Tampere University of Applied Sciences

Determination of High-Speed Turbo Compressor's Heat Recovery Qualities

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ABSTRACT

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Determination of High-Speed Turbo Compressor's Heat Recovery Qualities

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This thesis was commissioned by Tamturbo Oyj, a Finnish high-speed turbo compressor manufacturer. It studied the physics and thermodynamics of centrifugal air compressors and emphasized heat recovery systems incorporated into them. The goal was to analyze the factors influencing the thermal energy recovery from air compression and to create a user-friendly calculation tool. Using the data sets from the prior tests conducted by Tamturbo, an Excel-based tool was developed with a simplified interface for easy usability.

The case study focused on a Tamturbo TT325 turbo compressor, calculating the heat recovery in four scenarios representing summer and winter months in Houston, Texas, and Tampere, Finland. The results indicated a higher heat recovery in the hot, humid summer months relative to colder winter seasons. The tool enables the sales team to showcase the heat recovery qualities and compare the compressors' performance based on site location and parameters set for the unit, like pressure output and flow rate. The calculation tool streamlines the presentation of recovery capabilities to customers and offers a quick and easy method to perform comparisons.

Key words: air compressor, high-speed turbo compressor, heat recovery

CONTENTS

1 INTRODUCTION

Air compressors are a common sight in many industrial plants from food and beverage producers and pulp and paper sites to semiconductor manufacturers. Around 10 % of global industrial electricity usage is consumed by air compressors (R. Saidur, N.A. Rahim, M. Hasanuzzaman 2010). In exchange, a lot of heat is produced in the compression process. Currently the methods to recover the thermal energy are still not widely adopted and the potential benefits in terms of energy and cost savings are not yet well realised.

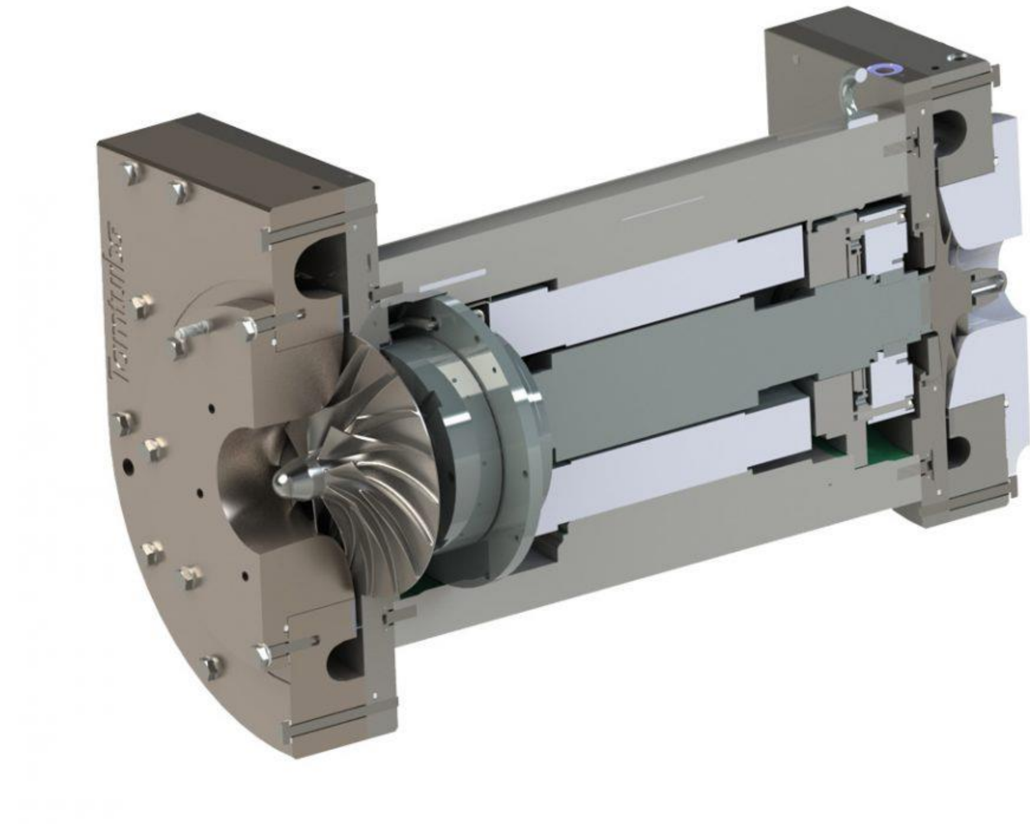
This thesis describes the current state in the industrial air compressor market. It also explains the working principle of centrifugal air compressors and the physics behind air compression as well as the thermodynamics associated with the process. The work also describes the potential of heat recovery as a part of air compressor technology and explains the sources of the thermal energy and how it can be collected. A calculation tool is created as a result of the research. It can be used to easily compare the heat recovery qualities of a turbo compressor at various climate conditions with different parameters set for the compressor unit. The purpose was to make it easily usable without requiring a thorough knowledge of how a turbo compressor operates. Only the necessary variables are needed to produce easily comparable results. The calculator can help the sales team to show and explain to a customer how a specific compressor model would behave from a heat recovery perspective in a certain climate. It also allows to experiment with different compressor parameters and therefore helps to find the best configuration for the conditions at the customer's site.

