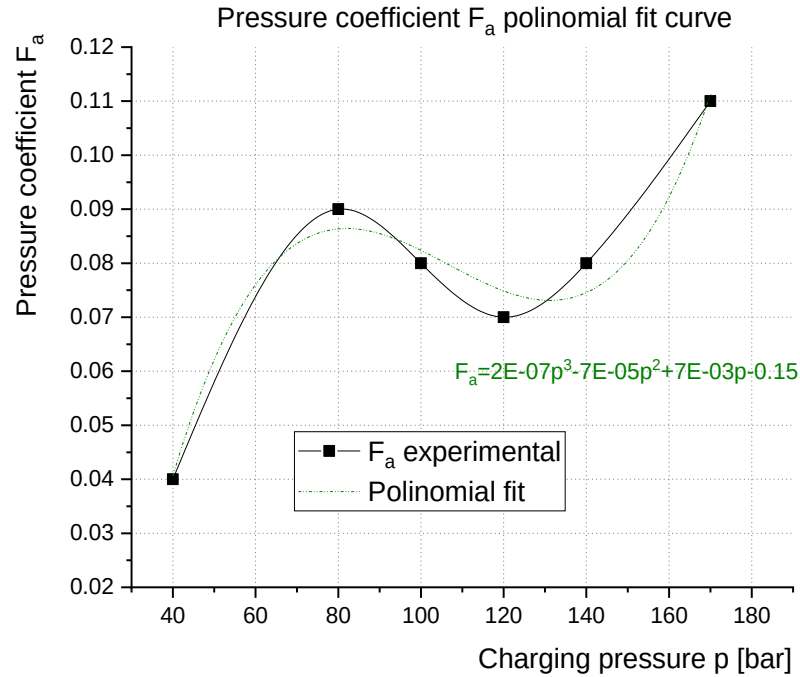


Figure 5.19 Polinomial fitting curve for pressure coefficient F_a (Equation 5.7) in 1.5mm nozzle diameter configuration

In terms of speed validation, the process was carried using data from FRP with no cover configuration and cover 4 configurations. The correct replication of the data was highly related to the windage losses equation used. Three model approach were used as per Table 5.2 to analyze the influence of the windage losses for the various components. The equation used are already mentioned in Chapter 4.

CASE	Windage losses Pelton	Windage losses inertia disc
A	Considered	Considered
B	Not Considered	Considered
C	Considered	Not considered

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Table 5.2 Case study for computational model comparison with FRP configuration

The first comparison is carried out using the free running Pelton without a casing. In case A both equations for a free rotating disc (replicating the inertia disc rotating) [147] and a free rotating Pelton [144] were used. In case B the equation for the windage of the Pelton is ignored and in case C the equation for the inertia disc is ignored. All the three cases are summarised in Figure 5.20 for an example case study for the free running Pelton with no cover at 80 bar with 1x1.5mm nozzle. The comparison between the cases shows that in terms of maximum speed the most accurate modelling is done with Case A or Case B in which the maximum speed difference with the experimental data is just 3.5% and 3.4% respectively whereas Case C presents an unacceptable error. Among these two best cases, Case A has an underestimation of the real time response of more than 30% whereas case B 23%. It is then assumed that the inertia disc windage loss contribution is correctly considered, whereas the oil injected in the Pelton can increase the inertia of the assembly around +40% of the original values computed for the assembly which is as well shown in Figure 5.20. A 3D fluid dynamic modelling could compute the amount of oil that the Pelton carries during the rotation to understand if the given reason through the preliminary modelling study is reasonable. A +40% inertia would mean considering approximately as the Pelton would carry continuously 58ml of oil. With the condition in Figure 5.20 the flow rate is 11.8 L/min which means that 58ml are ejected in 0.3s and can be remained trapped in the Pelton rotation causing an additional inertia. Case B is taken for the optimised FRP model for no casing configurations considers then the windage losses from the inertia disc and an increased inertia of +40%. The maximum speed is matched with an error of 1.9% and the time response with a underestimation of 5%.

A second comparison was carried out considering the Pelton with the casing 4. The three cases were studied as well, however the windage losses Pelton formula used was the encased Pelton formula from Zhang [162]. In this configuration a good match regarding the maximum speed is given with Case A (both windage of inertia disc and encased Pelton considered) (Figure 5.21) where the difference with the experimental speed is just 0.22%; however, the time response is overestimated of 24%. The best fit model for encased Pelton is then obtained with Case A+40% inertia where the maximum speed difference is

within the acceptable range of error being just 2% and the time response is underestimated of just 6%.

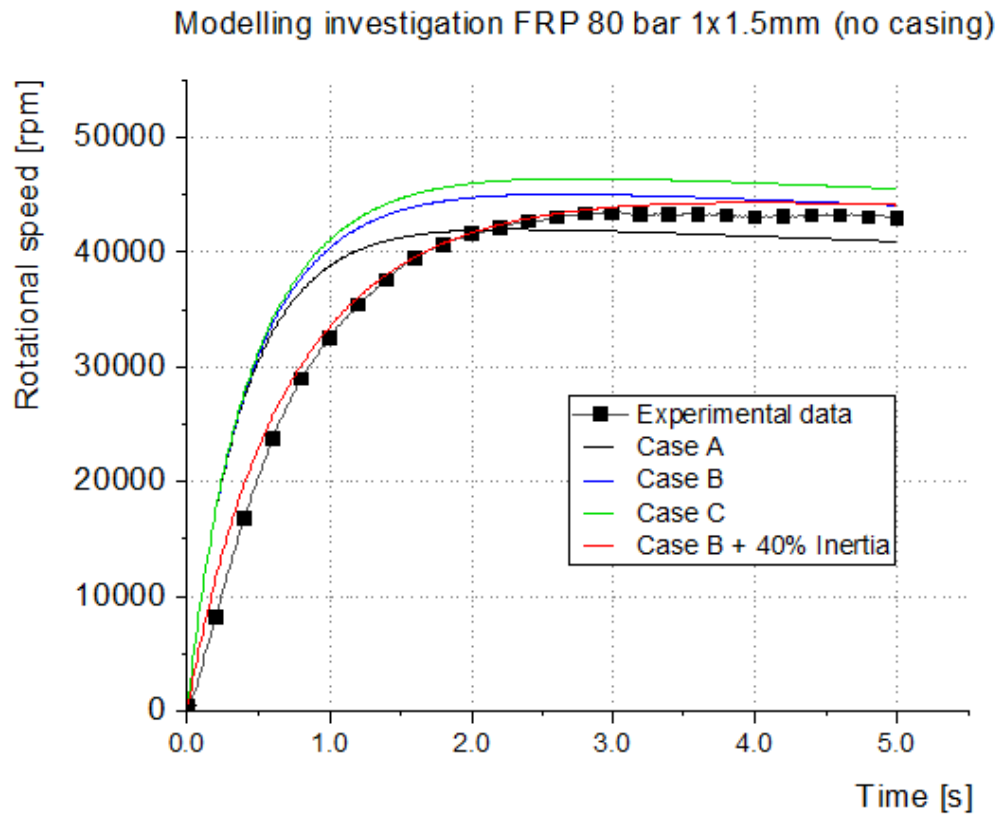


Figure 5.20 Model speed prediction tool experimental result match. No cover on free running Pelton*

*The reference for the case analyzed is available in Table 5.2

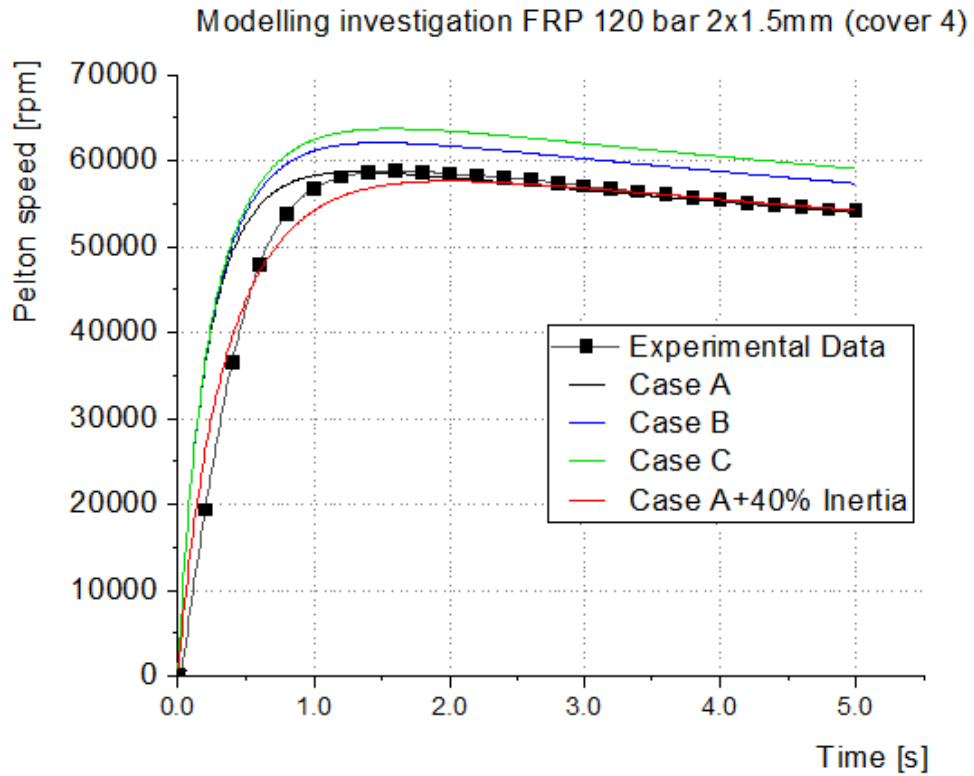


Figure 5.21 Model speed prediction tool experimental result match. Cover 4 on Pelton wheel*

*The reference for the case analyzed is available in Table 5.2

Using the optimised models some parametric studies were carried out. These were aimed to investigate the influence of parameters such oil accumulator charging volume, nozzle diameter, number of nozzles, charging pressure and Nitrogen chemical variations on Pelton speed. Firstly, the variation of the oil flow input was investigated, increasing then computationally the nozzle diameter and the number of nozzles. The increase of both parameter leads to a higher flow rate 46-50%; Figure 5.22 compares the improvement of max speed performances varying nozzle diameter and number of nozzles for different pressures. Each point represents the percentage improvement in maximum speed when a certain condition is applied (according to the legend). The time response is not considered.

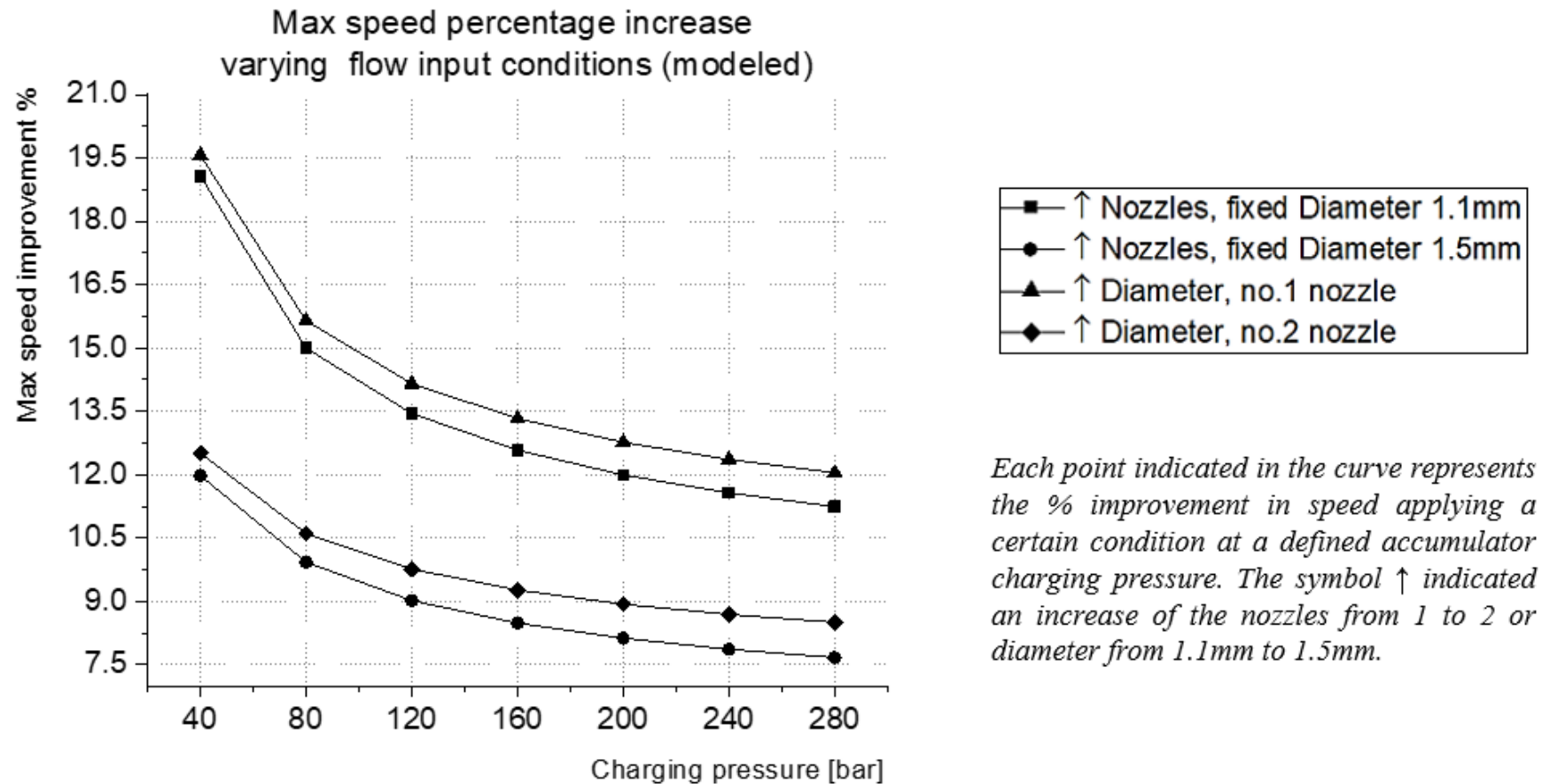


Figure 5.22 Modelled Top rotational speed percentage increase varying flow input conditions at the Pelton.

Figure 5.22 is important to show the influence of the impact of the higher flow rate to the final rotational speed. Each symbol in the curve represents the percentage of improvement when a certain change is applied. As an example, the square symbol trend identify the improvement obtained going from one to two nozzles but having fixed diameter of 1.1mm. This is done calculated for different accumulator charging pressures. The improvement is shown to be significant when 1.1mm nozzles are used increasing the number of nozzles from 1 to 2, or when 1 nozzle is used increasing the diameter from 1.1mm to 1.5mm. This corresponds always to a flow rate below 24 L/min. The relative cumulative improvement percentage shows that most of speed performances improvement varying the condition of nozzle number and diameter is achieved below 160 bar, leaving a poor improvement at higher pressures. The assumption is that although a higher flow rate should give higher input power to the Pelton, its benefit is shadowed by the higher windage losses above a certain pressure. Higher absolute speeds are however reached at high pressures.

The region of interest for a turbocharger is above 70,000 rpm and according to the experimental campaign on FRP, these speeds are reached using 2x1.5mm nozzles with pressure above 200 bar. Above this condition then, it is proved that increasing further the flow rate does not lead to a significant performance improvement. Summarizing, the main cause are the increasing windage losses which are directly proportional to the rotational speed, thus the flow rate. When more flow rate is injected, windage losses increase as well, and the potential speed improvement is slowed down until reaching a plateau. In Figure 5.23 an example of increasing windage losses is shown using FRP optimised model. The windage losses increase exponentially increasing the pressure and the nozzle number and diameter so ultimately the flow rate. The geometric intrinsic reason of this inefficiency lies as well in the choice of nozzle diameter which depends on optimization strategy in Pelton turbines and it is widely explained in Zhang [162] and a summary in Chapter 7 and APPENDIX C: Pelton wheel optimization is given as tool for future manufacturing and applications. In Figure 5.24 it is shown for a generic modelled configuration of the 32mm Pelton used in the research project how the increasing of the nozzle diameters leads to significant speed improvements up to 1.5mm; with higher diameters, the time response improves however the higher flow-rate does not justifies the percentage

improvement compared to lower diameter tested. For future work the optimum diameter can be identified for the specific application.

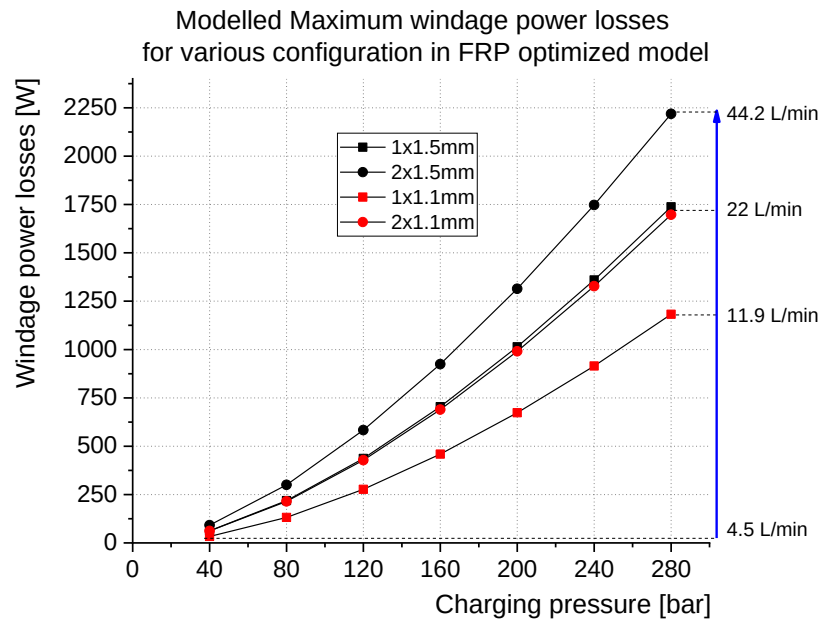


Figure 5.23 Modelled maximum windage losses for various configurations in FRP optimised model (no casing)

*Flow rate distribution increase is not equally proportioned displayed on the right vertical axes.

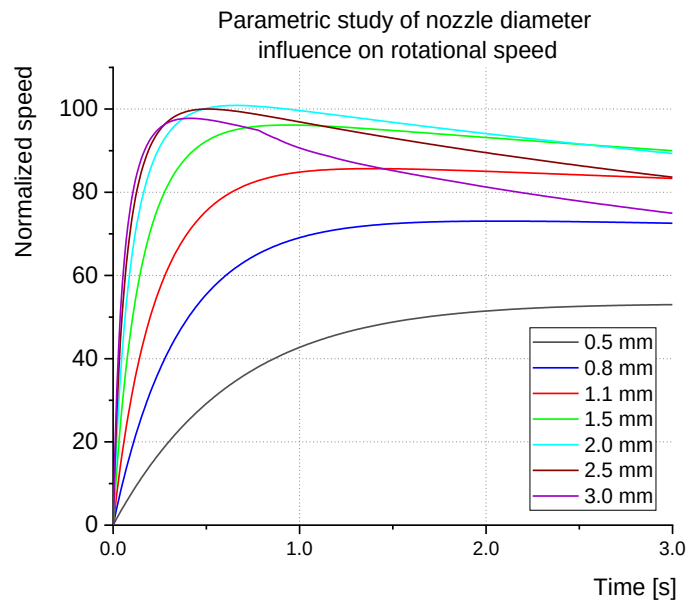


Figure 5.24 Modelled Parametric study of nozzle diameter influence on Pelton rotational speed

*normalised speed is defined as the rotational speed divided by the same factor for all the case studies (94487 rpm, which was identified as max speed simulated)

The variation of the isentropic coefficient gas value γ was investigated to determine whether the variation from the model for the ideal Nitrogen gas could affect the realistic case. At the hydraulic accumulator the expansion process of the gas when the solenoid valves are open can be consider adiabatic when the expansion is so slow that the heat exchange with the surrounding is negligible. For Nitrogen $\gamma=1.4$, however, if the expansion is not perfectly isentropic, the coefficient can vary and affect the pressure variation. The parametric study confirmed that the variation of γ would not significantly influence the pressure trend, hence the speed performances. At high pressure, considering an overestimated change of the $\gamma=1.4$ of $\pm 30\%$, the maximum difference in top speed is 2.6%, and the the time response is roughly the same.

As anticipated in the experimental campaign the accumulator oil volume charging could vary during the experiments maintaining the same other input conditions. The initial amount of oil charged into the accumulator influences the initial amount of volume available for the Nitrogen; this has an impact on the final volume of oil V_2 ejected from nozzle, thus pressure trend and the speed performances 5.9. The reason lies in the equation of isentropic expansion which depends on the initial volume V_1 of oil charged.

$$V_2 = V_1 + \int \frac{dV}{dt} dt \quad 5.9$$

The parametric investigation showed that among the oil charging volume tested, the 5% difference in experimental speed performances varying the parameter was confirmed from the model, therefore the initial assumption was confirmed through computational calculations.

5.5 Summary

This section highlights the main outcomes from the experimental campaigns on OTP and FRP to justify the final criteria for the new prototype designed and manufactured which is discussed in Chapter 6.

- Parametric experimental results of a hydraulic assist connected to a hydraulic accumulator discharge are presented in this chapter and the

summarised results represents an innovation in terms of optimization for a HAT prototype.

- The data analysis study confirms the reliability of the experimental results on a free running Pelton campaign. The fluctuation of results within 5% in maximum speed can be attributed to variations in hydraulic source mass.
- An encased micro-Pelton in a standard turbocharger plain-bearing housing in which high-pressure oil is injected leads to vibrations, leakages and hydraulic lock experiencing dramatically low rotational speeds.
- Adding a further draining source, optimise nozzle position and retrofitting ball bearings in the encased micro-Pelton leads to improvements from 14.5% to 47%.
- The encased Pelton improved design absolute maximum speed lies always below 20,000 rpm.
- Estimating the speed trend with a 0D computational model for a standard turbocharger plain-bearing housing retrofitted with a micro-Pelton can lead to an error of 76% because of the uncertainty around the geometrical data of the plain bearings.
- A free running Pelton-inertia disc assembly with 2x nozzles configuration shows a decrease in the maximum speed time response of 63% on average compared to 1x nozzle, slightly increasing the top speed.
- Increasing the diameter in a nozzle jet for a running Pelton from 1.1mm to 1.5mm at 80 bar charging pressure leads to +20% of maximum speed and -68% in max speed time response.

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- The vertical and horizontal positions optimization of the nozzle jet in a free running Pelton leads to increases in top speed of respectively +28% and +18.7%.
- The charging pressure of a hydraulic accumulator oil source for the Pelton is directly proportional to the final rotational speed and inversely proportional to time response. The maximum speed reached in the encased Pelton with an open draining is approximately 97,000 rpm with an absolute time response of 1.04s with 2x1.5mm nozzles at 280 bar charging pressure.
- Optimised speed performances investigation showed that the best option in terms of cover/casing for a 32mm Pelton to be implemented in a standard turbocharger for hydraulic assist with high-pressurised oil up to 46 L/min flow rate is to leave 40mm diameter in the peripheral direction and between 18mm and 36mm in the axial direction as well as for the draining opening.
- The optimised encased Pelton in the free wheel-inertia disc assembly showed +80% on average of maximum speed compared to a not optimised bearing housing of a hydraulically assisted turbocharger.
- A 0D computational model can be used to replicate a Pelton-inertia disc rotating assembly. The windage losses investigations shows that the introduction of oil in the Pelton without casing impacts increasing of +40% the inertia of the system and this is attributed to the dragging of the oil from the Pelton rotation.
- The 0D optimised computational model predicting pressure and speed trend using over 200 tests on a free running Pelton experimental campaign can match experimental results within 5% difference between experimented and modelled data. This variation can be attributed to oil charging volume variation.

- In a 32mm micro-Pelton, Simulated data showed that over 1.5mm nozzle diameter, an increment in nozzle diameter does not act significantly on the speed performances, increasing however with a direct proportional ratio flow rate thus windage losses.

6 Turbocharger Prototype Optimization

6.1 Introduction

The experimental investigation in Chapter 5 identified the constraints of the original unoptimised turbocharger prototype and its design faults. A new Turbo-Pelton Prototype (TPP) casing was designed to address the issues of low rotational speeds and oil overflow. The aims of the experimental campaign in this chapter were to: test the new TPP casing in a stand-alone configuration and to investigate its behaviour simulating realistic engine conditions. To achieve the latter, the TPP was mounted on a bespoke Turbo Pelton Stand Rig (TPSR). The experimental campaign is described highlighting the important outcomes regarding the time response and compressor boost. Discussions and future works following the experimental results are explained in the final Chapter 7 together with the comparison with the final Simulink model.

6.2 Turbo Pelton stand rig (TPSR) reasoning

6.2.1 Innovative Turbo Casing Design

The TPP is designed to have a bearing housing for accommodating the Pelton and ensuring no overflow occurs. The new casing is composed by 3 stainless steel pieces (Figure 6.1): the compressor section, the turbine section and the bearing cartridge. The distance between the compressor and the gas turbine is an important parameter to ensure the immediate retrofit of the assisted-turbocharger in an existing engine configuration. The detailed characteristics are described for each section below.

- Compressor section: this is designed in compliance with the required external diameter characteristics to respect the tolerances for the compressor wheel and its casing (provided from standard Garrett dimensional drawings, 86.15-96.16mm). The internal diameters are

designed to accommodate the selected bearing and its external shoulder. The standards for this can be found in bearings manufacturers catalogues (i.e. Schaeffler [197]). There are 3 pins which ensure concentricity when the turbine section is mounted before and secured with external bolts.

- Gas Turbine section: it is equipped with an O-ring groove, to ensure the seal with the compressor section. The internal diameters are designed to accommodate the second bearing and its shoulder.
- Bearing cartridge: it has the important role of supporting the Pelton and the bearings acting as a connection point with the shaft. Having the Pelton wheel and the bearings directly mounted on the shaft would have required the sequential mounting from just one side of the assembly, compromising the correct sealing of all the components.

Two nozzle holes are designed in correspondence of the jet circle diameter optimizing the vertical position of the jet as per design guidelines [162]. A water jacket to cool down the turbine section was included in the possibility of further implementation in a real-drive test. The overall dimension in length of the original casing was maintained; the diameter of the Pelton accommodation is 40mm and the axial length 19mm and the area of the open draining has same rectangular dimension to address the oil overflow directly into the sump when the HAT is in function. This new strategy compared to the previous design allows also to avoid the use of the Scavenge pump, optimizing and simplified the retrofit in the vehicle. On the cartridge an additional replaceable internal shoulder is provided to ensure support for one of the bearings, whereas on the other side a fixed one is in-built. This difference in the support depends on the inner dimensions of the wheel and the bearings (1mm) and on the one-way mounting nature of the assembly. The Pelton and the bearings are mounted on the bearing cartridge, the compressor and turbine section are secured together and finally the shaft is slid inside and secured as one piece with the bearing cartridge thanks to bolt fitting pushing against the outer diameter of the cartridge. An external compressor plate ensures the sealing of the assembly.

6 Turbocharger Prototype Optimization

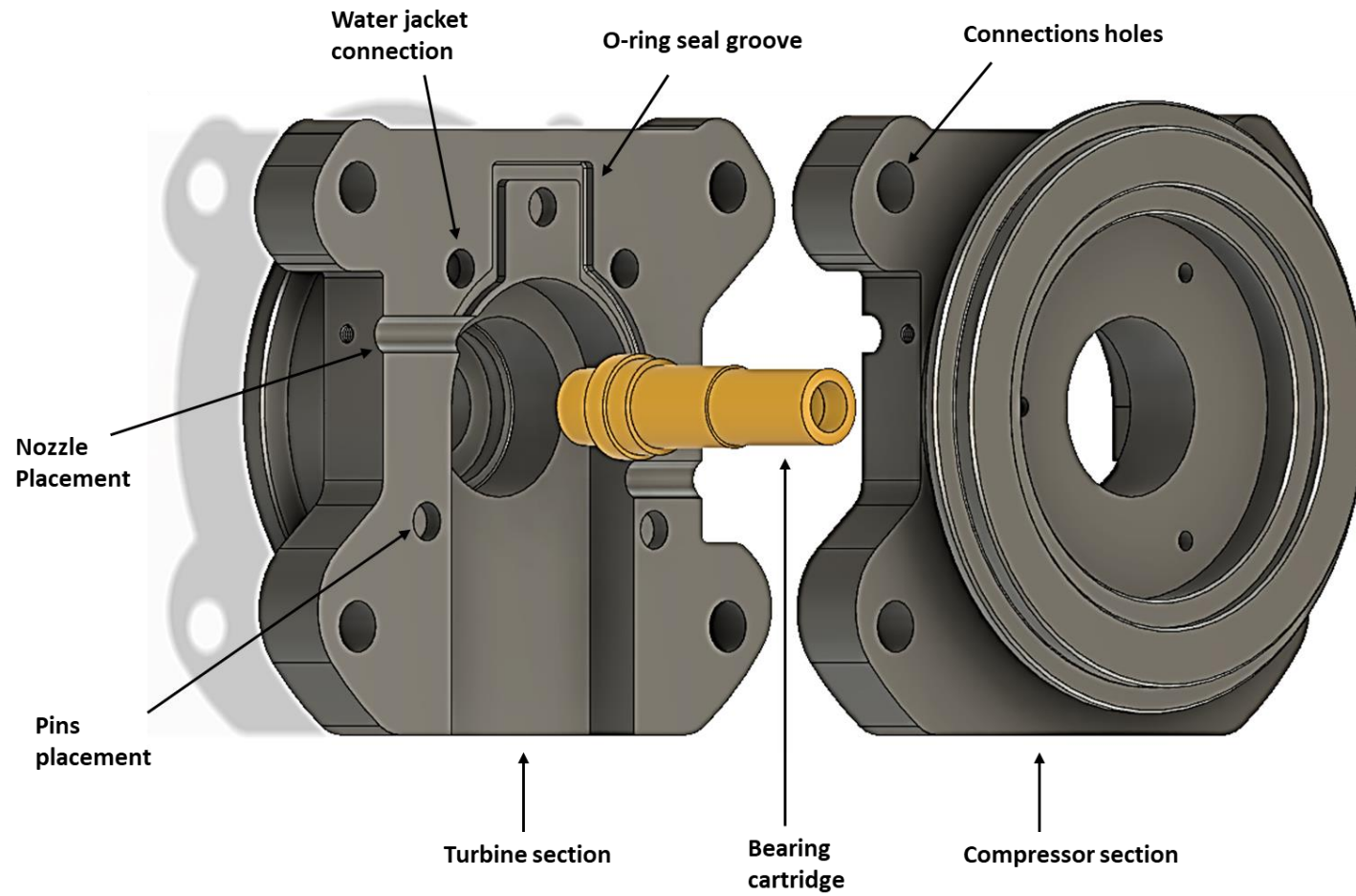


Figure 6.1 3D Representation of optimised HAT central casing

The configuration of 2x1.5mm internal diameter nozzles was selected. The reasons for this choice were firstly that the nozzle diameter of 1.5mm compared to 1.1mm leads to lower time response and higher maximum speed in the turbocharger performances. Secondly the 2x configuration leads to more than 60% faster response comparing to the one nozzle configuration. The higher flow rate involved does not represent an issue providing the open draining of the new configuration. The complete 3D modelled casing and bearing cartridge design is shown in Figure 6.1.

In the new design the plain bearings were replaced with more efficient rolling bearings. In these bearings the main load is transferred through balls or cylinders in rolling contact [186]. The main elements are the inner race, the rolling elements, the cage and the outer race. The rolling elements can be in different material (steel, Silicon Nitride, etc.) as well as the cage (steel, polyamide, etc.) depending on the application (high speed, high load, etc.). The cage has the important function of separating the elements so rubbing contact does not occur. Friction between rolling elements is considerably less than that encountered by sliding metal surfaces as in sleeve bearings. Ball bearings are usually mounted on shafts or in housings with interference fits. The interference fit of the bearing inner ring with the shaft is accomplished by pressing the former over the latter or using heating expansion to expand the inner ring, slipping it over the shaft and allowing to cool to accomplish the shrink fit. The tolerances to assembly all the sections in HAT are chosen to have a tight fit of the Pelton/bearings inner race on the bearing cartridge and a transition fit of the bearings outer race with the compressor and turbine section. Particularly, super precision single row hybrid angular contact ball bearings were selected for the HAT to stand the maximum load/speeds at maximum rig capacity [199][200]. These bearings can stand up to 130,000 rpm and they have inner and outer ring raceways displaced relative to each other in the direction of the bearing axis. This characteristic is designed to make these bearings able to accommodate simultaneously radial and axial loads [201]. Hybrid angular contact ball bearings have rings made of bearing steel and rolling elements made of bearing grade silicon nitride (ceramic). As ceramic balls are lighter and have a higher modulus of elasticity and lower coefficient of thermal expansion than steel balls, hybrid bearings can provide advantages like speed capability, reduced centrifugal and inertial forces,

frictional heat and mitigation of skid smearing damage and cage damage when subject to frequent rapid starts and stops. Usually, an anti-friction lubricant should be introduced to provide a film between the sliding and rolling surfaces, distribute and dissipate heat, prevent corrosion of surfaces and protect the parts from entrance of foreign matter [186]. Direct lubrication to bearings in the new design of HAT can be avoided as the ACBB bearing can ideally run without oil; furthermore, the oil ejected from Pelton buckets is in the direct trajectory to lubricate the bearings.

6.2.2 Test rig design condition

The test rig for the new HAT needed to be adapted to recreate the conditions of industrial testing to assess turbocharger performances; this test-rig configuration will be called TPSR (Turbo-Pelton Stand Rig) and it is shown in Figure 6.2. Its design is based on working temperatures, pressures and flow rates in the different sections. The automotive turbocharger operates under high-temperature flow conditions, given by the nature of the exhaust gas flow produced by the engine. The actual gas turbine of a turbocharger can reach up to 1050°C but they are usually in the range of 550-850°C [202] [203], however these temperatures are not safely achievable on an experimental lab test-stand. The tests are then carried out at lower temperatures scaling down the test-rig operating conditions. The maximum equivalent total-temperature was selected to be in the range 50-70 °C to avoid the condensation of vapour (Figure 6.3) at the rotor exit and to respect the temperature limit operation of the sensors used. The current experimental apparatus is set up with a conventional electrical air-heater to reach the desired turbine inlet temperature.

In a standard turbocharger gas test stand the results of the tests of the air source can be translated in real engine operation using conventional non-dimensional groups to reproduce the actual turbocharger operation. These conditions are referred as equivalent design conditions (or rig conditions) and they are experimentally set; the actual conditions corresponding to the real engine operation are obtained applying the principle of similarity between the turbocharger in standard operating condition (subscript act) and the test rig operating conditions (subscript rig) [204]. For this purpose, the equations 6.1 and 6.2 are usually used. To calculate Reynolds number, air density and

kinematic viscosity are considered at the respective temperatures [205]. In the equations, the effect of the gas constant R is considered small [206] and the Reynolds number effect of the turbine performance is considered to be of secondary importance [207]. This analysis can be suitably carried out when the only input the system is receiving is air; the final speeds in section 6.3.2 are the ones reached by Pelton aid only thus the following analysis is given as a guidance for future works and it cannot be correctly applied when the double input is applied.

$$\left(\frac{\dot{m}_t \sqrt{RT_0}}{P_0} \right)_{rig} = \left(\frac{\dot{m}_t \sqrt{RT_0}}{P_0} \right)_{act} \quad 6.1$$

$$\left(\frac{N}{\sqrt{RT_0}} \right)_{rig} = \left(\frac{N}{\sqrt{RT_0}} \right)_{act} \quad 6.2$$

6 Turbocharger Prototype Optimization

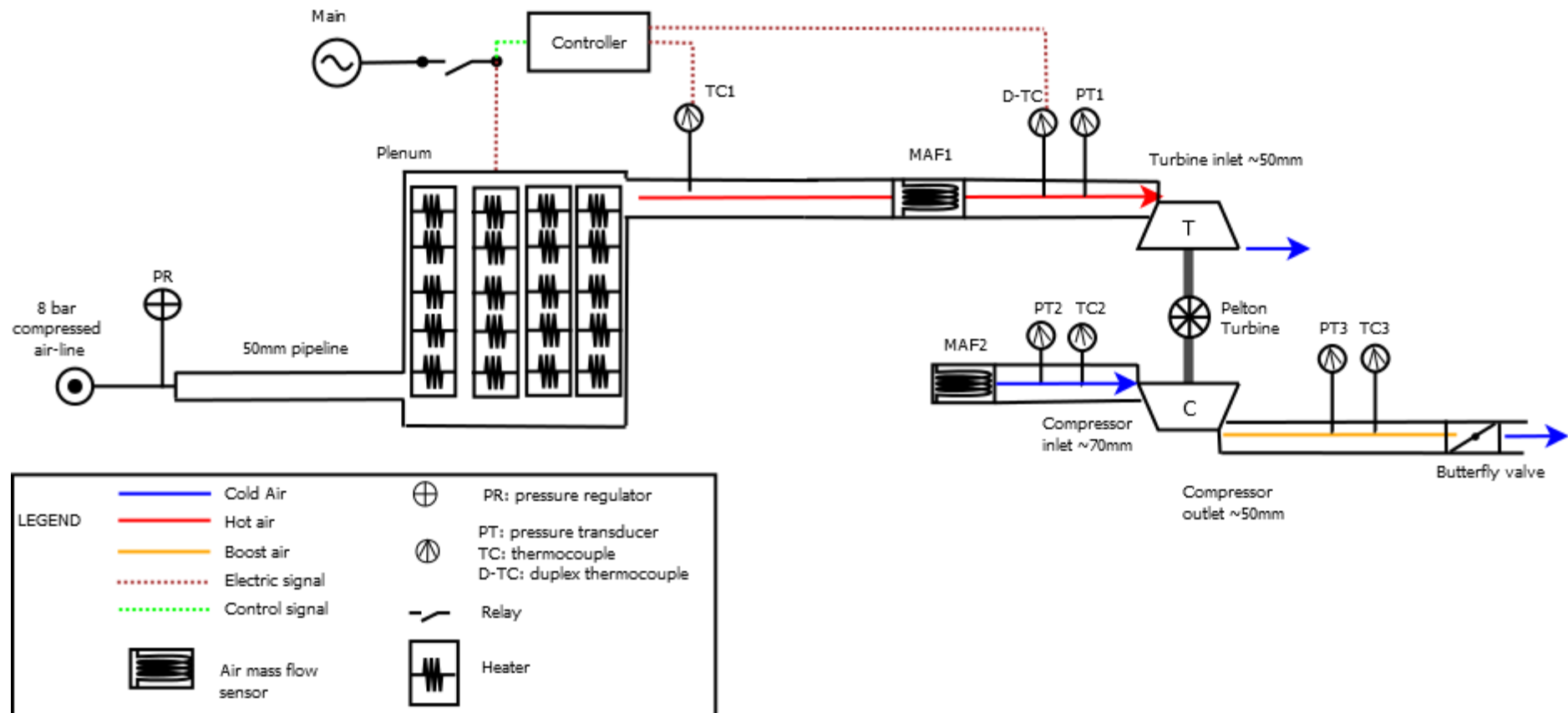


Figure 6.2 Turbo Pelton Stand Rig air-line schematic.

Electric connections are not exploded in details.

The design parameters for the TPSR were determined to select the sensors. To calculate the outlet temperature T_{2c} outlet at the compressor equation 6.3 is used. The assumption are to have a compression efficiency η_c of 0.8 [48], the pressure at the end of the compression $P_{2c}=1.1 - 3$ bar, the pressure at the beginning of the compression $P_{1c}=1$ bar and the temperature at the beginning of the compression $T_{1c}=288K$.

$$T_{2c} = \left(\frac{\frac{P_{2c}^{\frac{\gamma-1}{\gamma}}}{P_{1c}} - 1}{\eta_c} + 1 \right) T_{1c} \quad 6.3$$

The temperature T_{1t} at the inlet of the turbine is calculated with the constraints of maintaining turbine outlet temperature at least at $T_{2t}=278K$ to avoid ice formation. The T_{1t} is calculated in equation 6.4. This temperature is needed to design the heater needed for the inlet turbine pipeline. The assumption are that the expansion efficiency $\eta_t=0.85$, the end expansion pressure $P_{2t}=1$ bar, the initial expansion pressure $P_{1t}=1-3$ bar and an the end expansion temperature $T_{2t}=278K$.

$$T_{1t} = \frac{T_{2t}}{1 - \eta_t \left(1 - \frac{P_{2t}^{\frac{\gamma-1}{\gamma}}}{P_{1t}} \right)} \quad 6.4$$

The results for the variation of the temperature T_{1t} at the inlet of the turbine and T_{2c} at the outlet of the compressor are summarised in Figure 6.3. These results are parametrised varying the expansion and compression ratios of the devices. The turbine has a fixed expansion ratio, which can vary depending on the turbine. The thermal power of the heater required at the turbine to have air inlet temperature T_{1t} is shown on the right (ambient temperature is considered 288K). The model GTX3076R Gen II, which is similar to the frame used for the prototype, has a maximum expansion ratio of 2.8. The calculated temperatures

match the results from Karamanis at imperial college [203] which rig and parameters were inspired from.