2 TAMTURBO OYJ

2.1 Products

Tamturbo Oyj is a company based in Ylöjärvi, Finland that develops and assembles high-speed turbo compressors which produce clean, oil-free pressurized air. Tamturbo has also incorporated heat recovery systems into their compressors to collect the otherwise wasted thermal energy which is created in air compression and from the electrical components of the unit. The company designs the components, assembles and tests them. They also operate a testing laboratory in the facility. (Tamturbo n.d.)

The value that the company offers comes from high energy efficiency of their air compressors. The efficiency is achieved by using direct-drive electric motors with active magnetic bearings to power their turbo units as shown in PICTURE 1. These motors do not require transmissions or lubrication oil as the shafts levitate due to the bearings, therefore completely removing physical contact between the moving parts. (Tamturbo n.d.)



PICTURE 1. Cross section of the motor-turbo-structure (Tamturbo n.d.).

The oil-free construction allows to produce clean pressurized air that does not have any residue of lubrication oil mixed into it. For customers operating in food and beverage, pharmaceutical or semi-conductor industries, for example, this kind of product offers great value as they need to operate in environments with high hygiene levels. The compressors also require only minimal maintenance since the moving parts are not wearing as they do not come in contact with each other during operation. (Tamturbo n.d.)

Heat recovery is the other major value that Tamturbo’s products offer. It is achieved by including liquid cooling circulation in the compressors and allowing that circulation to be attached to the customer’s process network. Therefore, the customer can use the provided heat energy to partially replace other energy sources, such as natural gas. (Tamturbo 2023, a.)

2.2 Founding

Tamturbo was founded in 2010 after two years of technological and commercial pre-studies. Kimmo Laine and Jaakko Säiläkivi decided to combine their experience from working with compressor and high-speed motor technology. Kimmo had been working in the compressor industry since the 1960s after graduating from Tampere University of Applied Sciences. Most of his career Kimmo worked as a CEO of Tamrotor Oy and later as a CEO of Gardner Denver Oy after the two companies merged. Jaakko had over 30 years of experience from high-speed motor business development. He worked at High-Speed Tech Oy between 1990 and 1999 as well as at Miscel Oy from 1999 till 2010. (Gust n.d.) Both Kimmo and Jaakko had done long careers, and they knew each other from prior years. Kimmo had been working with screw compressors at Gardner Denver but he visioned that the future would be in the oil-free compressor technology. As Jaakko had worked with sewage aeration compressors at HST Oy and with high-speed motors at Miscel Oy, he formed a highly skilled and experienced partnership with Kimmo. The two wanted to develop compressors that would combine oil-free compression and high-speed motor technology. (Tamturbo 2023, b.)