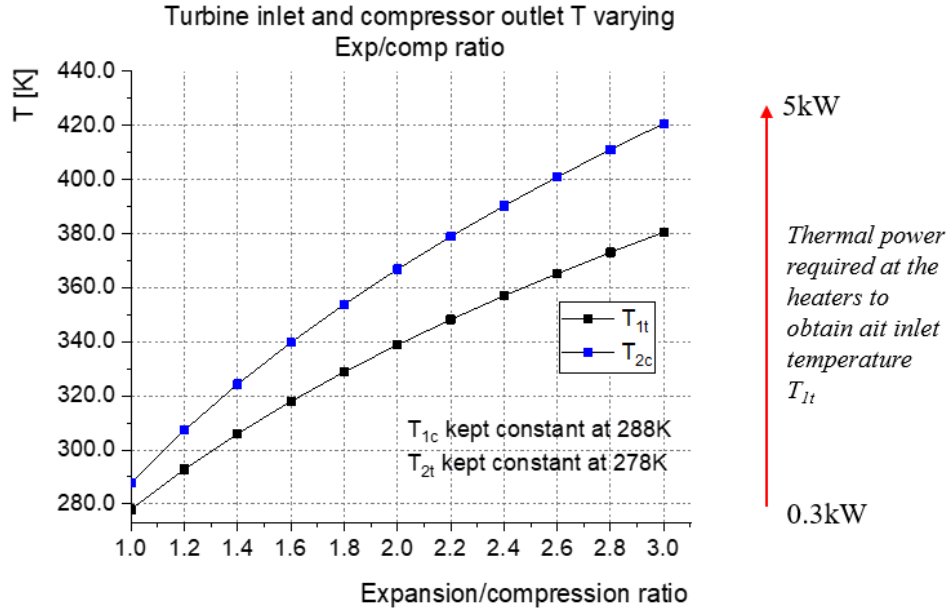


Figure 6.3 Turbine Inlet and Compressor Outlet Temperatures varying expansion and compression ratios.

Thermal power of the heater required at the turbine inlet to obtain T_{1t} is shown on the right (not in scale). Ambient temperature is considered 288K

Considering the properties of the air [208] and the error associated with neglecting the combustion products no more than 2% [209], the order of magnitude of mass flow rate flowing through a turbocharger can be calculated in 6.5 using parameters such as engine volume $V = 2L$, engine speed $N = 2000 \text{ rpm}$, air density at 900°C $\rho = 0.3 \text{ kg/m}^3$, stroke and volumetric efficiency $\eta_v = 0.9$. The value N_s is equal to 2 for a 4 stroke engine.

$$\dot{m}_a = \frac{VN\eta_v}{N_s\rho} \frac{10^{-3}}{60} = 0.1 \text{ kg/s} \quad 6.5$$

The turbine inlet in the prototype in L4 lab at the University of Nottingham is fed with a bespoke compressed air-line connection. The air source is the compressor Abac Genesis 15 kW and this line is capable of delivering a maximum of 0.047 kg/s (6.35 lbs/min). The complete turbocharger map goes

ideally up to 40 lbs/min, meaning the replicable conditions in lab are framed just in the first range of operations. For this reason, the experimental investigations will be focused on studying the feasibility of the prototype, the time response and the compressor boost achieved rather than detailing mapping the turbocharger prototype.

An air heater was used to achieve the calculated T_{1t} . The thermal power to preheat the air is calculated using the flow rate mass of the compressed air and the temperature difference at the turbine as per equation 6.6 and the results are available in Table 6.1. which summarises the air-line parameters expected in the different section/components of the rig

$$\dot{Q}_{air} = \dot{m}_{air} c_p (T_{2t} - T_{1t}) \quad 6.6$$

The suitable air mass flow sensor is selected to detect the flow rate required and the details are available in section 6.2.3.

<i>Parameter</i>	<i>Unit Measure</i>	<i>Turbine Inlet T_{1t}</i>	<i>Turbine outlet T_{2t}</i>	<i>Compressor Inlet T_{1c}</i>
Pressure	[bar]	1-3	~1	~1
Air flow-rate	[kg/s]	0-0.048*	/	0-0.13
Temperature	[K]	288-380	278	288
Air-Heater Thermal power	[kW]	0.3-5	/	/

Table 6.1 Turbo-Pelton Rig air-line parameters in KERB-HAT testing

*compressed air input

6.2.3 Rig design and components

This section explains in detail the components of the rig and their selection according to the requirements in 6.2.2 for KERB-HAT experimental campaign. The experimental setup of KERB-TPSR is based upon the standard configuration used for testing standard turbochargers in different operating conditions and it takes inspiration from a similar test rig already proven to be reliable in a research project at Imperial College [203]. All the nomenclature used is referred to Figure 6.2 and the real rig is shown in Figure 6.4. This air-line is in addition to the rig configuration shown in Chapter 3 and it is split into three sections: turbine inlet, compressor inlet and compressor outlet. The

thermocouples nomenclature is TC followed by the number of the appropriate section.

In the Turbine-inlet section the components are: a compressed air source, a pressure regulator, the pipework, a plenum, the heaters, the thermocouples TC1 and D-TC, the air mass flow rate sensor MAF1 and the pressure transducer PT1. The compressed air is delivered from the compressed air-line in built source from the compressor Abac Genesis 15 kW (maximum air delivery 8 bar, 40L/s, 0.048 kg/s) passing through a pressure regulator and into the plenum through a 50mm pipeline. The plenum accommodates 4 rectangular finned air heaters 325x40x70mm, 1000 W, 230 V and a 10mm non-ceramic millboard thermal insulating sheet is shaped along the internal sides of the plenum to prevent undesired heat transfer towards the external environment. A safety control is designed around this system to avoid overheating of the pipework and sensors nearby. The control switches off the heaters when the TC1 exceeds the set safety temperature which is chosen to be 20% less of the lowest temperature tolerated by the sensors. The HFM-5 BOSCH air mass flow sensor was selected for MAF1 and MAF2 to detect the air flow expected in the turbocharger; the accuracy of the sensor is $\pm 3\%$. The duplex thermocouple D-TC (that detects T_{1t}) has the double function of acquire data and provide the set temperature for the PID heating power control. All the thermocouples in the rig are T-type with accuracy of $\pm 1.1^\circ\text{C}$. The compressor inlet section is equipped with air mass flow sensor MAF2, the pressure transducer PT2 and the thermocouple TC2 (that detects T_{1c}). The compressor outlet is equipped with pressure transducer PT3 and thermocouple and with a butterfly valve which restricts the flow and creates the boost effect. The valve has notched indicators to ensure reliability of the position; the indicators were labelled from 0 to 9 and the shakedown revealed that significant position to study were 0 (fully open), 5 (55% closed) and 7 (77% closed).

The data acquisition system LabView VI was modified to ensure the correct reading and data acquisition of the additional sensors.

6 Turbocharger Prototype Optimization

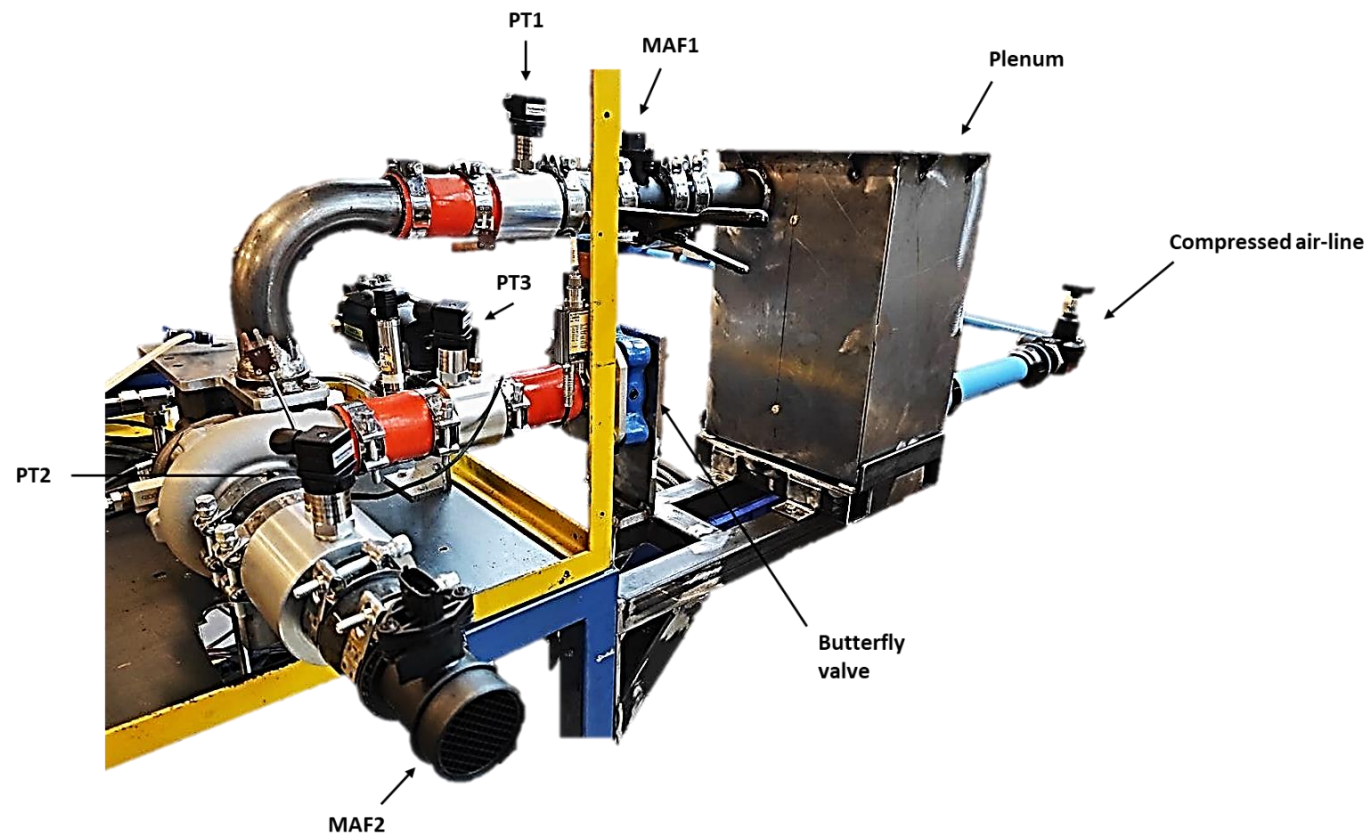


Figure 6.4 TPSR lab configuration.

6.3 TPSR Experimental investigations

6.3.1 Shakedown and test methodology

The preliminary shakedown of the experimental campaign aimed to:

- Test TPP in stand-alone configuration
- Calibrate the sensors
- Test pipeline pressure losses
- Test turbine correct rotation
- Verify temperature safety controls

Firstly, the TPP was tested in the hydraulic-assist configuration in Chapter 3. The results are available in Figure 6.5 and they are used as comparisons for the HAT-TPSR campaign.

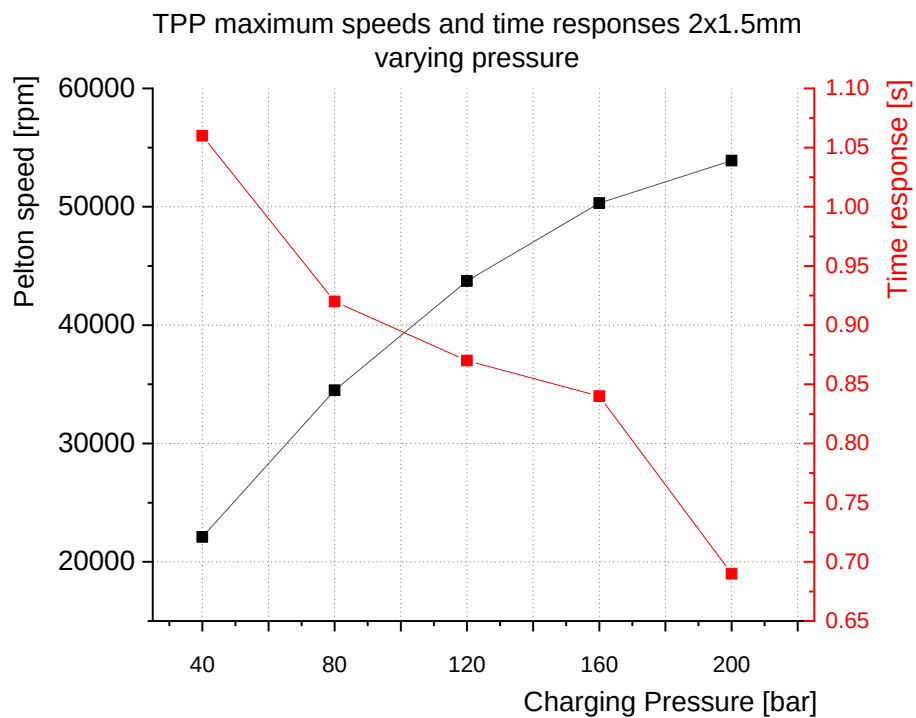


Figure 6.5 Optimised Turbo Pelton prototype maximum speed and absolute time responses in a stand-alone configuration (no air input)

To calibrate the thermocouples and the pressure transducers a voltage power supply was used to replicate the input correspondent to the set temperature and verify the reading was truthful. The air mass flow sensor calibration curve was extrapolated using a power fit curve adapted from the supplier data (Figure 6.6).

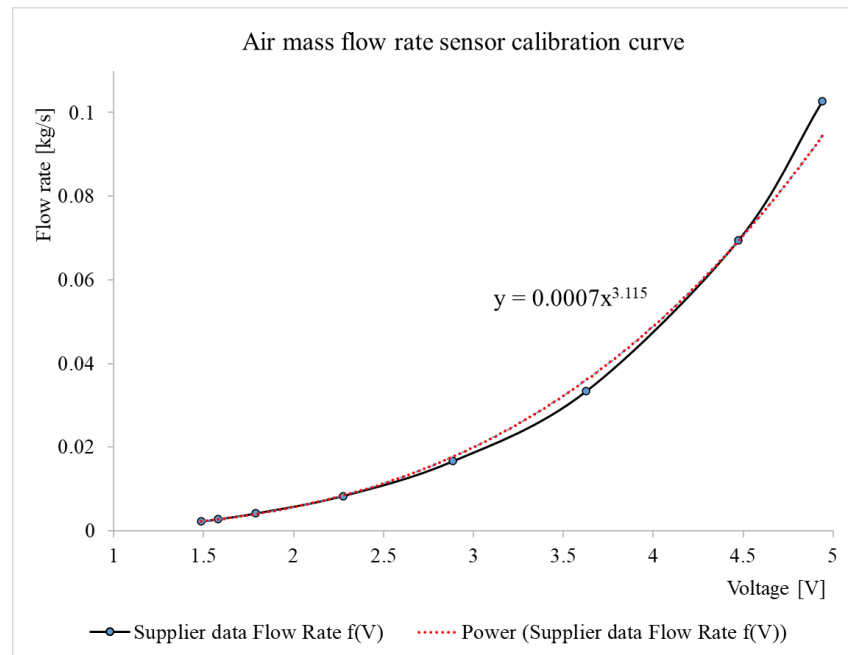


Figure 6.6 Calibration curve for air mass flow rate sensor BOSCH MAF-H5

The black trend is built from the supplier experimental calibration and the red line is the power fit curve extrapolated in excel

The turbine was tested activating only the air source and confirming that the pressure losses along the pipeline would not significantly impact the rotation. The maximum speed achievable with the air source only was between 15,000 rpm and 18,000 rpm depending on the compressed air flow rate available. The temperature control is carried out with a PID control (example in Figure 6.7). When the set temperature at D-TC is reached, power impulses are delivered to keep the value as constant as possible. The safety temperature control condition is instead carried out with an on-off control on TC1. A voltage power supply was used to replicate the identified temperatures; the control showed appropriate response respecting both conditions of switching off the heaters when the TC1 safe set temperature (90°C) was reached and maintaining the D-TC. The overshoot B in Figure 6.7 was around +7° C; considering the D-TC set to be 60° C, it meant that the maximum T of D-TC was 67°C with TC1 reaching a maximum of 80°C.

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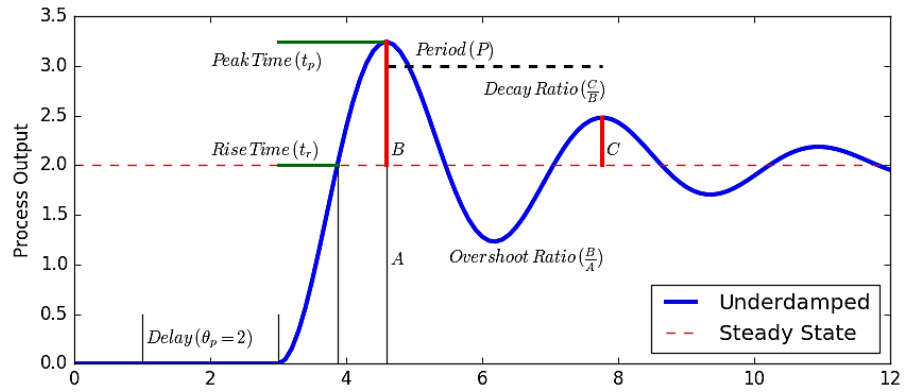
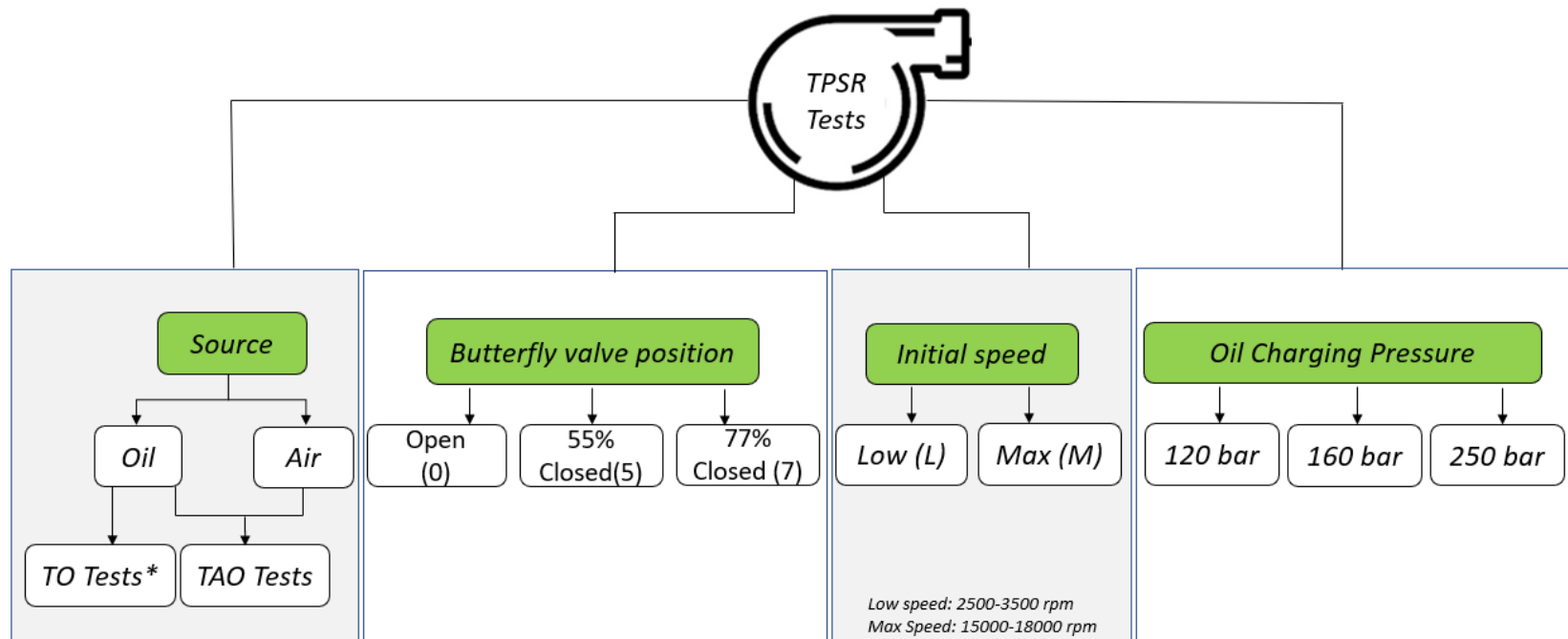


Figure 6.7 PID control example

The detailed configurations of TPSR tests are shown in Figure 6.8 which describes how to interpret the nomenclature of the generic test according to the combination of the parameters. The charging pressure has been pushed occasionally to 250 bar to explore the test the extreme performances of the rig.

6 Turbocharger Prototype Optimization



Test configuration example TAO-5L-120: air and oil source, butterfly valve half open, initial rotational speed of the assembly low, oil charging pressure 120 bar.

* The TO tests have zero initial speed of the turbine thus no denomination is indicated after the butterfly valve opening %

Figure 6.8 TPSR tests configurations and nomenclature

6.3.2 Experimental results on Speed, Time response, Compressor Boost and Flow-rate

The final purpose of the prototype subject of the research project was to determine if the device can be a valid alternative to mitigate the turbo-lag issue in turbocharger dynamic. The experimental campaign in TPSR collected results from over 100 tests to investigate absolute time response, speed and compressor boost. Different configurations were tested (Figure 6.8) and the data analysis methodology is the same described in Chapter 5.

The preliminary tests were carried out using the maximum air source input at the gas turbine giving an initial rotational speed on average of 15,000 rpm. This speed does not represent the standard condition of an assisted-turbocharger start, but it was useful to understand that the initial air flow-rate given at the gas turbine does not influence the maximum top speed achieved with the hydraulic assist. The comparison was carried out comparing TAO-7M-160 and TAO-7L-160 and the results is less than 0.15% in speed difference. The time response is indeed not influenced being maintained at 0.83s. The gas turbine is always in motion (2500-3500 rpm) thanks to the exhaust gases; the assist starts its operation in this range. For this reason, it was decided to carry out the main comparison test using an initial lower speed.