The first concept compressor unit was presented in 2013 but a lot of development was still required. The first unit was sold in 2017 to HK Scan after an introduction at the Hannover Messe in 2015. From there, the development continued, and model range expanded as well as the number of distributors in Europe, America and Asia. (Tamturbo 2023.) The timeline of Tamturbo's history is shown in TABLE 1. In 2022 Tamturbo's revenue was 5,1 million euros and it grew 110 % from the previous year. In 2023 Tamturbo employed 49 people. The company's chief executive officer from October 2021 has been Igor Nagaev. (Fonecta n.d.)

TABLE 1. Timeline of Tamturbo's history. (Tamturbo 2023)

2008–2009	Technical and commercial pre-studies
2010	Tamturbo founded by Kimmo Laine and Jaakko Säiläkivi.
2011–2012	Pre and concept design of high-speed turbo compressor.
2013–2014	R&D projects continued. Awarded in the top 10 in the Nordic Cleantech Open competition in 2014.
2015–2016	Validating products and concept performance, customer piloting. Introduction at Hannover Messe.
2017	First units sold and placed in service at a HKScan facility.
2018	Extending product range. Several distributors in Europe, America and Asia.
2019	Product range ready for the most common industrial oil-free market 100–350 kW. Several new multinational customers.
2020	Successful Nasdaq First North IPO cancelled due to Sulzer AG's investment. Multiple new industrial compressor sales professionals joined.
2021	Global frame agreement signed with AB InBev. Order intake increased from EUR 2m in 2020 to EUR 5.8m.
2022	Continued growth with orders from multiple new global customers. Turnover 5.1 MEUR.
2023	Continuing on a high-growth path with multiple new global customers. >60% follow-up orders and high customer satisfaction. Estimate 8-9 MEUR turnover. 45 people in 4 countries.

3 COMPRESSED AIR MARKET

3.1 Business models

Compressed air has multiple use cases in various industries. It is sometimes referred to as the fourth utility after electricity, water and natural gas. The use cases for compressed air include powering of tools, operating cylinders, atomising paint and separating different solids. It also functions as a medium to transfer energy in heat or kinetic forms. Especially nowadays as automated equipment becomes more and more standardized, pneumatic systems are in demand. Pressurized air offers safe, efficient and cost-effective way to control these devices. (K S Sudhakaran n.d.)

There are primarily two major players in the compressed air market: Atlas Copco AB and Ingersoll Rand. Atlas Copco is a Swedish company founded in 1873 in Stockholm (Atlas Copco, a n.d.). In 2022 they did 12,3 billion euros in revenue of which air compressor products contributed for 5,3 billion euros (Atlas Copco n.d.). Ingersoll Rand was founded in New York in 1871 with a name of Ingersoll Rock Drill company (Cooper Services n.d.). In 2019 Ingersoll Rand merged with Gardner Denver (Ingersoll Rand n.d). The total revenue of the company in 2022 was 5,5 billion euros of which industrial technologies was 4,3 billion euros. In addition to compressors this segment also includes vacuum and blower products as well as power tools and lifting equipment. (Ingersoll Rand n.d.)

The business models differ between selling turbo compressors with oil-free compression units with active magnetic bearings and traditional oil-injected versions which have mechanical contact between components. The portion of service revenue compared to just equipment sales associated with traditional compressor products is significant. For example, Atlas Copco states that in 2022 43 % of their revenue in the compressor segment came from servicing. (Atlas Copco, b n.d.) This major financial incentive is one of the reasons why traditional air compressor companies are not selling truly oil free compressors that require only minimal maintenance – they would lose close to half of the existing revenue from their air compressor customers. The share of revenues from servicing compared to equipment sales is shown in FIGURE 1.