A comparison is carried out considering the maximum speed and time response achievable introducing the low-speed air input at the gas turbine (TAO-L tests). Compared with the test just with oil (TO tests), the introduction of the air as an input source at the gas turbine does not have a significant impact on speed as the maximum speed difference always stays within the $\pm 5\%$ (Table 6.2). A more noticeable impact is observed on the time response. When the butterfly valve is closed at 77%, the introduction of air as input source causes an increase of time response of at least 10%. These values are still comparable values in terms of absolute time response for an assisted turbocharger considering an electrical assisted turbocharger. The potential of the HAT compared to EAT will be carried out in details in Chapter 7.

The influence of the compressor boost on the rotational speed was investigated using the results in Table 6.3. An increased boost is achieved decreasing the degree of opening of the butterfly valve. The results show that the valve closed at 77% corresponded to the increase of the compressor boost up to 16% starting

from ambient pressure. The time response increases as well from 6% to 16%, determining a negative impact of the boost on this parameter. This can be explained considering that the air density is increased subject to a higher pressure. It is assumed that the denser air causes an increased inertia effect on the assembly, reducing the time response. The higher boost increases instead the top rotational speed up to 11%. The assumption to explain this is attributed to the reduced work at the compressor wheel when more boost is produced in TPSR specific setup. The compressor is assumed indeed to be less subject to pumping losses as the input airflow is restricted from the limited air source. To calculate the work at the compressor the experimental data of inlet and outlet temperatures and the flow rate going through the compressor were considered. This calculation was however not possible in the case of valve closed at 77% as the sensor was subject to excessive noise. For this reason, the assumption previously stated could not be experimentally verified. It is recommended that in future works a plenum is added at the compressor outlet to avoid sensors disturbance given the proximity of the latter to the butterfly valve. In this study the work at the compressor was not calculated as the boost pressure range explored is not representative of the realistic efficiency of a real operating turbocharger. The main purpose of the campaign was indeed to test the feasibility of the HAT in early prototype stage.

The maximum boost of 1.31 bar was achieved at highest pressure 250 bar and closing the butterfly valve at 77% with 2x1.5mm nozzles. The real boost in a turbocharger goes up to 2 bar or slightly above depending on the model, however these tests are useful to understand the behaviour of the turbocharger under boost conditions. The minimum time response achieved was 0.64s with 250 bar oil input with a maximum speed of 63,000 rpm. The final maximum speed are in the same range of the TPP tested in the prototype as well as the time responses (Figure 6.5).

6 Turbocharger Prototype Optimization

Test	Max speed [rpm]	Time response [s]
TO-5-160	53120	0.76
TAO-5L-160	51480 ↓ -3%	0.79 ↑ +4%
TO-7-160	56790	0.75
TAO-7L-160	56410 ↓ -0.7%	0.83 ↑ +10%
TO-5-120	46310	0.81
TAO-5L-120	45680 ↓ -1.4%	0.83 ↑ +2%
TO-7-120	51210	0.87
TAO-7L-120	51070 ↓ -0.3%	0.98 ↑ +11%

Table 6.2 Air input influence on TPSR time response and maximum speed

Test	Max speed [rpm]	Time response [s]	Compressor boost
TAO-7L-120	51070 ↑ +11%	0.98 ↑ +16%	1.202 ↑ +15%
TAO-5L-120	45680	0.83	1.158
TAO-0L-120	45180	0.82	1.024
TAO-7L-160	56410 ↑ +8%	0.83 ↑ +6%	1.247 ↑ +16%
TAO-5L-160	51480	0.79	1.21
TAO-0L-160	51840	0.78	1.046

Table 6.3 Compressor boost influence on speed and time response of TPSR

Finally for the main tested charging pressure with configuration TAO-5L, Figure 6.9 put in relation the boost and flow rate measured at the compressor inlet. These are within the range expected for the standard turbocharger used for light-duty engines of 1.2L.

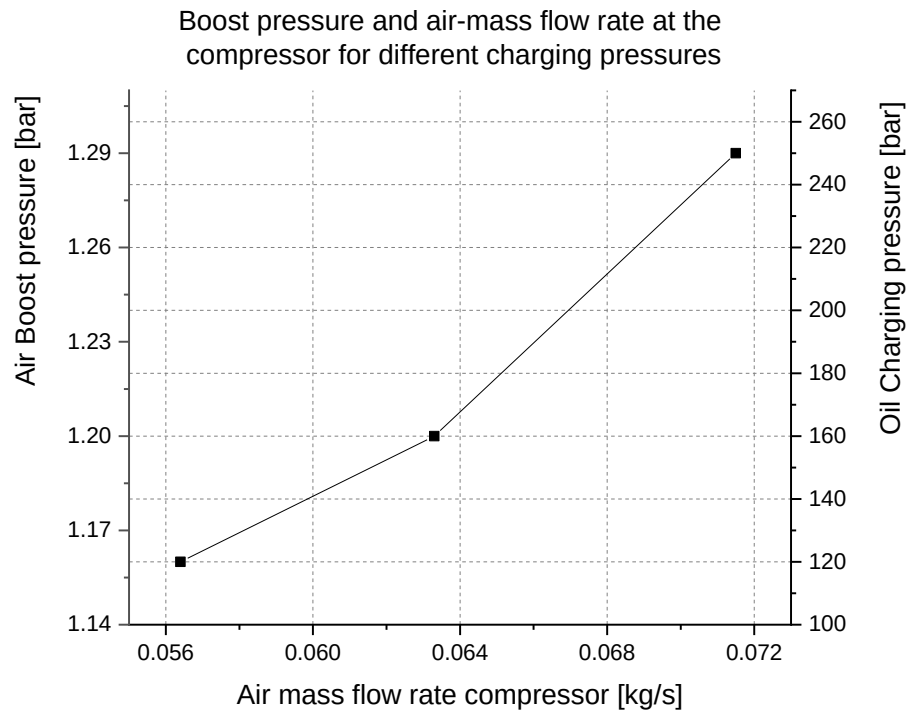


Figure 6.9 Boost pressure and air-mass flow rate at the compressor for different charging pressure in TPSR

6.4 Summary

This section highlights the main outcomes of the experimental campaigns on TPSR.

- An innovative hydraulic assisted turbocharger (HAT) including an open draining, angular contact ball bearings and optimised 2x1.5mm nozzles experimentally tested revealed interesting potentialities in terms of maximum speed and time response to tackle turbo-lag issue. The parametric study of rotational speed varying accumulator charging pressure and boost pressure are for the first time presented in literature.
- A Bearing Cartridge assembly is the most efficient way to avoid vibrations and ensure the sealing of the components during assembly and functioning operations when introducing a micro-hydraulic turbine in the turbocharger casing.
- To ensure safe and reliable testing, avoiding ice condensation of the air at the inlet of the turbine, a turbocharger stand rig compressed air source

must be kept at a constant temperature of 60°C using a 4kW air-heater and a PID control.

- A butterfly valve allows to replicate the pressure boost caused by real driving conditions in the range of 1.05-1.31 bar with a fictitious compressor work of 400-600W. The excessive closing of the valve can cause significant noise in the sensors which prevents to have reliable readings on pressures and temperatures.
- With compressed air source at the turbine inlet, the maximum rotational speed achievable was 15,000-18,000 rpm without the hydraulic assist.
- The initial gas turbine air input in the turbocharger test stand has no significant impact on maximum final rotation speed when the hydraulic assist is introduced. With the butterfly valve closed at 77%, the final speed difference reached with the hydraulic assist in the HAT with an initial speed of 15,000 rpm is just 0.15% higher than when started at 3000rpm.
- The butterfly valve closed at 77% translates in slightly higher boost compared to 55% closure, however, the time response increases of +10% introducing the compressed air source at the turbine.
- The introduction of a Pelton turbine in a bespoke central housing rotating assembly of a Garrett turbocharger GTX2867R GEN II allows a time response in the range 0.98-0.64s with an input oil hydraulic pressure of 250-120 bar.
- In the HAT the maximum performance was achieved injecting 250 bar of pressurised oil and it consisted in a maximum rotational speed of 63,000 rpm reached in less than 0.65s creating a boost of 1.31 bar.

7 Discussion

7.1 Introduction

In this chapter the overall experimental results are used to frame the HAT prototype in a potential application in heavy duty vehicles.

Initially the prototype evolution is described focusing on the improvement compared to the concept idea. The comparison with computational prediction showed that two main models for the windage losses matched the maximum speed results, whereas the time response is underestimated 12%. The Pelton turbine optimization process is shown, highlighting how the number of buckets and their geometry are essential parameter to design to improve the overall performance. The application for the HAT is thoroughly discussed highlighting advantages and challenges. The performance and requirement characteristics of the main competitor of the HAT, the electrical-assisted turbocharged (EAT), show its limitation in the use of heavy-duty and off-road machinery vehicles, whereas the hydraulic source can be a valid alternative. Finally, a brief paragraph on alternative application for the HAT shows the potentiality of this device for other purposes.

7.2 HAT prototype evolution and modelling

The TPSR configuration described in Chapter 6 was built to assess how the optimised HAT would behave in engine like situation. The focus was on the impact the boost can have on the assembly dynamic. To replicate HAT conditions, the computational model was modified to address the following:

1. *Increased Turbocharger assembly inertia*: it was evaluated combining 3D analysis techniques and acquisition of empirical data. The overall rotating mass was 466g with an inertia of $9.9653 \cdot 10^{-5} \text{ kgm}^2$.
2. *High-performance Angular Contact Ball Bearings (ACBB) axial friction losses*: these losses were added in the model considering the maximum air input available from the external compressor in L4 laboratory. An overall maximum force $F_{\text{max}}=0.98\text{N}$ was computed in addition in the gas

turbine. This value is not significant causing just a 2% in rotational speed reduction.

3. *Additional Compressor/turbine windage losses*: this was added in the windage loss equation and it explain the lower speed in the HAT compared to FRP.

The HAT was optimised geometrically and fluid dynamically; compared to the original unoptimised HAT prototype described in Chapter 5, experienced an average maximum speed increase of 67%. Compared to the FRP design, however, the maximum speed was 22% lower because of the additional windage losses of the compressor/turbine. These are computed in the modelling as a combination of the equations in Chapter 4. The gas turbine windage losses are also considered as the air driving the turbine has the only purpose of replicating real engine conditions of starting of the transient event at $\omega > 0$ and the Pelton drags all the assembly during the rotatory phenomenon. To replicate windage losses in compressor and turbine wheel, two models are used and compared: Equation 7.1 from Buckingham [194] is a preliminary approach which does not consider the casing of the compressor/turbine; Equation 7.2 from Etemad [152] is a readaptation of the basic studies on windage losses which takes in account the casing influence with the ratio between the shrouding air gap s and the wheel diameter r .

$$\text{Buckingham } T_{w,c/t} = \left(1 + 590 \left(\frac{L_{c/t}}{D_{c/t}}\right)^2\right) \left(\frac{60}{2\pi} \omega\right)^2 2D_{c/t}^5 \rho_{air} 10^{-6} \quad 7.1$$

$$\text{Etemad } T_{w,c/t} = 0.0311 * \left(\frac{s}{R_{c/t}}\right)^{-0.25} \mu_{air}^{0.25} \rho_{air}^{0.75} \left(\frac{60}{2\pi} \omega\right)^{7/4} R_{c/t}^{9/2} \quad 7.2$$

The model was parametrically investigated comparing the experimental results to determine the best-fit. A preliminary unoptimised model comparison is shown in Figure 7.1. The modelled speed slope is lower than the experimental data suggesting the inertia of the system was underestimated. The experimental maximum speed is higher than the computed one suggesting windage losses are

overestimated. The parameters in Equation 7.1 proposed by Buckingham are experimentally determined and calibrated for a specific geometry. A coefficient k_w is then introduced to consider that the shrouding in standard rotating wheels/gears mitigate the windage losses is valid for evaluating windage losses of the gas turbine/compressor rotating free in air without being encased [143]. [156] shows that an axial or peripheral shrouding can reduce the windage losses at lower speed than the prototype (20k rpm) up to 70%. Using Etemad approach [152] the shrouding is already considered within the term s/r . The air gap s between the wheel and the casing is not specified in the geometrical characteristic of the turbocharger adopted in this thesis, however, the set clearances of the peripheral shrouding typically vary between 0.03mm and 0.5mm depending on the turbocharger model. An empirical value was then chosen to fit the experimental results. In the Buckingham [194] adapted model the shrouding coefficient was considered $k_w=0.7$. This translated in a mitigation of 30% in windage losses, well within the expected range [156]. In Etemad [152] adapted model the coefficient k_w was considered 1, as the shrouding effect is already considered with the term s/r , and the value of peripheral shrouding to fit the experimental result was $s=0.4\text{mm}$. Either the Buckingham formula with $k_w=0.7$ or Etemad formula with $s=0.4\text{mm}$ can be used for this HAT configuration. The results obtained with these two windage losses models matched the experimental data with more than 95% precision regarding the maximum speed. The difference between the two model trend points is less than 0.3%, thus, in Figure 7.1, just one trend is represented with the nomenclature “Optimised Model”. This, was compared with experimental data at different charging pressures, namely 120 and 160 bar. At both charging pressures the predicted time response is 12% higher than the experimental one.

The optimised model can reproduce the experimental data with some limitations due to various reasons. Firstly, the nature of the windage losses equation. Buckingham [194] highlighted how, different designs require different coefficients. The realistic windage losses are indeed usually computed from the manufacturer, which provides empirically data to evaluate the specific configuration. Another limitation of the model is related to the intrinsic discrepancy in the hydraulic efficiency between model and prototyped Pelton

turbines [162]. The reason can be attributed to the difference in the modelled Reynolds number and the real experimental one.

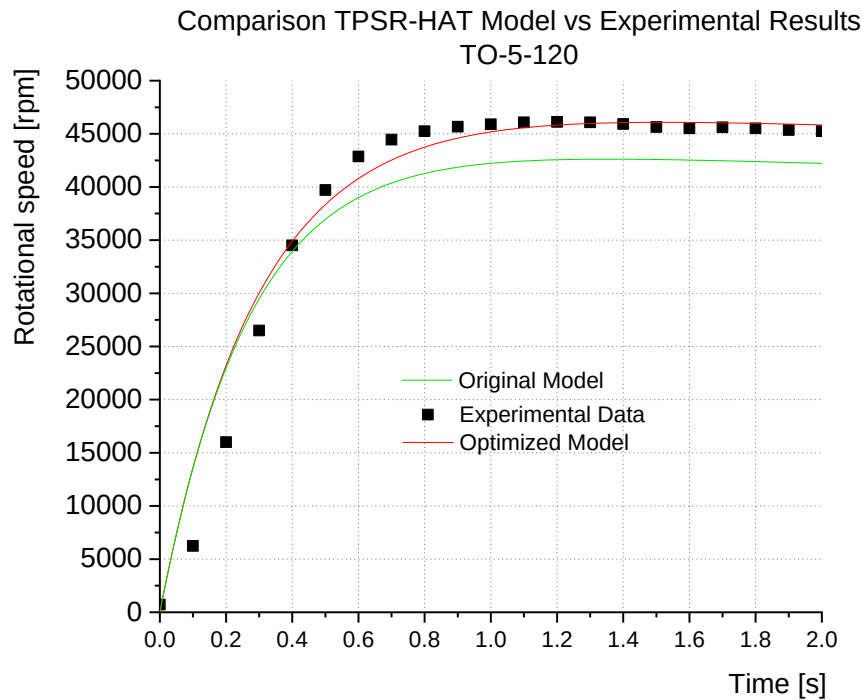


Figure 7.1 Comparison Computational model with experimental results of TO-5-120 tests in TPSR*

*The Model TPSR optimised is obtained with Buckingham formula introducing the shrouding coefficient $k_w=0.7$ or Etemad adapted model with $k_w=1$ and $s=0.4\text{mm}$. In experimental points are skipped at 10pt although the acquisition rate is 100Hz.

Through the optimised model it was possible to prove the higher degree of windage losses experienced in HAT-TPSR compared to FRP. The windage loss in the Hydraulic-assisted turbo is an order of magnitude higher than the Free running configuration; the optimised model computes that the maximum total windage loss is 1600 W in HAT compared to the 600 W in FRP at the same input conditions.

To understand the behaviour of the prototype for heavy-duty application, a turbocharger assembly to model could be the Garrett GTX4508r which is suitable for engines with displacement up to 8L. The optimised model can be considered a first approach, however the precision in computing geometrical characteristics of the turbocharger is essential for reliable results. For these reasons, here just few considerations are given to understand the potential

differences and qualitative prediction but the complete computational modelling is recommended as part of future works on the project. A larger size turbocharger could imply an increase flow rate to reach the desired power output; the introduction of more oil could however counteract the correct functioning of the rotating machinery, increasing the relative windage losses up to 40% doubling the flow rate involved [156]. Another interesting investigation using the information in section 7.3 and in APPENDIX C: Pelton wheel optimization is to optimise the Pelton wheel according to the power and flow rate requirements without incurring in over-designing the geometrical characteristics.

7.3 Pelton wheel optimization

The hydraulic wheel mounted in the prototype was selected during a previous work on KERB [49]. The guidelines for optimised Pelton can be found in Zhang [162] and the detailed procedure is shown in APPENDIX C: Pelton wheel optimization. A summary is given in this paragraph regarding the main geometrical characteristics of an efficient Pelton which are:

- The bucket width B should be approximately 3 times the jet diameter d_0
- The pitch circle diameter D_m should be greater than $3B$.
- The number of buckets is defined as $N_b = 15 + 0.5 \frac{D_m}{d_0}$
- The liquid injected into the bucket should be exposed only to a continuous and smooth curve with no sudden changes in gradient in any direction.

The hydraulic turbine in KERB-HAT should have had 24 buckets however it was manufactured with 12 buckets, 11mm of width and an external diameter of 32mm to speed up the testing process and because of manufacturing limitations. Despite the proposed optimization methods, the turbine tests in an experimental environment remain the most credible and reliable way to verify and optimise the hydraulic design of a prototype wheel [162].

7.4 KERB-HAT automotive application

This paragraph provides the tools to discuss the best application for the Hydraulic-Assisted Turbocharger; the discussion is based on the available Literature Review in Chapter 1 and Chapter 2 and on the experimental results of HAT in Chapter 6. The parameters to attention in turbocharger real-engine operations are the speed, compressor boost and the air flow rate. The experimental range of boost in KERB-HAT was 1.05-1.31 bar; to understand the potentiality of the device, these absolute values need to be framed into the operation of a standard engine. A first approach is to relate the results to the Brake Mean Effective Pressure (*bme_p*) calculated considering the turbocharger effect in the engine. A preliminary theoretical approach [210] is available in APPENDIX D: Theoretical approach to Turbocharger contribute to *bme_p*., however the coefficients used vary depending on engine characteristics. For the HAT prototype application an empirical approach was adopted; this consisted in relating the boost pressure experimental data to the *bme_p* /engine load data collected on a 4 cylinders Ford Puma 2.2L engine with maximum power of 92 kW @ 3700 rpm available in L4 laboratories at the University of Nottingham. The range of engine speeds between 1600 rpm and 2400 rpm was considered as reference values. In this range the achieved boost corresponds to just 16% of the overall engine load conditions.

The overall boost depends ultimately on the maximum speed achieved and this depends on several factors in the hydraulic-assisted turbocharger design. Firstly, the hydraulic source needs to be investigated in higher range of magnitude. A valid alternative to a hydraulic accumulator is a standard diesel pump which is capable of delivering up to 2000 bar. The optimised computational model shows that for input pressure of 500 bar, still considering the pressure trend decrease given from the accumulator discharge nature, the system is capable of delivering 105,000 rpm in less than 0.6s. With the current geometrical conditions of the standard turbocharger and the hydraulic turbine, it would mean to increase the oil flow-rate of +30% compared to the maximum tested pressure of 250 bar with an absolute value of 59 L/min. The transient event in which the assist would be used is less than 2s meaning the hydraulic circuit would need to take in account a ready availability of about 2L for each transient event. A second step would be

to investigate an optimised geometry of the hydraulic wheel to further increase the achievable performance.

Ultimately the information available in this thesis are essential to understand the potential application of the prototype, however further studies are required to improve the output maximum speed response and frame the HAT in a more realistic engine-like condition.

7.5 Comparison *KERB-HAT* vs *EAT*

In Chapter 2 it was highlighted how turbocharger-lag can be mitigated with different techniques; among these, the main competitor to the HAT is the electrically-assisted-turbocharger EAT. The overview on EAT devices identified that the most suitable application was in light-duty vehicles. The EAT has been shown to represent an efficient solution to decrease time response for 1.5L engines [211] however, the HAT prototype is a valid alternative to the e-turbo advanced technology considering the parameters to attention regarding the lag issue: the time response, maximum speed and an easy retrofit in the vehicle. Computational models show the potentiality of the EAT in commercial light-duty engines [212] however simulated and experimental EAT papers in some cases do not give specific time response/torque to discuss and compare [59], [213]–[217]; for this reason, the comparison in this paragraph will be done considering mainly the reliable experimental data in literature for EAT and the experimental data obtained from HAT in TPSR campaign.

An interesting study was carried out in a 7L off-highway diesel engine using a 5 kW reluctance motor integrated in a turbocharger test rig based on the model BorgWarner BV63 VTG with a modified bearing housing [218]. To fit the assisted source, the shaft length of the prototype needed to be significantly increased compared to the standard model of the turbocharger. This is avoided in KERB-HAT prototype as one of the design criteria was to maintain the bearing housing external dimension to ensure retrofit in an existing turbocharger. The best results obtained in the EAT study was with a starting speed of 65k rpm reaching a final speed of around 85k rpm in 1.68s. As shown in experimental results in Chapter 6, the time response experienced in the HAT was from 0.64s to 0.98s depending on the configuration tested. The starting speed was either 0 or 2500-3000 rpm, reaching approximately from 51k rpm to 63k rpm.

Comparing the results with [218] and considering that a further investigation showed that the starting speed did not influence the HAT prototype time response, it can be assumed that the performances of KERB-HAT are higher in the absolute value having a lower time response of 42-57% average with a peak of 62%. The same reasoning can be carried out for comparing a tested EAT in a 6 cylinder 7.8L heavy-duty diesel engine in which a speed of 120k rpm was reached in approximately 1.2s starting however from 70krpm [85]. HAT minimum time response is then 46% lower.

It must be noted that the turbochargers models used in [85], [218] are meant for heavy-duty engine whereas the Garrett GTX2867r in the TPSR experimental campaign is meant for passenger cars. Other studies are then compared on improving time response using a e-turbo in passenger car engines 1-2.5L. Mitsubishi tested a 1.5L engine with a two-stage electric turbocharging system [82] which was able to reach 90k rpm and a boost of 1.6 bar in less than 0.6s; the KERB-HAT maximum speed was 63k rpm in 0.64s thus it is reasonable to think the projected performance would match if not overcome the electric turbo in this case. Another example that reinforces the above projected comparison is [83] in which 74k rpm were reached in 1.3s on an EAT mounted in 2L diesel engine, reminding that the time response is inversely proportional to the maximum speed. Katrasnik et al. [84] developed a computational model, validated with experimental data, suitable for a 6.87L cylinder but equipped with an EAT with inertia $6e-5 \text{ kgm}^2$, less than KERB-HAT of $9.96e-5 \text{ kgm}^2$. The results can then be compared and the study showed a time response of the turbocharger at 65k rpm of approximately 1s, demonstrating then 36% worse performances compared to KERB-HAT.

Regarding the retrofit of the HAT prototype, it can be imagined to be framed in both light-duty and heavy-duty vehicles, however the most interesting application is considered for heavy-duty vehicles. The reasons for this choice are summarised below:

- E-turbo is already efficiently used in light-duty where it matches the wide electrification process that it is taking place.
- The roadmap shown in Chapter 1 makes clear that ICE improvement efficiency techniques are highly more requested for heavy-duty vehicles

where electrification cannot meet the power requirements demand of heavy-duty panorama.

- In heavy-duty and off-road machinery it is more likely to have a hydraulic source of recovery energy (i.e. Diesel pump, hydraulic circuits).
- The typical assisted motor in EAT have a power in the range of 1.5-10kW, whereas the power required for a larger turbocharger would be higher than 10kW [83][92][93].
- The electrification of trucks is avoided at the moment because of logistic reasons as for example the mismatch between long-time of charging required for the vehicle battery and journey schedule of transportation trucks as well as the weight of the batteries involved.

It must be noted that the speed results obtained with the HAT in the TPSR campaign in Chapter 6 were achievable mainly with the hydraulic assist and the initial flow rate is different from the results in literature for EAT. It is recommended for future studies to confirm the comparisons in this section, to have a external source of initial air flowrate capable of explore a realistic compressor map.

On a final note, a larger size turbocharger used in 2-8L engines can be used as the speed required are in line with the preliminary tested boosts and they are typically around 70,000 rpm [219] as confirmed from the turbocharger map of a Garrett GTX4508R typical for engine up to 8L displacement [220].

7.6 Alternative applications of HAT

KERB-HAT, other than heavy-duty off-road diesel engines, could be considered for hydrogen-fuelled vehicles which still rely on some of the traditional improvement efficiency techniques for ICE; in these vehicles the use of a turbocharger was proven to increase the power up to 41% compared to a naturally aspirated engine [221]. Furthermore, a 2L turbocharged hydrogen-engine exceeded by 123% and 195% the peak power and maximum torque respectively of the correspondent NA engine[222].

The stand-alone compression output of a turbocharger, without the energy recovery, can be used to produce the oxygen required for the Fuel cell power

generation where preliminary studies shows that the pressure ratio boost required lies below 1.7 [223]. The European Commission supported this idea for the Horizon 2020 series, funding a project investigating a compact and efficient turbo compressor for fuel cells vehicles [224], [225].

Exploring outside the automotive environment, the principle of energy recovery and efficiency savings using hydraulic source to create boost pressure has been used in desalination systems since the early '90 [226]. During operation the high-pressure brine rejected from reverse osmosis elements enters the turbine side of the unit and the pressure energy is transferred to a pump impeller. This strategy can produce a boost pressure of 2bar with 2500 L/min required at the pump [227]. An interesting analysis on scaling solutions for the HAT could confirm if the power requirements are suitable for the performances revealed in this thesis.

Summary

The discussion chapter aims to relate the results from the experimental campaign to the modelling of KERB-HAT framing its potential application in a real-driving application; the main takeaways are summarised below:

- A HAT rotational speed can be modelled using Pelton wheel dynamic equations [162], bearing frictions equations [189] and windage losses equations optimizing the final trend including a coefficient k_w to consider the compressor/turbine shrouding [194] or having reliable measurement of the casing clearance s [152].
- A geometrically and fluid dynamically optimised HAT with an ACBB housing compared to a standard turbocharger with plain bearing housing retrofitted with a Pelton wheel experiences an average maximum speed increase of +67%, but -22% maximum speed compared to the same Pelton wheel free to run with an inertia disc. This last results is attributed to the additional windage losses of the turbocharger which are shown to be an order of magnitude higher.

- In the KERB-HAT application the values that gives a reliable prediction model for the maximum speed are $k_w=0.7$ using a gas turbine wheel formula [194] and $k_w=1$ and $s=0.4\text{mm}$ using a formula for high-speed machinery which includes the shrouding effect [152]. The empirical parameters in the formula includes Pelton friction effect, the maximum speed is matched and the time response is underestimated by 12%.
- In a HAT the hydraulic wheel must be geometrically optimised to achieve higher performance: the bucket width B must be three times the jet diameter and the pitch circle diameter must be greater than $3B$.
- A tested hydraulic-assist device retrofitted in a standard fixed geometry turbocharger for engine displacement 1.4-2.5 L produces a maximum power boost which covers 16% of the interested range of engine load-speed of a standardised WLTC cycle for a light-duty engine.
- The simulated hydraulic-assisted turbocharger with +30% of flow rate and doubling the input pressure is capable of delivering 105,000 rpm in less than 0.6s, however further optimization is needed for the geometrical and input source characteristics.
- A HAT compared to the main competitor with electric-assist achieves 23% lower speed on average but faster time response with an average improvement of the latter of 49% in light-duty applications.
- The HAT technology is framed more efficiently in heavy-duty application because of the wide electrification process of light-duty, ICE improvement requirements in heavy-duty industry and power specifications of the latter.

8 Conclusions

8.1 Summary of findings

The PhD research project aimed to investigate the hydraulic-assisted turbocharger potentiality as a strategy to mitigate the intrinsic turbo-lag issue in automotive applications. The conclusions in this section are drawn to meet the research objectives in section 1.6. In section 8.2 the novelty and contribution of the research project is presented.

- The charging pressure at the accumulator was identified as the main parameter influencing the rotational speed of the hydraulic wheel, whereas the oil volume charging does not have a significant impact (<5%).
- An initial HAT prototype inherited showed that an unoptimised design can incur in high windage losses, high plain bearing friction losses and hydraulic lock causing low speeds.
- A free running Pelton configuration can be used to identify the optimised criteria in terms of speed and time response, bearing friction losses, windage losses and hydraulic lock. For the proposed application the best setup was achieved using ceramic angular contact ball bearings, a 40mm diameter shrouding with an open draining of 19x40mm and 2x1.5mm nozzles centred onto the bucket splitter horizontally, and in the correspondent pitch circle Pelton position vertically.
- The optimised design can be embedded into a Garrett turbocharger GTX2867r without altering its original dimensions so that the HAT can be potentially retrofitted in an HHV.

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- To achieve a higher target boost, a higher time response of the HAT is required. For example, increasing the target boost by 15-16% leads to a 6-15% increase in time response.
- Experimentally, a 250 bar oil hydraulic assist leads to a maximum speed of 63,000 rpm and time responses of 0.64s producing a 1.31 bar boost pressure. Theoretical predictions show that, with an increase of 30% of oil flow rate supplied, over 105,000 rpm can be achieved increasing consequently the boost pressure and reducing the time response. The minimum accumulator pressure for this configuration and performance in a HHV should be a minimum of 2-5L.
- The overall time response in similar conditions of an EAT is lower compared to the HAT; in particular a decrease of time response from 42% to 57% is achievable, however the absolute maximum speed is on average 23% lower.
- In the HAT configuration two different adapted models, namely Buckingham and Etemad can be used to predict windage losses; the new adapted Buckingham model requires a shrouding coefficient k_w equal to 0.7. To account for the shrouding effect in the Etemad model the clearance s is set to 0.4mm. This allowed, with both models, to predict the maximum experimental speed, however the time response is underestimated by 12%.
- The HAT, compared to the EAT in real-driving automotive applications, finds its place in heavy-duty vehicles because of lower turbocharger maximum speeds required, lack of electrification for this category and high-hydraulic flow rate required.

8.2 Novelty of the research

The innovative topics faced in the thesis are here summarised:

- In literature, separate models are available to predict the discharge of a hydraulic accumulator and the rotational speed of a Pelton with constant jet speed. The computational model described in Chapter 4 represents a tool to correlate the equations of the two phenomena; this is a development and innovative contribution in the research. The model allows to predict the variation in time of the rotational speed of the Pelton turbine based on the variation of the charging pressure of the hydraulic accumulator. This is an added value of what is currently available in research. Furthermore, the effect of several parameters (number of nozzles, nozzle diameter, change of rig geometry, etc.) on the final speed/time response of the turbine can be parametrically investigated.
- In literature there is no detailed methodology available on how to experimentally test a HAT. Chapter 3 and Chapter 6 provide the description on how to build and manage a reliable rig to obtain experimental results with a hydraulic accumulator connected with a HAT.
- Although some results are available in literature for some configurations of a HAT (section 2.3.4), this thesis provide for the first time parametric and optimization studies for the use of a HAT; the findings are available in Chapter 5 and Chapter 6 and summarised in sections 5.5 and 6.4.

8.3 Suggestion for Experimental improvements

The research Project mainly focused on showing experimentally the feasibility of a HAT prototype. Speed and time response were successfully investigated, however, the rig could be equipped with a torque meter to better investigate effectively the windage losses from different configurations. To have better control of the hydraulic source, a digital pressure transducer could be adapted in the charging kit of the accumulator. To increase the oil input pressure it is advised to use a high-pressure pump similar to the one used in diesel injection systems; higher speeds and boost would be achieved giving the opportunity to exploring more points of the compressor map. The turbocharger rig stand should be also modified increasing the distance between the compressor outlet and the butterfly valve, to allow more space for the sensor and minimise noise. Difficulty

in reading the output temperature was encountered because of the turbulent flow generated from the excessive restriction when the valve was 77% closed. Another important suggestion is to have an external source more powerful to replicate the initial flow rate for the testing. This is the TPSR campaign was limited from the available compressor of the lab described in section 6.2.3.

8.4 Suggestion for HAT Design improvements

In terms of HAT design, further optimizations can be carried out manufacturing a new Pelton turbine; the wheel should be characterised for the specific application for heavy-duty having a larger-size turbocharger. The bearing housing of the prototype is equipped with high-performance angular contact ball bearings; in the design, the oil pattern was not included to simplify the manufacturing process and because the oil injected in the Pelton chamber was assumed to give enough lubrication for the duration of the tests. The bearings are claimed to be run ideally with no oil, however, it is advised to include, if necessary, an oil feed in the bespoke bearing housing.

8.5 Suggestion for Modelling improvements

The 0D computational model shown in the thesis was a powerful tool to explore multiple conditions, however, it is considered just as a baseline regarding the theoretical approach. A 2D/3D study can be carried out using CFD techniques to investigate the Pelton flow dynamic and assess the nature of windage losses. This can also help in investigating the assumed effect of increased inertia. Furthermore, precise data from manufacturers are required when simulating the turbocharger. A study of vibrations in the device is advised to investigate the operational frequency of the rotating assembly and understand the capacity limits. Additionally, a study on the behaviour of the hydraulic oil flow subjected at the heat transfer deriving from the gas turbine side in operation is advised to explore possible limitative operative conditions. To characterise the assisted turbocharger and matching it with a real-engine application, the preliminary modelling approach is described by Watson [48]. A more inclusive approach would be to use GT power software that can provide a more complete simulation environment. This is considered an important procedure to relate the stand-alone

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turbocharger performances in a heavy-duty vehicle or off-road machinery application.

References

- [1] E. O.-O. Max Roser, Hannah Ritchie, “Future population growth - our world in data,” 2019. <https://ourworldindata.org/world-population-growth> (accessed Sep. 14, 2021).
- [2] V. Smil, “Global primary energy consumption by source - Our world in data,” 2019. <https://ourworldindata.org/energy-overview> (accessed Sep. 14, 2021).
- [3] H. Ritchie, “Sector by sector: where do global greenhouse gas emissions come from?,” 2020. <https://ourworldindata.org/ghg-emissions-by-sector> (accessed Jul. 21, 2021).
- [4] H. Ritchie, “Cars, planes, trains: where do CO2 emissions from transport come from?,” 2020. <https://ourworldindata.org/co2-emissions-from-transport> (accessed Oct. 05, 2021).
- [5] H. Ritchie and M. Roser, “United Kingdom: CO2 Country Profile,” 2020. <https://ourworldindata.org/grapher/ghg-emissions-by-sector?time=latest&country=~GBR> (accessed Jul. 21, 2021).
- [6] H. Ritchie and M. Roser, “Annual greenhouse gas emissions by section - Interactive chart - Our World in Data,” 2020. <https://ourworldindata.org/emissions-by-sector#by-country-greenhouse-gas-emissions-by-sector> (accessed Nov. 01, 2021).
- [7] UNFCCC. Conference of the Parties (COP), “Paris Climate Change Conference-November 2015, COP 21,” *Adoption of the Paris Agreement. Proposal by the President.*, vol. 21932, no. December, p. 32, 2015, doi: FCCC/CP/2015/L.9/Rev.1.
- [8] S. Polzin, “COVID-19 ’ s Effects on The Future of Transportation,” 2021. doi: <https://doi.org/10.21949/1520705>.
- [9] E. Mulholland, J. Teter, P. Cazzola, Z. McDonald, and B. P. Ó Gallachóir, “The long haul towards decarbonising road freight – A global assessment to 2050,” *Appl Energy*, vol. 216, no. February, pp. 678–693, 2018, doi: 10.1016/j.apenergy.2018.01.058.
- [10] E. (European A. manufacturers Association), “Medium and Heavy Trucks over 3.5 T New Registrations by Fuel Type in the European Union,” 2019.

References

- https://www.acea.auto/uploads/press_releases_files/ACEA_trucks_by_fuel_type_full-year_2019.pdf
- [11] G. Giuliano, M. Dessouky, S. Dexter, J. Fang, S. Hu, and M. Miller, “Heavy-duty trucks: The challenge of getting to zero,” *Transp Res D Transp Environ*, vol. 93, no. March, p. 102742, 2021, doi: 10.1016/j.trd.2021.102742.
 - [12] R. Dua, O. al Harbi, G. Collins, and T. Sheldon, “A cost-effective pathway to a low-emissions transportation future-G20 insights,” 2020. Accessed: Sep. 16, 2021. [Online]. Available: https://www.g20-insights.org/policy_briefs/a-cost-effective-pathway-to-a-low-emissions-transportation-future/
 - [13] M. Albrahim, A. Al Zahrani, A. Arora, R. Dua, B. Fattouh, and A. Sieminski, “An overview of key evolutions in the light-duty vehicle sector and their impact on oil demand,” *Energy Transitions*, vol. 3, no. 1–2, pp. 81–103, 2019, doi: 10.1007/s41825-019-00017-7.
 - [14] UK Research and Innovation (UKRI), “Driving the electric revolution-2020 annual report,” 2021. Accessed: Sep. 22, 2021. [Online]. Available: <https://www.ukri.org/wp-content/uploads/2021/05/UKRI-070521-DrivingTheElectricRevolution2020AnnualReport.pdf>
 - [15] W. World Health Organization, “Ambient (outdoor) air pollution.” [https://www.who.int/news-room/fact-sheets/detail/ambient-\(outdoor\)-air-quality-and-health](https://www.who.int/news-room/fact-sheets/detail/ambient-(outdoor)-air-quality-and-health) (accessed Sep. 22, 2021).
 - [16] Advanced Propulsion Centre, “Roadmap 2020 Heavy Goods >3.5t and Off-highway Vehicle,” 2021. Accessed: Oct. 18, 2021. [Online]. Available: <https://www.apcuk.co.uk/app/uploads/2020/11/Product-Roadmap-HGV-and-Off-highway.pdf>
 - [17] Advanced Propulsion Centre UK, “APC UK Roadmap to Net-Zero,” 2020. Accessed: Jun. 12, 2022. [Online]. Available: <https://www.apcuk.co.uk/media-type/roadmaps/>
 - [18] P. Technologies, “the Roadmap Report Towards 2040: a Guide To Automotive Propulsion Technologies,” 2018. Accessed: Sep. 26, 2018. [Online]. Available: www.apcuk.co.uk
 - [19] T. Zeng, Y. Men, D. Upadhyay, and G. Zhu, “Energy availability study for a regenerative hydraulically assisted turbocharger,” in *ASME 2018*

References

- Dynamic Systems and Control Conference*, 2018. doi: <https://doi.org/10.1115/DSCC2018-9134>.
- [20] T. Zeng, D. Upadhyay, H. Sun, and E. Curtis, “Regenerative hydraulic assisted turbocharger,” *J Eng Gas Turbine Power*, vol. 140, no. 102602–1, 2018, doi: DOI: 10.1115/1.4039937.
- [21] H. Turbocharger and T. Zeng, “Modeling and Control of a Diesel Engine With Regenerative hydraulic-assisted turbocharger,” *J Eng Gas Turbine Power*, vol. 141, no. May, 2019, doi: 10.1115/1.4041932.
- [22] S. G. Timoney, “Turbochargers with Hydraulic Assist,” in *SAE International Congress and Exposition*, Feb. 1983. doi: <https://doi.org/10.4271/830012>.
- [23] R. P. Kepner, “Hydraulic Power Assist – A Demonstration of Hydraulic Hybrid Vehicle Regenerative Braking in a Road Vehicle Application,” *SAE International*, no. 2002-01–3128, 2002.
- [24] United Nations Climate change, “Global Car Industry Must Shift to Low Carbon to Survive - CDP,” 18/01/2018, 2018. <https://unfccc.int/news/global-car-industry-must-shift-to-low-carbon-to-survive-cdp> (accessed Sep. 20, 2018).
- [25] L. Fletcher, K. Marcell, and T. Crocker, “CDP Europe annual report 2018,” 2018. Accessed: Sep. 26, 2018. [Online]. Available: https://cdn.cdp.net/cdp-production/cms/reports/documents/000/004/753/original/CDP_Europe_annual_report_FY2018-2019_spreads.pdf?1572600922
- [26] UK Government, “Government takes historic step towards net-zero with end of sale of new petrol and diesel cars by 2030,” 18 November 2020, 2020. <https://www.gov.uk/government/news/government-takes-historic-step-towards-net-zero-with-end-of-sale-of-new-petrol-and-diesel-cars-by-2030> (accessed Aug. 24, 2021).
- [27] S. J. Davis *et al.*, “Net-zero emissions energy systems,” *Science* (1979), vol. 360, no. 6396, 2018, doi: 10.1126/science.aas9793.
- [28] International Energy Agency, “Energy Technology Perspectives 2020,” 2020. doi: 10.1787/ab43a9a5-en.
- [29] İ. A. Reşitoğlu, K. Altinişik, and A. Keskin, “The pollutant emissions from diesel-engine vehicles and exhaust aftertreatment systems,” *Clean*

References

- Technol Environ Policy*, vol. 17, no. 1, pp. 15–27, 2015, doi: 10.1007/s10098-014-0793-9.
- [30] Department for Environment Food & Rural Affairs, “Clean Air Strategy 2019-UK Government,” 2019. doi: 10.5694/j.1326-5377.1962.tb59377.x.
- [31] EU, “Regulation (Eu) 2017/1129 of the European Parliament and of the Council,” *Official Journal of the European Union*, vol. L 168, no. 12, p. 71, 2017.
- [32] Diesel.net, “EU: Heavy-Duty truck and Bus Engines,” 2021. <https://dieselnet.com/standards/eu/hd.php> (accessed Oct. 05, 2021).
- [33] European Parliament And The Council Of The European Union, “Directive 2005/55/EC,” *Official Journal of the European Union*, no. L 275, pp. 1–163, 2005.
- [34] Vehicle Certification Agency UK Government, “New Car Fuel Consumption & Emissions Figures,” 2020. Accessed: Sep. 07, 2021. [Online]. Available: <http://www.vcacarfueldata.org.uk/downloads/may2010.asp>
- [35] W. World Health Organization, “WHO Air quality guidelines for particulate matter, ozone, nitrogen dioxide and sulfur dioxide: Global update,” pp. 1–21, 2005, doi: 10.1016/0004-6981(88)90109-6.
- [36] Department for Environment Food and Rural Affairs, “The Air Quality Strategy for England , Scotland , Wales and Northern Ireland: Volume 1,” 2007. Accessed: Oct. 05, 2021. [Online]. Available: <http://archive.defra.gov.uk/environment/quality/air/airquality/strategy/documents/air-qualitystrategy-vol1.pdf>
- [37] E. A. Piersall, “The Road To Zero,” 2018. Accessed: Sep. 07, 2021. [Online]. Available: https://assets.publishing.service.gov.uk/government/uploads/system/uploads/attachment_data/file/739460/road-to-zero.pdf
- [38] U. Government, “CO2 emission performance standards for new passenger cars and light commercial vehicles,” 2020. <https://www.gov.uk/government/consultations/regulating-co2-emission-standards-for-new-cars-and-vans-after-transition/co2-emission-performance-standards-for-new-passenger-cars-and-light-commercial-vehicles> (accessed Jul. 09, 2021).