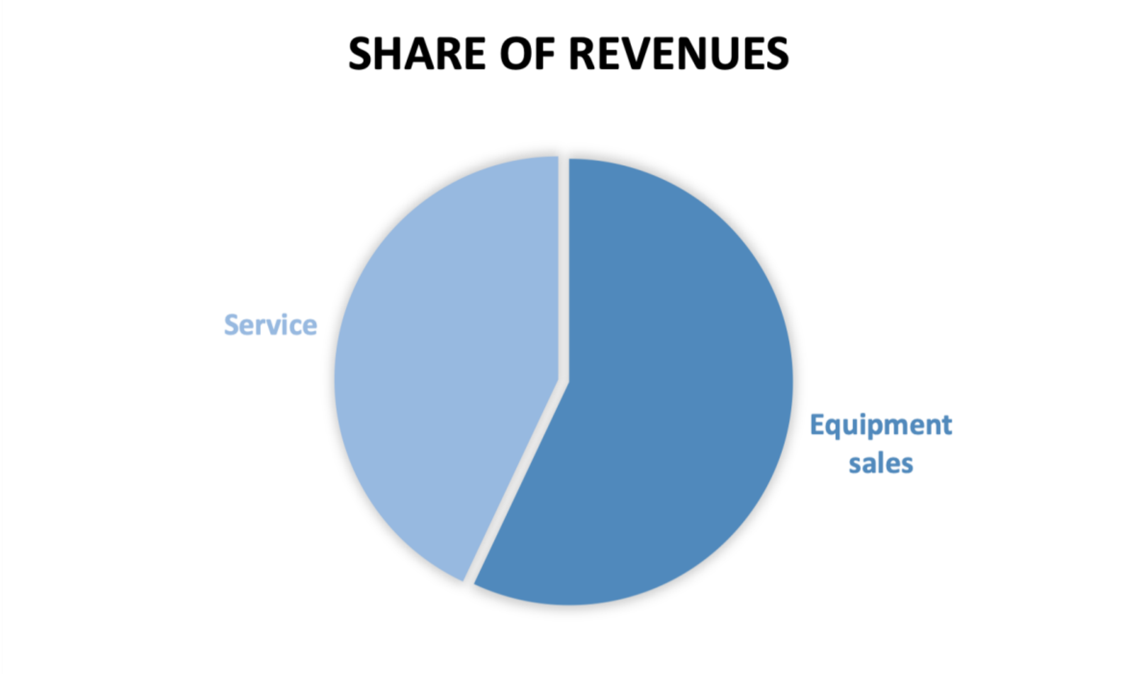


FIGURE 1. Traditional compressed air revenue shared between equipment and service segments. (Lahti 2024).

From the customer's point of view the initial investment of an oil lubricated compressor is usually lower compared to an oil-free unit with active magnetic bearings. However, due to inefficiencies in energy usage and high maintenance expenses, the upkeep costs of traditional lubricated compressors will exceed the total cost of ownership of the totally oil-free models. (Tamturbo 2023, a.)

3.2 Utilization of recovered thermal energy

Traditionally heat recovery systems have not been widely adopted into air compressors. The most popular compressor types for medium to high pressure applications are piston and screw compressors. For industries requiring lower pressure output the choice has traditionally been a turbo or a screw compressor. (U.S Department of Energy 2003.) The amount of recoverable heat energy from these units that could have been taken advantage of has been relatively low. The reasons are that most of the cooling has been with air rather than liquid and the heat exchangers used have not been able to raise the liquid temperature high enough. Therefore, the low amounts of energy and relatively cool liquid temperatures have led to the problem of having no real use cases for it. (Tamturbo 2022.) The concept of recovering heat from the compressor is also not so well understood and it is often seen as difficult to understand and implement into existing infrastructure.

Energy costs make up the largest portion of the total cost of ownership of a compressor unit during its lifetime as shown in FIGURE 2. Tamturbo's high-speed turbo compressors have dedicated heat recovery built into them which is able to collect over 90% of the supplied electrical energy to the fluid circulation. The air stream goes through heat exchangers after each compression stage and the heat is transferred to a circulating liquid flow which can be heated up to 90°C. When the liquid can reach these higher temperature levels, it opens up opportunities for various use cases where traditional energy sources, such natural gas or electricity can be partially or completely replaced. The savings in annual costs can reach hundreds of thousands of euros by combining the supplies of compressed air and heat energy into a single machine rather than paying separately for each. (Tamturbo 2022.)