References

- [39] G. Sovran and D. Blaser, “Quantifying the potential impacts of regenerative braking on a vehicle’s tractive-fuel consumption for the U.S., European, and Japanese driving schedules,” *SAE Technical Papers*, vol. 2006, no. 724, 2006, doi: 10.4271/2006-01-0664.
- [40] European Union, “Hybrids for road transport: Status and prospects of hybrid technology and the regeneration of energy in road vehicles,” 2005. Accessed: Oct. 25, 2021. [Online]. Available: <https://op.europa.eu/en/publication-detail/-/publication/22bb42f4-8525-4bf3-9621-5800801f7a0e>
- [41] J. Choi, J. Jeong, Y. il Park, and S. W. Cha, “Analysis of regenerative braking effect to improve fuel economy for E-REV bus based on simulation,” *World Electric Vehicle Journal*, vol. 7, no. 3, pp. 366–370, 2015, doi: 10.3390/wevj7030366.
- [42] R. Chicurel, “A compromise solution for energy recovery in vehicle braking,” *Energy*, vol. 24, no. 12, pp. 1029–1034, 1999, doi: 10.1016/S0360-5442(99)00054-7.
- [43] H. Ritchie, “Cars, planes, trains: where do CO2 emissions from transport come from?,” 2020. <https://ourworldindata.org/co2-emissions-from-transport> (accessed Oct. 05, 2021).
- [44] A. Jain, T. Nueesch, C. Naegele, P. M. Lassus, and C. H. Onder, “Modeling and Control of a Hybrid Electric Vehicle With an Electrically Assisted Turbocharger,” *IEEE Trans Veh Technol*, vol. 65, no. 6, pp. 4344–4358, 2016, doi: 10.1109/TVT.2016.2533585.
- [45] M. Karabektas, “The effects of turbocharger on the performance and exhaust emissions of a diesel engine fuelled with biodiesel,” *Renew Energy*, vol. 34, no. 4, pp. 989–993, 2009, doi: <https://doi.org/10.1016/j.renene.2008.08.010>.
- [46] S. Arnold, K. Slupski, M. Groskreutz, G. Vrbas, R. Cadle, and S. M. Shahed, “Advanced turbocharging technologies for heavy-duty diesel engines,” *SAE Technical Papers*, vol. 110, pp. 2048–2055, 2001, doi: 10.4271/2001-01-3260.
- [47] N. Terdich and I. C. London, “Impact of electrically assisted turbocharging on the transient response of an off-highway Diesel Engine,” 2015. doi: <https://doi.org/10.25560/25395>.

References

- [48] N. Watson and M. Janota, *Turbocharging the internal combustion engine*. 1982. Accessed: Oct. 10, 2021. [Online]. Available: <https://books.google.co.uk/books?id=MDFdDwAAQBAJ>
- [49] J. Justus, “Theoretical and Experimental Investigation of a Novel Hydraulically Assisted Turbocharger System for Future Automotive Applications,” 2016. Accessed: May 29, 2018. [Online]. Available: <http://bura.brunel.ac.uk/handle/2438/14763>
- [50] Advanced Propulsion Centre, “Roadmap 2020 Heavy Goods >3.5t and Off-highway Vehicle,” 2021. Accessed: Oct. 18, 2021. [Online]. Available: <https://www.apcuk.co.uk/app/uploads/2020/11/Product-Roadmap-HGV-and-Off-highway.pdf>
- [51] Department for Environment Food & Rural Affairs, “Clean Air Strategy 2019-UK Government,” 2019. doi: 10.5694/j.1326-5377.1962.tb59377.x.
- [52] G. H. Abdollah, “Using Exhaust Gas Recirculation in Internal Combustion Engines: a Review,” *Energy Convers Manag*, vol. 43, no. X, pp. 1027–1042, 2002.
- [53] H. Ritchie and M. Roser, “CO₂ and Greenhouse Gas Emissions,” 2020. <https://ourworldindata.org/co2-and-other-greenhouse-gas-emissions> (accessed Nov. 01, 2021).
- [54] European Commission-Climate change action, “Reducing CO₂ emissions from heavy-duty vehicles,” 2021. Accessed: Nov. 30, 2021. [Online]. Available: https://ec.europa.eu/clima/eu-action/transport-emissions/road-transport-reducing-co2-emissions-vehicles/reducing-co2-emissions-heavy-duty-vehicles_en
- [55] European Union, “Regulation (EU) 2019/1242 reducing CO₂ Emissions from Heavy duty vehicles,” 2019. Accessed: Jul. 20, 2022. [Online]. Available: <https://www.legislation.gov.uk/eur/2019/1242/contents>
- [56] N. Watson and M. Janota, *Turbocharging the Internal Combustion Engine*. Macmillan Press, 1982. Accessed: Oct. 10, 2021. [Online]. Available: <https://books.google.co.uk/books?id=MDFdDwAAQBAJ>
- [57] J. B. Heywood, *Internal Combustion Engine Fundamentals*. McGraw-Hill, 1988. [Online]. Available: <https://books.google.co.uk/books?id=O69nQgAACAAJ>

References

- [58] E. CANLI, S. DARICI, and M. OZGOREN, “Intercooler Effect on Conventional Supercharging Systems,” *Unitech International Scientific Conference Gabrovo*, vol. 2, no. November, pp. 242–248, 2010.
- [59] P. Dimitriou, R. Burke, Q. Zhang, C. Copeland, and H. Stoffels, “Electric turbocharging for energy regeneration and increased efficiency at real driving conditions,” *Applied Sciences (Switzerland)*, vol. 7, no. 4, 2017, doi: 10.3390/app7040350.
- [60] C. E. Weaver and Ford Research and Advanced Engineering, “Advanced gasoline turbocharged direct injection (GTDI) engine development,” Department of Energy Vehicle Technology, USA, 2014. Accessed: Nov. 08, 2021. [Online]. Available: https://www.energy.gov/sites/prod/files/2014/07/f17/ace065_weaver_2014_o.pdf
- [61] D. A. Singer, “Comparison of a supercharger vs. a turbocharger in a small displacement gasoline engine application,” *SAE Technical Papers*, vol. 94, pp. 268–279, 1985, doi: 10.4271/850244.
- [62] W. Lee, E. Schubert, Y. Li, S. Li, D. Bobba, and B. Sarlioglu, “Overview of Electric Turbocharger and Supercharger for Downsized Internal Combustion Engines,” *IEEE Transactions on Transportation Electrification*, vol. 3, no. 1, pp. 36–47, 2017, doi: 10.1109/TTE.2016.2620172.
- [63] Turbo Dynamics, “How does a Turbocharger works,” 2020. <https://www.turbodynamics.co.uk/technical/understanding-turbochargers/> (accessed Jul. 05, 2022).
- [64] Garrett, “Garrett advancing motion training.” <https://www.garrettmotion.com/installer-connect/training/level-1/> (accessed Jun. 22, 2021).
- [65] Garrett Advancing Motion, “Performance Catalog Vol 8,” *Garret Motion*, 2020. https://www.garrettmotion.com/wp-content/uploads/2020/06/Catalog-Volume-8_Full-Version_FINAL_web-version_June_2020_PAGES.pdf (accessed Nov. 30, 2021).
- [66] S. Arnold, C. Balis, D. Jeckel, S. Larcher, P. Uhl, and S. M. Shahed, “Advances in Turbocharging Technology and its Impact on Meeting

References

- Proposed California GHG Emission Regulations,” *Engineering*, no. 724, 2011, doi: 10.4271/2005-01-1852.
- [67] “Pollution from Internal Combustion Engine Vehicles,” 2018. Accessed: Jul. 09, 2018. [Online]. Available: <https://www.essaysauce.com/environmental-studies-essays/pollution-internal-combustion-engine-vehicles/>
- [68] E. G. Giakoumis, “Review of Some Methods for Improving Transient Response in Automotive Diesel Engines through Various Turbocharging Configurations,” *Front Mech Eng*, vol. 2, no. May, 2016, doi: 10.3389/fmech.2016.00004.
- [69] P. Walzer, “Future power plants for cars,” *SAE International*, 2001, doi: <https://doi.org/10.4271/2001-01-3192>.
- [70] S. M. Shahed, “An Analysis of Assisted Turbocharging with Light Hybrid Powertrain,” *SAE International*, 2006, doi: 10.4271/2006-01-0019.
- [71] D. Petitjean, L. Bernardini, C. Middlemass, and S. M. Shahed, “Advanced Gasoline Engine Turbocharging Technology for Fuel Economy Improvements,” *SAE International*, no. 724, 2004, doi: 10.4271/2004-01-0988.
- [72] R. Merker, Günter P., Schwarz, Christian, Teichmann, *Combustion Engines Development*. Springer, 2012.
- [73] M. Alshammari, F. Alshammari, and A. Pesyridis, “Electric boosting and energy recovery systems for engine downsizing,” *Energies*, vol. 12, no. 24. MDPI AG, Dec. 06, 2019. doi: 10.3390/en12244636.
- [74] M.-B. Ricardo, A. Pesyridis, and M. Yang, “Overview of boosting options for future downsized engines,” *Sci China Technol Sci*, vol. 54, pp. 318–331, Jul. 2011, doi: 10.1007/s11431-010-4272-1.
- [75] “What is a turbo wastegate valve?,” 2017. <http://www.gp-turbo.com/turbo-wastegate-valve/> (accessed Oct. 15, 2018).
- [76] S. Rajoo, A. Pesyridis, and R. Martinez-Botas, “Novel method to improve engine exhaust energy extraction with active control turbocharger,” *International Journal of Engine Research*, vol. 15, no. 2, pp. 236–249, Feb. 2014, doi: 10.1177/1468087412472414.

References

- [77] “What is a turbocharger and how it works.” <http://blog.solediesel.com/en/what-is-a-turbocharger-and-how-it-works/> (accessed Oct. 15, 2018).
- [78] P. Newman *et al.*, “Electrical supercharging for future diesel powertrain applications,” *11th International Conference on Turbochargers and Turbocharging*, pp. 207–216, 2014, doi: 10.1533/978081000342.207.
- [79] G. Tavčar, F. Bizjan, and T. Katrašnik, “Methods for improving transient response of diesel engines - Influences of different electrically assisted turbocharging topologies,” *Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering*, vol. 225, no. 9, pp. 1167–1185, 2011, doi: 10.1177/0954407011414461.
- [80] T. Suzuki, Y. Hirai, and N. Ikeya, “Electrically Assisted Turbocharger as an enabling technology for improved fuel economy in New European Driving Cycle operation,” *11th International Conference on Turbochargers and Turbocharging*, pp. 217–226, 2014, doi: <http://dx.doi.org/10.1533/978081000342.217>.
- [81] S. G. Alain Lefebvre, “Transient response of a Turbocharged SI Engine with an electrical boost pressure supply,” *SAE International Spring Fuels & Lubricants Meeting, Yokohama, Japan*, vol. 2003-01–18, 2003.
- [82] A. Byeongil, N. Shibata, H. Suzuki, and M. Ebisu, “Development of Two-stage Electric Turbocharging system for Automobiles,” *Mitsubishi Heavy Industries Technical Review*, vol. 52, no. March 2015, pp. 71–76, 2015.
- [83] S. Ibaraki, Y. Yamashita, K. Sumida, H. Ogita, Y. Jinnai, and M. H. Industries, “Development of the ‘hybrid turbo,’ an electrically assisted turbocharger,” *Mitsubishi Heavy Industries Technical Review*, vol. 43, no. 3, pp. 1–5, 2006, doi: 10.4271/2002-01-0378.
- [84] T. Katrašnik, S. Rodman, F. Trenc, A. Hribernik, and V. Medica, “Improvement of the dynamic characteristic of an automotive engine by a turbocharger assisted by an electric motor,” *J Eng Gas Turbine Power*, vol. 125, no. 2, pp. 590–595, 2003, doi: 10.1115/1.1563246.
- [85] F. Millo, F. Mallamo, E. Pautasso, and G. Ganio Mego, “The potential of electric Exhaust Gas Turbocharging for HD diesel engines,” *SAE Technical Papers*, no. 724, 2006, doi: 10.4271/2006-01-0437.

References

- [86] R. Birmann, “Turbocharging of the internal combustion engine,” 2,968,914, 1961 Accessed: Nov. 15, 2021. [Online]. Available: <https://patentimages.storage.googleapis.com/eb/f1/7b/1b2254f6bfdc6b/US2968914.pdf>
- [87] Byrne, “Hydraulic assisted turbocharger system patent US4285200_1981” Accessed: Sep. 26, 2018. [Online]. Available: <https://patentimages.storage.googleapis.com/3d/b3/df/ea67175f5a502d/US4285200.pdf>
- [88] R. J. Kobayashi, “Hydraulic assist turbocharger system,” 5,113,658, 1992 Accessed: Nov. 23, 2021. [Online]. Available: <https://patentimages.storage.googleapis.com/a9/44/f2/af6d2d49a5d88c/US4622817.pdf>
- [89] D. Kapich, “Hydraulic Turbine-Pump Hybrid Turbocharger System,” US 2012/0180482 A1, 2012 Accessed: Nov. 23, 2021. [Online]. Available: <https://patentimages.storage.googleapis.com/cc/47/3d/e363ba0c124e17/US20120180482A1.pdf>
- [90] F. Z. S. Harold Hiumin Sun, David R. Hanna, Michael Levin, Eric Warren Curtis, “Regenerative Assisted Turbocharger System,” US 2012/0198846 A1 Accessed: Nov. 23, 2021. [Online]. Available: <https://patentimages.storage.googleapis.com/ba/6e/be/df2239db8c3aac/US8915082.pdf>
- [91] D. Kapich, “Turbo-Hydraulic Engine Exhaust Power Recovery System,” in *SAE Powertrain & Fluid Systems Conference & Exhibition*, 2002, vol. 2002-01–27. doi: <https://doi.org/10.4271/2002-01-2731>.
- [92] A. Rezk, J. Carter, M. L. Jupp, A. K. Nickson, and J. Allport, “Investigation of an electrically assisted turbocharger with energy recovery for a heavy duty diesel engine,” in *12th International Conference on Turbochargers and Turbocharging 2016*, vol. 2016-May, pp. 319–330.
- [93] T. Noguchi, Y. Takata, Y. Yamashita, Y. Komatsu, and S. Ibaraki, “220,000-r/min, 2-kW PM motor drive for turbocharger,” *Electrical Engineering in Japan (English translation of Denki Gakkai Ronbunshi)*, vol. 161, no. 3, pp. 31–40, 2007, doi: 10.1002/eej.20408.

References

- [94] L. Martins, J. Brito, A. Rocha, and J. Martins, “regenerative braking potential and energy simulations for a Plug-in hybrid electric vehicle under real driving conditions,” *ASME*, pp. 1–8, 2009.
- [95] B. Bolund, H. Bernhoff, and M. Leijon, “Flywheel energy and power storage systems,” *Renewable and Sustainable Energy Reviews*, vol. 11, no. 2, pp. 235–258, 2007, doi: 10.1016/j.rser.2005.01.004.
- [96] A. Boretta, “Improvements of truck fuel economy using mechanical regenerative braking,” *SAE Technical Papers*, 2010, doi: 10.4271/2010-01-1980.
- [97] W. Wu, H. Liu, J. Zhou, J. Hu, and S. Yuan, “Energy efficiency of hydraulic regenerative braking for an automobile hydraulic hybrid propulsion method,” *Int J Green Energy*, vol. 16, no. 13, pp. 1046–1053, Oct. 2019, doi: 10.1080/15435075.2019.1653875.
- [98] J. Zhang, C. Lv, J. Gou, and D. Kong, “Cooperative control of regenerative braking and hydraulic braking of an electrified passenger car,” *Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering*, vol. 226, no. 10, pp. 1289–1302, 2012, doi: 10.1177/0954407012441884.
- [99] W. J. B. Midgley, H. Cathcart, and D. Cebon, “Modelling of hydraulic regenerative braking systems for heavy vehicles,” *Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering*, vol. 227, no. 7, pp. 1072–1084, 2013, doi: 10.1177/0954407012469168.
- [100] M. Budt, D. Wolf, R. Span, and J. Yan, “A review on compressed air energy storage: Basic principles, past milestones and recent developments,” *Appl Energy*, vol. 170, pp. 250–268, May 2016, doi: 10.1016/j.apenergy.2016.02.108.
- [101] Andrew English, “How It Works: The Hybrid Air Car,” 2013. <https://www.popularmechanics.com/cars/hybrid-electric/a9252/how-it-works-the-hybrid-air-car-15724045/> (accessed Jan. 06, 2023).
- [102] C. Lv, J. Zhang, Y. Li, and Y. Yuan, “Mechanism analysis and evaluation methodology of regenerative braking contribution to energy efficiency improvement of electrified vehicles,” *Energy Convers Manag*, vol. 92, pp. 469–482, 2015, doi: 10.1016/j.enconman.2014.12.092.

References

- [103] E. Nakamura, M. Soga, A. Sakai, A. Otomo, and T. Kobayashi, “Development of electronically controlled brake system for hybrid vehicle,” *SAE Technical Papers*, no. 724, 2002, doi: 10.4271/2002-01-0300.
- [104] Y. Aoki, K. Suzuki, H. Nakano, K. Akamine, T. Shirase, and K. Sakai, “Development of hydraulic servo brake system for cooperative control with regenerative brake,” *SAE Technical Papers*, vol. 2007, no. 724, 2007, doi: 10.4271/2007-01-0868.
- [105] V. Paladini, T. Donato, A. de Risi, and D. Laforgia, “Super-capacitors fuel-cell hybrid electric vehicle optimization and control strategy development,” *Energy Convers Manag*, vol. 48, no. 11, pp. 3001–3008, 2007, doi: <https://doi.org/10.1016/j.enconman.2007.07.014>.
- [106] M. Uzunoglu and M. S. Alam, “Dynamic modeling, design and simulation of a PEM fuel cell/ultra-capacitor hybrid system for vehicular applications,” *Energy Convers Manag*, vol. 48, no. 5, pp. 1544–1553, 2007, doi: <https://doi.org/10.1016/j.enconman.2006.11.014>.
- [107] Z. Junzhi, L. Yutong, L. Chen, and Y. Ye, “New regenerative braking control strategy for rear-driven electrified minivans,” *Energy Convers Manag*, vol. 82, pp. 135–145, 2014, doi: <https://doi.org/10.1016/j.enconman.2014.03.015>.
- [108] J. Zhang, C. Lv, M. Qiu, Y. Li, and D. Sun, “Braking energy regeneration control of a fuel cell hybrid electric bus,” *Energy Convers Manag*, vol. 76, pp. 1117–1124, 2013, doi: <https://doi.org/10.1016/j.enconman.2013.09.003>.
- [109] C. Lv, J. Zhang, Y. Li, and Y. Yuan, “Mechanism analysis and evaluation methodology of regenerative braking contribution to energy efficiency improvement of electrified vehicles,” *Energy Convers Manag*, vol. 92, pp. 469–482, 2015, doi: 10.1016/j.enconman.2014.12.092.
- [110] G. Sovran and D. Blaser, “Quantifying the potential impacts of regenerative braking on a vehicle’s tractive-fuel consumption for the U.S., European, and Japanese driving schedules,” *SAE Technical Papers*, vol. 2006, no. 724, 2006, doi: 10.4271/2006-01-0664.