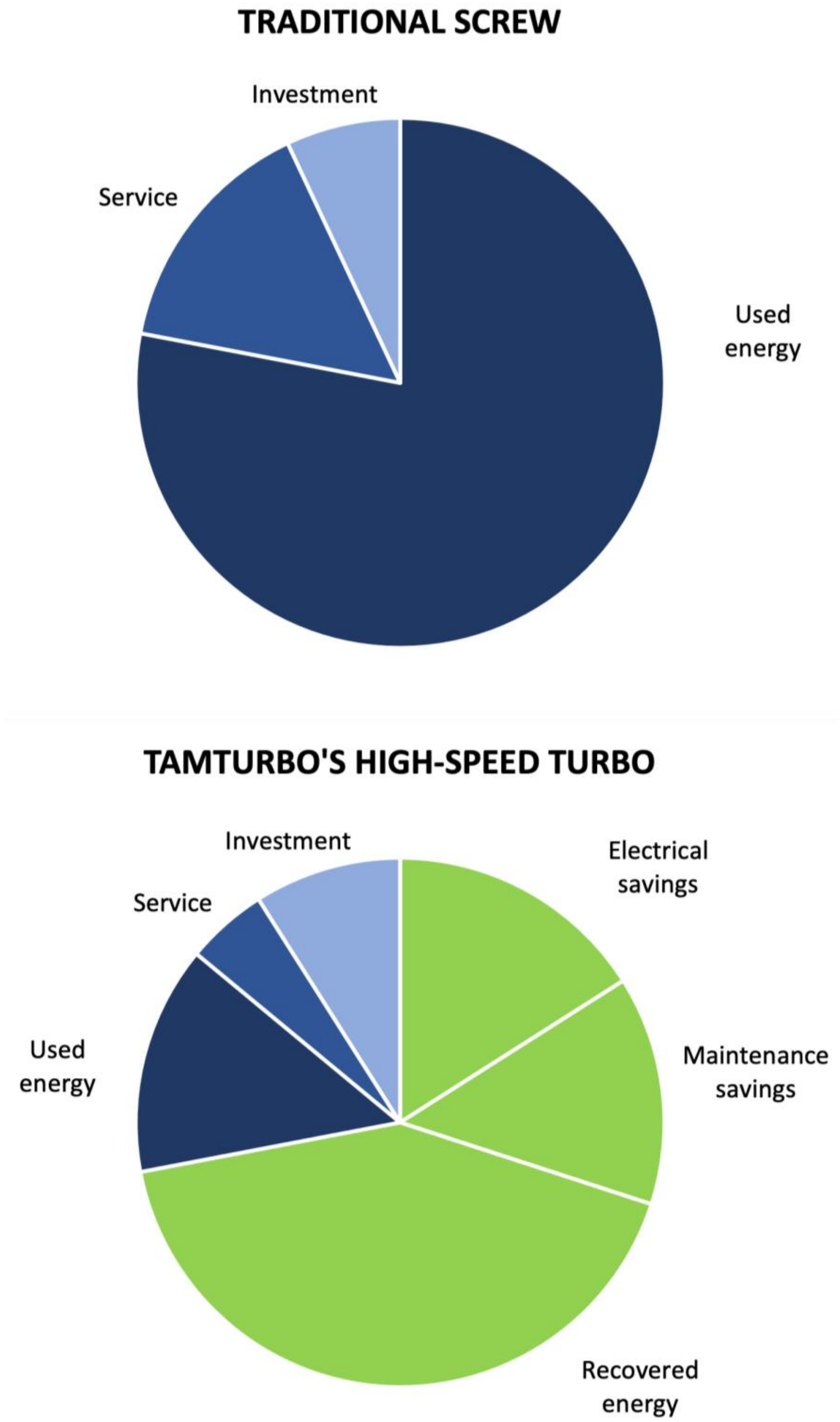


FIGURE 2. Comparison of total cost-of-ownership of a traditional screw compressor and Tamturbo’s high-speed turbo compressor. (Lahti 2024.)