References

- [111] F. An and D. J. Santini, “Mass impacts on fuel economies of conventional vs. hybrid electric vehicles,” *SAE Technical Papers*, vol. 113, pp. 258–276, 2004, doi: 10.4271/2004-01-0572.
- [112] N. Brinkman, M. Wang, T. Weber, and T. Darlington, “Well-to-Wheel Analysis of Advanced Fuel/Vehicle Systems — A North American Study of Energy Use , Greenhouse Gas Emissions, and Criteria Pollutant Emissions,” *General Motors*, no. May, 2005.
- [113] UNECE (United Nations Economic Commission for Europe), “Worldwide harmonized Light vehicles Test Procedure (WLTP),” 2012. Accessed: Oct. 25, 2021. [Online]. Available: <https://wiki.unece.org/pages/viewpage.action?pageId=2523179>
- [114] S. Saxena, A. Phadke, A. Gopal, and V. Srinivasan, “Understanding fuel savings mechanisms from hybrid vehicles to guide optimal battery sizing for India,” *International Journal of Powertrains*, vol. 3, no. 3, p. 259, 2014, doi: 10.1504/ijpt.2014.064325.
- [115] A. Mahmoudzadeh Andwari, A. Pesyridis, S. Rajoo, R. Martinez-Botas, and V. Esfahanian, “A review of Battery Electric Vehicle technology and readiness levels,” *Renewable and Sustainable Energy Reviews*, vol. 78. Elsevier Ltd, pp. 414–430, 2017. doi: 10.1016/j.rser.2017.03.138.
- [116] O. M. Govardhan, “Fundamentals and Classification of Hybrid Electric Vehicles,” *International Journal of Engineering and Techniques*, vol. 3, no. 5, pp. 194–198, 2017.
- [117] University of Palermo, “I sistemi di trasmissione.” University of Palermo, pp. 1–30, 2005.
- [118] I. S. Organization, “ISO 14040:2006 - Environmental management - Life cycle assessment - Principles and framework,” 2006.
- [119] J. P. Deaton, ““How Hydraulic Hybrids Work,”” 11 April 2008. <https://auto.howstuffworks.com/hydraulic-hybrid.htm> (accessed Oct. 19, 2018).
- [120] EPA, “Hydraulic Hybrid Vehicle Layouts,” 2016. <https://www.epa.gov/cati/hydraulic-hybrid-vehicles> (accessed Nov. 11, 2020).

References

- [121] Ltd. Artemis Intelligent Power, “Hydraulic Hybrids That Go Heavy Duty.” <https://www.designnews.com/automotive-0/12-hydraulic-hybrids-go-heavy-duty> (accessed Oct. 19, 2018).
- [122] EPA, “How Parallel Hydraulic Hybrid Vehicles Work,” 2016. <https://archive.epa.gov/otaq/technology/web/html/how-it-works-parallel.html> (accessed Oct. 23, 2018).
- [123] T. R. C. Kai Loon Cheong, Perry Y. Li, “Optimal Design of Power-Split Transmissions for Hydraulic Hybrid Passenger Vehicles,” *American control conference*, 2011.
- [124] A. Boretti, “Analysis of the regenerative braking efficiency of a latest electric vehicle,” *SAE Technical Papers*, vol. 12, 2013, doi: 10.4271/2013-01-2872.
- [125] EPA, “Hydraulic Hybrid Vehicles,” 2006. <https://www.epa.gov/cati/hydraulic-hybrid-vehicles> (accessed Oct. 22, 2018).
- [126] M. P. Lammert, J. Burton, P. Sindler, and A. Duran, “Hydraulic Hybrid and Conventional Parcel Delivery Vehicles ’ Measured Laboratory Fuel Economy on Targeted Drive Cycles,” no. October, pp. 7–9, 2014, doi: 10.4271/2014-01-2375.
- [127] J. Taylor, W. Rampen, D. Abrahams, and A. Latham, “Demonstration of a Digital Displacement Hydraulic Hybrid Bus,” *JSAE Annual Congress*, pp. 1–6, 2015.
- [128] EPA, “Progress Report on Clean and Efficient Automotive Technologies Under Development at EPA,” 2004. Accessed: Mar. 01, 2021. [Online]. Available: <https://archive.epa.gov/otaq/technology/web/pdf/420r04002.pdf>
- [129] B. Wu, C. C. Lin, Z. Filipi, H. Peng, and D. Assanis, “Optimal power management for a hydraulic hybrid delivery truck,” *Vehicle System Dynamics*, vol. 42, no. 1–2, pp. 23–40, 2004, doi: 10.1080/00423110412331291562.
- [130] Z. Filipi *et al.*, “Combined optimisation of design and power management of the hydraulic hybrid propulsion system for the 6 × 6 medium truck,” *Int J Heavy Veh Syst*, vol. 11, pp. 372–402, 2004.

References

- [131] T. P. Barbosa, L. A. R. da Silva, F. J. P. Pujatti, and J. C. H. Gutiérrez, “Hydraulic hybrid passenger vehicle: Fuel savings possibilities,” *Mechanics Based Design of Structures and Machines*, vol. 0, no. 0, pp. 1–19, 2020, doi: 10.1080/15397734.2020.1714447.
- [132] A. Macor, A. Benato, A. Rossetti, and Z. Bettio, “Study and Simulation of a Hydraulic Hybrid Powertrain,” *Energy Procedia*, vol. 126, pp. 1131–1138, 2017, doi: 10.1016/j.egypro.2017.08.279.
- [133] P. Matheson and J. Stecki, “Development and simulation of a hydraulic-hybrid powertrain for use in commercial heavy vehicles,” *SAE Technical Papers*, vol. 112, no. May 2021, pp. 114–123, 2003, doi: 10.4271/2003-01-3370.
- [134] F. A. Bender, T. Bosse, and O. Sawodny, “An investigation on the fuel savings potential of hybrid hydraulic refuse collection vehicles,” *Waste Management*, vol. 34, no. 9, pp. 1577–1583, 2014, doi: 10.1016/j.wasman.2014.05.022.
- [135] S. Hui, Y. Lifu, J. Junqing, and L. Yanling, “Control strategy of hydraulic/electric synergy system in heavy hybrid vehicles,” *Energy Convers Manag*, vol. 52, no. 1, pp. 668–674, 2011, doi: 10.1016/j.enconman.2010.07.045.
- [136] EPA, “Hydraulic Hybrid Research,” 2016. <https://archive.epa.gov/otaq/technology/web/html/research-hhvs.html#overview> (accessed Oct. 23, 2018).
- [137] H. Zhou, Z. Xu, and L. Liu, “Design and validation of a novel hydraulic hybrid vehicle with wheel motors,” vol. 103, no. 1, pp. 1–25, 2020, doi: 10.1177/0036850419878024.
- [138] T. P. Barbosa, L. A. R. da Silva, F. J. P. Pujatti, and J. C. H. Gutiérrez, “Hydraulic hybrid passenger vehicle: Fuel savings possibilities,” *Mechanics Based Design of Structures and Machines*, vol. 0, no. 0, pp. 1–19, 2020, doi: 10.1080/15397734.2020.1714447.
- [139] F. A. Bender, T. Bosse, and O. Sawodny, “An investigation on the fuel savings potential of hybrid hydraulic refuse collection vehicles,” *Waste Management*, vol. 34, no. 9, pp. 1577–1583, 2014, doi: 10.1016/j.wasman.2014.05.022.

References

- [140] A. Macor, A. Benato, A. Rossetti, and Z. Bettio, "Study and Simulation of a Hydraulic Hybrid Powertrain," *Energy Procedia*, vol. 126, pp. 1131–1138, 2017, doi: 10.1016/j.egypro.2017.08.279.
- [141] P. Achten, G. VAEL, M. I. SOKAR, and T. KOHMASCHER, "Design Series And Fuel Economy Hybrid Of A Vehicle Hydraulic," *Proceedings of the 7th JFPS International Symposium on Fluid Power*, vol. 1, no. 1, pp. 47–52, 2008.
- [142] S. Hui, Y. Lifu, J. Junqing, and L. Yanling, "Control strategy of hydraulic/electric synergy system in heavy hybrid vehicles," *Energy Convers Manag*, vol. 52, no. 1, pp. 668–674, 2011, doi: 10.1016/j.enconman.2010.07.045.
- [143] C. N. Eastwick and G. Johnson, "Gear windage: A review," *Journal of Mechanical Design, Transactions of the ASME*, vol. 130, no. 3, pp. 1–6, 2008, doi: 10.1115/1.2829983.
- [144] Y. Diab, F. Ville, and P. Velez, "Investigations on power losses in high-speed gears," *Proceedings of the Institution of Mechanical Engineers, Part J: Journal of Engineering Tribology*, vol. 220, no. 3, pp. 191–198, 2006, doi: 10.1243/13506501JET136.
- [145] Th. von Karman, *On Laminar and Turbulent Friction*. National Advisory Committee for Aeronautics, 1921.
- [146] P. R. N. Childs, *Rotating Flow*. Elsevier, 2011. doi: 10.1016/C2009-0-30534-6.
- [147] C. Mayes, H. Schlichting, E. Krause, H. J. Oertel, and K. Gersten, *Boundary-Layer Theory*. Springer Berlin Heidelberg, 2003. [Online]. Available: <https://books.google.co.uk/books?id=8YugVtom1y4C>
- [148] J. W. Daily and R. E. Nece, "Chamber dimension effects on induced flow and frictional resistance of enclosed rotating disks," *Journal of Fluids Engineering, Transactions of the ASME*, vol. 82, no. 1, pp. 217–230, 1960, doi: 10.1115/1.3662532.
- [149] R. E. Nece and J. W. Daily, "Roughness effects on frictional resistance of enclosed rotating disks," *Journal of Fluids Engineering, Transactions of the ASME*, vol. 82, no. 3, pp. 553–560, 1960, doi: 10.1115/1.3662656.

References

- [150] A. T. Ippen, “The influence of viscosity on centrifugal pump performance, Trans. ASME, Vol. 68 (1946), pp. 823-848,” *Trans. ASME*, vol. 68, pp. 823–848, 1946.
- [151] N. J. Soo, S.L. Princeton, “Laminar flow over an enclosed rotating disk,” *Trans. ASME*, vol. 80, pp. 287–296, 1958.
- [152] M. R. Etemad, K. Pullen, C. B. Besant, and N. Baines, “Evaluation of Windage Losses for High-Speed Disc Machinery,” *Proceedings of the Institution of Mechanical Engineers, Part A: Journal of Power and Energy*, vol. 206, no. 3, pp. 149–157, 1992, doi: 10.1243/PIME_PROC_1992_206_025_02.
- [153] H. N. Ketola and J. M. McGrew, “Pressure, frictionl resistance, and flow characteristics of the partially wetted rotating disk,” *J Tribol*, vol. 90, no. 2, pp. 395–404, 1968, doi: 10.1115/1.3601573.
- [154] F. Schultz-Grunow, “Der Reibungswiderstand rotierender Scheiben in Gehäusen,” *ZAMM - Journal of Applied Mathematics and Mechanics / Zeitschrift für Angewandte Mathematik und Mechanik*, vol. 15, no. 4, pp. 191–204, 1935, doi: 10.1002/zamm.19350150403.
- [155] Stodola Aurel and Louis C. Loewenstein, *Steam Turbines : With an Appendix on Gas Turbines and the Future of Heat Engines*. New York: D. Van Nostrand, 1905.
- [156] A. A. Lord, “An experimental investigation of geometric and oil flow effects on gear windage and meshing losses,” 1998.
- [157] S. S. H. Anderson, N. E., Loewenthal S.H., “Spur gear systems efficiency at part and full load,” *NASA tech paper 1622*, no. February, 1980.
- [158] P. H. Dawson, “Windage Loss in Larger High-Speed Gears.,” *Proceedings of the Institution of Mechanical Engineers, Part A: Journal of Power and Energy*, vol. 198, no. 1, pp. 51–59, 1984, doi: 10.1243/pime_proc_1984_198_007_02.
- [159] M. Y. Susumu Matsumoto, Sadao Asanabe, Kazuya Takano, “Evaluation method of power loss in high-speed gears,” *Elsevier Science Publ Co*, vol. 3, pp. 1165–1170, 1985.
- [160] R. W. Mann and C. H. Marston, “Friction drag on bladed disks in housings,” *Journal of Fluids Engineering, Transactions of the ASME*, vol. 83, no. 4, pp. 719–723, 1961, doi: 10.1115/1.3662307.

References

- [161] A. A. Lord, “An investigation into the mechanisms of windage power loss,” Final year thesis, University Wales Swansea, Mech. Eng., 1993.
- [162] Z. Zhang, “Pelton turbines,” *Pelton Turbines*, no. index 1, pp. 1–311, 2016, doi: 10.1007/978-3-319-31909-4.
- [163] D. Polichronis, R. Evaggelos, G. Alcibiades, G. Elias, and P. Apostolos, “Turbocharger Lubrication-Lubricant Behavior and Factors That Cause Turbocharger Failure,” *International Journal of Automotive Engineering and Technologies*, vol. 2, pp. 40–54, 2013.
- [164] C. Bell, *Maximum Boost: Designing, Testing and Installing Turbocharger Systems*. 1997.
- [165] BP, “BP turbo oil 2380,” 2012. [https://msdspds.bp.com/bpglis/FusionPDS.nsf/Files/802577040037FB4B80257AD800035A64/\\$File/BPXE-92RGEV.pdf](https://msdspds.bp.com/bpglis/FusionPDS.nsf/Files/802577040037FB4B80257AD800035A64/$File/BPXE-92RGEV.pdf) (accessed Jun. 15, 2018).
- [166] Castrol, “Edge 5w-20,” 2012. [https://msdspds.castrol.com/bpglis/fusionpds.nsf/files/3d49c4140b440936802579d10048864a/\\$file/bpxe-8syfhh.pdf](https://msdspds.castrol.com/bpglis/fusionpds.nsf/files/3d49c4140b440936802579d10048864a/$file/bpxe-8syfhh.pdf) (accessed Jun. 15, 2018).
- [167] Joseph Shigley, *Mechanical Engineering Design*. McGraw-Hill Education (India) Pvt Limited, 2002.
- [168] Society of Tribologists and Lubrication Engineers, “Journal Bearings and Their Lubrication,” 2003. <https://www.machinerylubrication.com/Read/779/journal-bearing-lubrication> (accessed Jul. 31, 2018).
- [169] V. S. Knežević, Darko, “Mathematical Modeling Of Changing Of Dynamic Viscosity, As A Function Of Temperature And Pressure, Of Mineral Oils For Hydraulic Systems,” vol. 4, pp. 27–34, 2006.
- [170] M. Leader, “Understanding Journal Bearings,” 2001. Accessed: Sep. 20, 2018. [Online]. Available: http://edge.rit.edu/edge/P14453/public/Research/2-_LEADER_-_Understanding_Journal_Bearings.pdf
- [171] C. E. Brennen, “Cavitation,” Cambridge University Press, 2011. Accessed: Jun. 16, 2020. [Online]. Available: <https://app.knovel.com/hotlink/khtml/id:kt0089F2C2/hydrodynamics-pumps/cavitation>

References

- [172] C. Pumps, “Pump hydraulics,” 2013. Accessed: May 30, 2018. [Online]. Available: <https://www.tuthillpump.com/en-gb/>
- [173] Sensorland, “How deadweight tester work.” <https://www.sensorland.com/HowPage079.html> (accessed Jan. 12, 2022).
- [174] D. Walz, S. R. Caplan, D. R. L. Scriven, and D. C. Mikulecky, *Methods of mathematical modelling*. 1995. doi: 10.1007/978-3-0348-7318-5_2.
- [175] M. Tenenbaum and H. Pollard, *Ordinary differential equations: an elementary textbook for students of mathematics, engineering, and the sciences*. Harper & Row, 1963.
- [176] J. C. Butcher, *Numerical Methods for Ordinary Differential Equations*. Wiley, 2003.
- [177] M. A. Stenhjem and NTNU Norwegian University of Science and Technology, “Analysis of Accumulator Discharge,” 2017.
- [178] Reasontek, “Bladder accumulators.” <https://www.reasontek.com/bladder-accumulators/> (accessed Mar. 15, 2022).
- [179] E. J. Finnemore and J. B. Franzini, *Fluid Mechanics With Engineering Applications*. McGraw-Hill Education, 2001.
- [180] R. F. B. Y. Nakayama, “Introduction to Fluid Mechanics,” 2012, doi: 10.1111/j.1549-8719.2009.00016.x.Mechanobiology.
- [181] J. A. Sullivan, *Fluid power: theory and applications*. Reston Pub. Co., 1982.
- [182] D. S. Miller, *Internal Flow Systems*. BHRA, 1990.
- [183] H. Boysen, “Kv: what , why , how , whence ?,” 2011. <https://studylib.net/doc/8896519/kv---what--why--how--whence%3F> (accessed Jun. 15, 2018).
- [184] R. L. Mott, *Applied Fluid Mechanics*, no. v. 1. Prentice Hall, 2006.
- [185] R. S. Subramanian, “Minor Losses.” Accessed: May 30, 2018. [Online]. Available: <https://web2.clarkson.edu/projects/subramanian/ch330/notes/Minor%20Losses.pdf>
- [186] Joseph Shigley, *Mechanical Engineering Design*, vol. New York,. McGraw-Hill Education - Europe, 2002. doi: 10.1007/s13398-014-0173-7.2.

References

- [187] Y. Wang and Q. J. Wang, “Stribeck Curves,” in *Encyclopedia of Tribology*, Q. J. Wang and Y.-W. Chung, Eds. Boston, MA: Springer US, 2013, pp. 3365–3370. doi: 10.1007/978-0-387-92897-5_148.
- [188] A. Palmgren, *Ball and roller bearing engineering / by Arvid Palmgren ; translated by Gunnar Palmgren and Bryce Ruley.*, 2nd ed. Philadelphia: SKF Industries, 1946.
- [189] T. A. Harris and M. N. Kotzalas, *Essential Concepts of Bearing Technology*. CRC Press, 2006. doi: <https://doi.org/10.1201/9781420006599>.
- [190] NTN, “Ball and roller bearings,” 2009. <https://www.ntnglobal.com/en/products/catalog/en/2203/index.html> (accessed Jul. 28, 2020).
- [191] Y. Diab, F. Ville, and P. Velez, “Windage Losses in High Speed Gears — Preliminary Experimental,” vol. 126, no. September 2004, pp. 903–908, 2013, doi: 10.1115/1.1767815.
- [192] British Standards Institution, “Field acceptance tests to determine the hydraulic performance of hydraulic turbines, storage pumps and pump-turbines BS EN 60041:1995.” 1995.
- [193] E. Tondell, “CO₂-expansion work recovery by impulse turbine,” Norwegian University of Science and Technology, 2006. Accessed: Aug. 06, 2020. [Online]. Available: https://ntnuopen.ntnu.no/ntnu-xmlui/bitstream/handle/11250/233323/121953_FULLTEXT01.pdf?sequence=1&isAllowed=y
- [194] Buckingham, “Windage resistance of steam turbine wheels,” Forgotten Books , 1914.
- [195] MathWorks, “Simulink and Model-Based Design.” <https://it.mathworks.com/help/simulink/index.html> (accessed May 18, 2020).
- [196] Koyo, “Bearing Mounting.” <https://koyo.jtekt.co.jp/en/support/bearing-knowledge/15-3000.html> (accessed Dec. 09, 2019).
- [197] Schaeffler, “Super precision bearings-Schaeffler catalog,” 2019. https://www.schaeffler.com/remotemedien/media/_shared_media/08_media_library/01_publications/schaeffler_2/catalogue_1/downloads_6/sp1_de_en.pdf (accessed Sep. 30, 2019).

References

- [198] E. Sjöberg, “Friction Characterization of Turbocharger Bearings,” KTH Industrial Engineering and Management, 2013. Accessed: Jul. 09, 2018. [Online]. Available: <http://www.diva-portal.org/smash/get/diva2:644848/FULLTEXT01.pdf>Mar
- [199] SKF, “Bearing self study guide,” 2008. <https://www.skf.com/binary/12-69177/457640.pdf> (accessed Aug. 06, 2019).
- [200] MACHiNE Design, “What’s the Difference Between Bearings?” <https://www.machinedesign.com/whats-difference-between/what-s-difference-between-bearings-1> (accessed Sep. 13, 2019).
- [201] SKF, “Angular contact ball bearings.” <https://www.skf.com/in/products/bearings-units-housings/ball-bearings/angular-contact-ball-bearings/index.html> (accessed May 15, 2019).
- [202] H. Aghaali and H. E. Angstrom, “Temperature estimation of turbocharger working fluids and walls under different engine loads and heat transfer conditions,” *SAE Technical Papers*, vol. 6, 2013, doi: 10.4271/2013-24-0123.
- [203] N. KARAMANIS, “Inlet and exit flow characteristics of mixed flow turbines in advanced automotive turbocharging,” PhD Thesis-Imperial College of Science, Technology and Medicine, University of London, 2000.
- [204] A. Pesyridis, W. S. I. W. Salim, and R. F. Martinez-Botas, “Turbocharger matching methodology for improved exhaust energy recovery,” *Institution of Mechanical Engineers - 10th International Conference on Turbochargers and Turbocharging*, no. November 2017, pp. 203–218, 2012, doi: 10.1533/9780857096135.4a.203.
- [205] “Engineering Toolbox-air properties,” 2005. https://www.engineeringtoolbox.com/dry-air-properties-d_973.html (accessed Nov. 23, 2021).
- [206] A. P. Dale, “Radial, vaneless, turbocharger turbine performance,” Ph.D. Thesis, Imperial College of Science, Technology, and Medicine, University of London, England, 1990.
- [207] Szanca, E. M., and Szhum, H. J., “Experimental determination of aerodynamic performance.” NASA BP-290, vol. 3, Chapter 12, 1972.

References

- [208] R. H. Perry, *Perry's Chemical Engineers' Handbook*, 6th editio. 1984.
- [209] Hannu Jääskeläinen-Diesel.net, “Diesel Exhaust Gas,” 2020. https://dieselnet.com/tech/diesel_exh.php (accessed Nov. 25, 2021).
- [210] E. Pipitone and S. Caltabellotta, “The Potential of a Separated Electric Compound Spark-Ignition Engine for Hybrid Vehicle Application,” *J Eng Gas Turbine Power*, vol. 144, no. 4, Apr. 2022, doi: 10.1115/1.4053393.
- [211] K. Nishiwaki, M. Iezawa, H. Tanaka, T. Goto, and B. An, “Development of High Speed Motor and Inverter for Electric Supercharger,” *Mitsubishi Heavy industries, Ltd., technical review*, 2013, doi: 10.4271/2013-01-0931.
- [212] M. Alshammari, N. Xypolitas, and A. Pesyridis, “Modelling of Electrically-Assisted Turbocharger Compressor Performance,” *Energies (Basel)*, vol. 12, no. 6, 2019, doi: 10.3390/en12060975.
- [213] A. Darlington, D. Cieslar, N. Collings, and K. Glover, “Assessing Boost-Assist Options for Turbocharged Engines Using 1-D Engine Simulation and Model Predictive Control,” in *SAE 2012 International Powertrains, Fuels & Lubricants Meeting*, Sep. 2012. doi: <https://doi.org/10.4271/2012-01-1735>.
- [214] T. Suzuki, Y. Hirai, and N. Ikeya, *Electrically Assisted Turbocharger as an enabling technology for improved fuel economy in New European Driving Cycle operation*. Woodhead Publishing Limited, 2014. doi: 10.1016/B978-0-081000-33-5.50018-4.
- [215] D. Gerada, X. Huang, C. Zhang, H. Zhang, X. Zhang, and C. Gerada, “Electrical Machines for Automotive Electrically Assisted Turbocharging,” *IEEE/ASME Transactions on Mechatronics*, vol. 23, no. 5, pp. 2054–2065, 2018, doi: 10.1109/TMECH.2018.2849081.
- [216] N. Terdich, R. F. Martinez-Botas, D. A. Howey, C. D. Copeland, and A. Costall, “Off-road diesel engine transient response improvement by electrically assisted turbocharging,” in *SAE Technical Papers*, 2011. doi: 10.4271/2011-24-0127.
- [217] K. Ekberg and L. Eriksson, “Improving Fuel Economy and Acceleration by Electric Turbocharger Control for Heavy Duty Long Haulage,” *IFAC-PapersOnLine*, vol. 50, no. 1, pp. 11052–11057, 2017, doi: <https://doi.org/10.1016/j.ifacol.2017.08.2486>.

References

- [218] E. Winward *et al.*, “Performance testing of an electrically assisted turbocharger on a heavy duty diesel engine,” in *12th International Conference on Turbochargers and Turbocharging*, May 2016.
- [219] C. Balis, C. Middlemass, and S. M. Shahed, “Design & Development Of E-Turbo For Suv And Light Truck Applications,” in *Diesel Engine Emissions Reduction Conference, Newport, RI*, 2003.
- [220] Garrett, “Garrett Performance catalog Turbochargers | Intercoolers | Accessories,” 2021. Accessed: Jun. 15, 2022. [Online]. Available: <https://www.garrettmotion.com/racing-and-performance/performance-catalog/turbo/gtx4508r/>
- [221] J. Lee, C. Park, Y. Kim, Y. Choi, J. Bae, and B. Lim, “Effect of turbocharger on performance and thermal efficiency of hydrogen-fueled spark ignition engine,” *Int J Hydrogen Energy*, vol. 44, no. 8, pp. 4350–4360, 2019, doi: <https://doi.org/10.1016/j.ijhydene.2018.12.113>.
- [222] L. Bao *et al.*, “Development of a turbocharged direct-injection hydrogen engine to achieve clean, efficient, and high-power performance,” *Fuel*, vol. 324, p. 124713, 2022, doi: <https://doi.org/10.1016/j.fuel.2022.124713>.
- [223] T. Sugawara, T. Kanazawa, N. Imai, and Y. Tachibana, “Development of Motorized Turbo Compressor for Clarity Fuel Cell,” in *SAE Technical Papers*, Mar. 2017, vol. 2017-March, no. March. doi: 10.4271/2017-01-1187.
- [224] European Commission-Horizon 2020, “Compact, light, efficient and reliable turbo compressor for fuel cell vehicles,” 2019. <https://cordis.europa.eu/project/id/790379> (accessed Jun. 24, 2022).
- [225] P. Fröhlich and P. Celeroton, “Development of an oil free turbo compressor for mobile fuel cell applications-challenges and results,” *Fuel Cell Conference FC³ Chemnitz*, 2019.
- [226] J. Lozier, E. Oklejas, and M. Silbernagel, “The hydraulic turbochargerTM: a new type of device for the reduction of feed pump energy consumption in reverse osmosis systems,” *Desalination*, vol. 75, pp. 71–83, 1989, doi: [https://doi.org/10.1016/0011-9164\(89\)85006-4](https://doi.org/10.1016/0011-9164(89)85006-4).
- [227] Y. Ji, X. Li, and H. Li, “Hydraulic turbocharger for energy recycling in the reverse osmosis seawater desalination system,” *IOP Conf Ser Earth*

References

- Environ Sci*, vol. 467, p. 12047, Jun. 2020, doi: 10.1088/1755-1315/467/1/012047.
- [228] Airgas, “Material Safety Datasheet Nitrogen,” 2002. <https://www.airgas.com/msds/001040.pdf> (accessed Dec. 14, 2018).
- [229] A. Židonis and G. A. Aggidis, “Pelton turbine: Identifying the optimum number of buckets using CFD,” *Journal of Hydrodynamics*, vol. 28, no. 1, pp. 75–83, Feb. 2016, doi: 10.1016/S1001-6058(16)60609-1.
- [230] M. Nechleba, *Hydraulic turbines : their design and equipment*. Artia Prague, 1957. Accessed: Jun. 22, 2022. [Online]. Available: <https://archive.org/details/HydraulicTurbinesTheirDesignAndEquipmentNechleba/page/n1/mode/2up>
- [231] B. Nasir, “Design of high efficiency Pelton turbine for microhydropower plant,” *Int J Electr Eng Technol (IJEET)*, vol. 4, pp. 171–183, Jun. 2013.
- [232] Ferrari Giancarlo, *Motori a Combustione Interna*, 2nd ed. EdiText, 1996.
- [233] J. M. Owen and R. H. Rogers, *Flow and Heat Transfer in Rotating-disc Systems: Rotor-stator systems*. Research Studies Press, 1989. [Online]. Available: <https://books.google.co.uk/books?id=SNZSAAAAMAAJ>

APPENDIX A: Calculation for safe Nitrogen discharge after testing

According to the safety requirement in the MSDS of Nitrogen, to avoid asphyxia, the amount of Nitrogen in the room should be such to maintain Oxygen above 19.5% [228]. The room in which the rig is situated has a volume of 500 m³; the air volume splits then as follow according to air composition: 78.084% of Nitrogen correspondent to 390.42 m³, 20.946% of Oxygen correspondent to 104.73 m³ and 0.97% of other gases correspondent to 4.85 m³. Assuming the maximum charging pressure of 300 bar and the volume of accumulator full expanded bladder of 17.8 L, according to maximum gas volume allowable in the bladder, the amount of Nitrogen inside the accumulator can be calculated through the perfect gas equation and it is correspondent to 6kg at ambient temperature of 298 K. The equivalent volume of Nitrogen released at ambient pressure into the room is then 5.34 m³. The new percentage of air composition inside the room, considering the addition of Nitrogen would be: Nitrogen 79.15%, Oxygen 19.88% and Other gases 0.97%. The percentage of Oxygen in the room would be 19.88%, above the minimum allowed for safety conditions, in the worst-case scenario of having closed doors/windows and no ventilation.

APPENDIX B: Test matrix free running Pelton experimental campaign

Each run with same combination of parameters was carried out at least 3 times. Some casings were not tested further and not included in the matrix as they showed at low pressure overflowing, thus they were discarded.

<i>Run</i>	<i>Pressure</i>	<i>Nozzle Diameter</i>	<i>Nozzle number</i>	<i>Casing</i>
1	40	1.5	2	No Casing
2	80	1.5	2	No Casing
3	120	1.5	2	No Casing
4	160	1.5	2	No Casing
5	200	1.5	2	No Casing
6	280	1.5	2	No Casing
7	40	1.5	1	No Casing
8	80	1.5	1	No Casing
9	120	1.5	1	No Casing
10	160	1.5	1	No Casing
11	200	1.5	1	No Casing
12	280	1.5	1	No Casing
13	40	1.1	2	No Casing
14	80	1.1	2	No Casing
15	120	1.1	2	No Casing

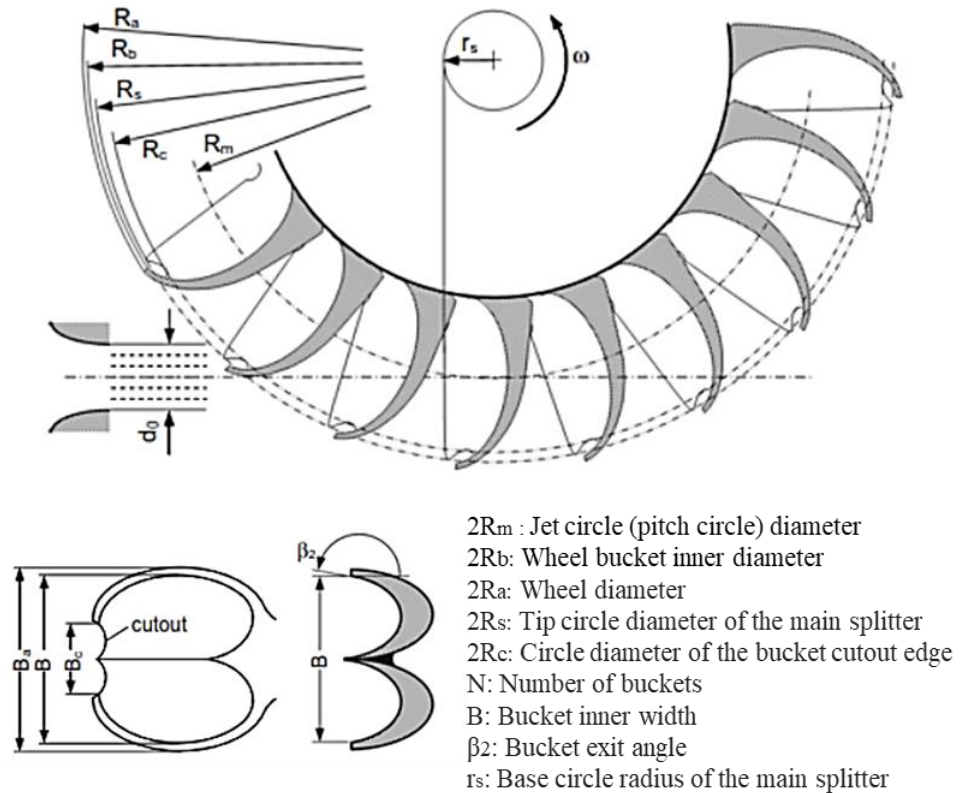
APPENDIX B

16	160	1.1	2	No Casing
17	40	1.1	1	No Casing
18	80	1.1	1	No Casing
19	120	1.1	1	No Casing
20	160	1.1	1	No Casing
21	40	1.5	1	Cover 0
22	80	1.5	1	Cover 0
23	120	1.5	1	Cover 0
24	160	1.5	1	Cover 0
25	280	1.5	1	Cover 0
26	80	1.5	1	Cover 1
27	120	1.5	1	Cover 1
28	160	1.5	1	Cover 1
29	220	1.5	1	Cover 1
30	40	1.5	1	Cover 2
31	80	1.5	1	Cover 2
32	40	1.5	1	Cover 3
33	80	1.5	1	Cover 3
34	40	1.5	1	Cover 4
35	80	1.5	1	Cover 4
36	120	1.5	1	Cover 4
37	160	1.5	1	Cover 4
38	200	1.5	1	Cover 4
39	280	1.5	1	Cover 4
40	40	1.5	2	Cover 4
41	80	1.5	2	Cover 4
42	120	1.5	2	Cover 4
43	160	1.5	2	Cover 4
44	200	1.5	2	Cover 4
45	280	1.5	2	Cover 4

Appendix Table 1

APPENDIX C: Pelton wheel optimization

The Pelton wheel should be designed and manufactured determining the parameters in Appendix Figure 1. The fundamental design of these characteristics depends on the flow specification in the turbine system and specifically on the permissible rotational speed of the assembly. The bucket exit angle β_2 for example is determined considering the required exit flow conditions and the optimum bucket number can be derived with coincidence and symmetry conditions. Number of buckets, bucket radial position and bucket angular position are interrelated characteristic of the hydraulic wheel and identifying the best combination leads to efficiency improvement and can be beneficial as well regarding manufacturing complexity and costs [229]. The specific problems formulation can be found in Zhang [162].



Appendix Figure 1 Typical geometric parameter specification of a Pelton wheel [162]

The classic hydraulic design to determine the available hydraulic power supply and the size for a Pelton turbine requires, as fundamental parameters, the desired flow rate and the net head on the pressure side. The first steps of the design process are related to the dimensioning of the wheel, the choice of rotational speed and the number of injectors. To understand the wheel dimensioning, some essential parameters must then be introduced. The peripheral speed coefficient k_m is defined as the ratio between the peripheral speed U_m of the Pelton wheel on the jet circle to the jet speed C_0 . The value of the peripheral speed is usually considered in the range between 0.45-0.48 to have maximum efficiencies of the wheel. The specific speed n_q is defined as per equation C.1 and it is a parametric quantity widely used in rotating fluid machinery. $\dot{Q}_{jet}$ is the volumetric flow rate and H is the pressure head when calculating the specific speed and the formula is normalised to have same unit measures. The specific speed is essentially a geometric parameter which represents the diameter ratio $\frac{d_0}{D_m}$. It is particularly used for a Pelton turbine design when a flow rate and head pressure are given.

$$n_q = n \frac{\dot{Q}_{jet}^{1/2}}{H^{3/4}} \quad C.1$$

Having the basic parameters of a Pelton turbine, the algorithm for dimensioning the Pelton wheel is based on the indications arising from established design procedure in Engineering practice and the specific equation are available in [162]. The main outcomes are the following:

- The bucket width B (Appendix Figure 1) should be approximately 3 times the jet diameter d_0 and the pitch circle diameter D_m should be greater than $3B$.
- The peripheral speed coefficient k_m should be kept in the range of 0.45-0.48.
- The specific speed referred to the nominal flow rate at one injector is usually $n_q < 0.13$

Another essential step in the design process is the determination of the minimum number of buckets in the Pelton wheel; this helps indeed in avoiding energy losses associated with the counter torque of the jet entering the bucket and in ensuring that no liquid particles are lost during the transition from one bucket to the next. The study is done having the precondition that the turbine is working under nominal flow rate and that no liquid in the jet bypasses unused the wheel. The analysis is carried out considering the outmost jet layer in the turbine and geometrical constraint for the jet dynamic on the buckets. In the Engineering practice, however, the number of buckets is usually higher than the minimum calculated. The practical criteria to optimise the number of buckets, is based on the coincidence and symmetry conditions, which are depicted in detail in section 5.4 of Zhang [162]. A final empirical formula for the optimised number of bucket is in equation C.2 [162], [230], [231]. The equation can be also adapted with the term on the left knowing the pitch circle diameter D_m and the injector jet diameter d_0 . The derivation details of the formula and the complete assumptions can be found in section 5.5 of Zhang [162].

$$N_b = 15 + 1.3 \frac{k_m}{n_q} = 15 + 0.5 \frac{D_m}{d_0} \quad C.2$$

On a final note, it must be noted that the Pelton wheel in the Research project was manufactured using some standard techniques as nose ball cutting, however the shape of the bucket needs to be optimised as well to obtain high performances. As a rule of thumb, the liquid injected into the bucket should be exposed only to a continuous and smooth curve with no sudden changes in gradient in any direction. In section 18.2 of Zhang [162] the mathematical model for the bucket shape design optimization is characterised using the Pelton geometrical main performance parameters and described in details; the method proposed is to have the cross sections defined by translated and rotated ellipses. The boundary conditions are applied to comply with the optimization guidelines previously shown that set entry and exit condition of the flow and 6 unknown parameters equations are solved to obtain the final design.

APPENDIX C

According to the calculations previously shown, the number of buckets optimised for the Pelton characteristics of KERB should have been 24, however the manufacturing limitation led the count to be 12. Although the proposed optimization methods, the turbine tests in an experimental environment remain the most credible and reliable way to verify and optimise the hydraulic design of a prototype wheel [162].

APPENDIX D: Theoretical approach to Turbocharger contribute to bmep

The *bmep* is usually used to compare vehicles independently from the size of the engine, it is then considered an efficient way of obtaining the general fit for purpose operations of the turbocharger prototype. The physical measure unit is the same as a pressure (Pa), however the operative meaning is related to the work losses at the engine. Out of the Gross Indicated Work per cycle $W_{i,c}$ (equation D.1) delivered to the piston over compression and expansion strokes in a ICE, only part is actively transferred to the output shaft as brake work. The other part is shared among other tasks at the engine which are to induct and exhaust the gas from the cylinder, to drive engine ancillaries (water pumps, etc.) and to overcome the surface frictions between the moving parts at the engine.

$$W_{i,c} = W_p + W_f + W_a + W_b \quad D.1$$

The pumping work, friction work, ancillaries work and brake work in equation D.1, that sum together to give then the work per cycle, can be similarly represented in terms of mean effective pressure in equation D.2. The mean effective pressure calculated from the Brake torque generated at the crankshaft is called brake mean effective pressure. The *bmep* represents the measure of a cylinder load proportional to brake torque for a given engine and it is calculated as the ratio between the brake power and the product of the engine swept volume and the engine speed. It measures the ability to provide high air flow and use it effectively over the full range. For a given a speed regime, then, the *bmep* parameter gives then an indication of how efficiently the engine is design relatively to the displaced volume and its output energy.

$$Imep = pmep + fmep + amep + bmep \quad D.2$$

To apply the definition in KERB application and calculate the potential $bmep$, the overall friction losses indicated as $fmep_c$ need to be calculated; the ancillaries and pumping work are not considered in this preliminary case study. The to calculate the new $bmep_c$ is defined in equation D.3. The value of $imep_c$ can be calculated through equation D.4 [232][57]. The parameters shown are respectively: λ_v engine load factor, p_o ambient air pressure, p_c boost air pressure, δ_a air density, δ_f fuel density, H_i fuel calorific value, η_i fuel efficiency and k polytropic index.

$$bmep_c = imep_c - fmep_c \quad D.3$$

$$imep_c = \mu_c imep_0 \frac{\lambda'_v}{\lambda_v} + (p_c - p_o) \quad D.4$$

$$imep_0 = \lambda_v \frac{\delta_a H_i}{\left(\alpha + \frac{1}{\delta'}\right)} \eta_i \quad D.5$$

$$\frac{1}{\delta'} = \frac{\delta_a}{\delta_f} \quad D.6$$

$$\mu_c = \frac{p_o}{p_c} \sqrt{\frac{T_o}{T_c}} \quad D.7$$

$$\frac{\lambda'_v}{\lambda_v} = 1 + \frac{p_c - p_o}{kp_c(\rho - 1)} \quad D.8$$

The value of $fmep_c$ can be ideally calculated through the Chen-Flynn correlation model [210]. In equation D.9 u is the normalised piston speed $u = u_m/u_{max}$ and A, B, C and D are parameters determined using ordinary least-squares

APPENDIX D

method to approximate the solution of data fitting of the experimental values of f_{mep_c} and i_{mep_c} .

$$f_{mep_c} = A + B * i_{mep_c} + C * u + D * u^2 \quad D.9$$

As an example to understand the range of values that can be obtained, Appendix Table 2 shows some results for a gasoline baseline engine.

Symbol	Denomination	Value
δ_a	Air Density	1.22 kg/m ³
/	Number of cylinders	4
/	Bore	72.9mm
/	Stroke	80.9mm
/	Maximum mean speed piston	17 m/s
α	Engine air fuel mass ratio	14.7
A	Friction loss parameter	0.6304 bar
B	Friction loss parameter	0.0398 bar
C	Friction loss parameter	-0.0580 bar
D	Friction loss parameter	1.0580
ε	Engine cycle parameter	2 (4-stroke engine)
V	Engine displacement volume	1352 cc
c_p	Specific heat air	1.005 kJ/kgK
ρ	Engine compression ratio	11
/	Maximum bmep	12.6 bar at 3780 rpm
/	Maximum power	73.5 kW at 5880 rpm
/	Maximum bsfc	232.9 g/kWh

Appendix Table 2 Parameters used to calculate ideal b_{mep_c} in [210]

Typically, the b_{mep} of turbocharged diesel engines can go from 7-12 bar depending on the application (light and heavy-duty vehicles) and it goes up to 15-18 for marine applications [57], [232].