3.3 Environmental impact

5% out of all the electricity consumed globally and 10% of all the electricity consumed by industrial applications is used by air compressors (VPInstruments n.d.).

As the trend is towards reducing consumption of the earth's resources, optimizations to the efficiency of air compressors will provide major opportunities. Savings in consumed energy will also contribute to savings in the carbon dioxide emissions.

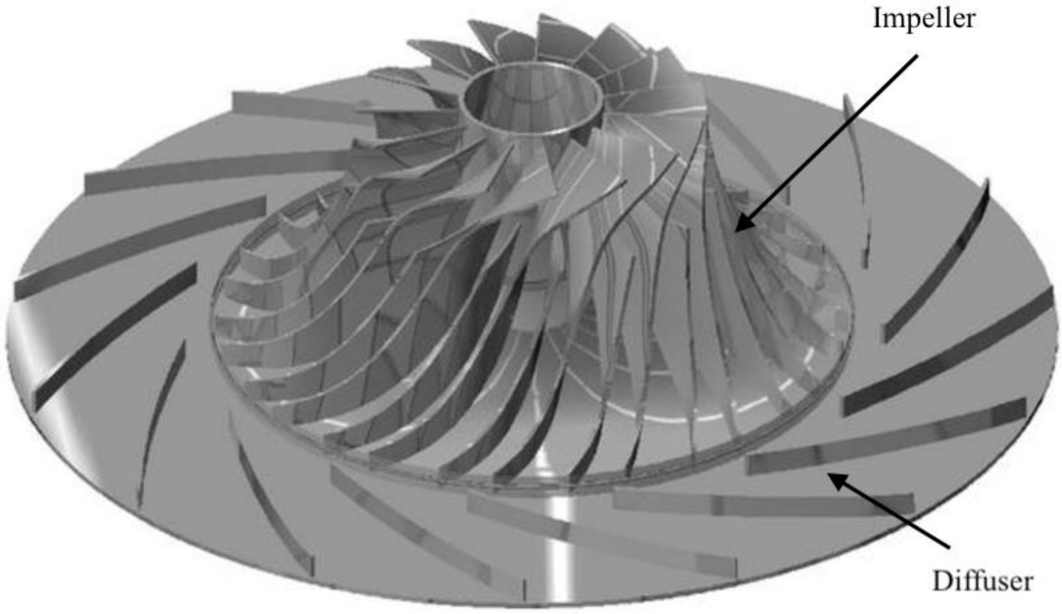
Recovering the otherwise wasted thermal energy is one of the ways to optimize compressors but improving the efficiency of the unit itself should also be taken into consideration. By reducing the amount of electricity that is put into the system in the first place the environmental impact as well as the electricity expenses will be reduced. Tamturbo's efficient high-speed turbos which operate with direct drive motors that have levitating axels via active magnetic bearings waste very little energy for friction and require only minimal maintenance work. In addition to heat recovery these optimizations add up to significant reductions in emissions and costs over the life cycle of the compressor unit. (Tamturbo 2023, c.)

4 COMPRESSOR THEORY

4.1 Gas compression

The gas entering the inlet of a centrifugal compressor is guided via an inlet nozzle to the impeller shown in PICTURE 2. The impeller is shaped to have multiple rotating vanes that provide kinetic energy to the gas. The impeller moves the gas from the centre to the outer perimeter where the gas then hits a stationary diffuser shown in PICTURE 3. The diffuser converts part of the gas's kinetic energy into potential pressure energy by slowing down its velocity. (Rainer Kurz 2015.)

PICTURE 2. An impeller and a turbo frame. (Evolution magazine from SKF 2023).



PICTURE 3. Impeller and diffusor structure of a centrifugal compressor. (Bulot & Trébinjac 2009, modified).

The diffuser blades will apply force to the flowing gas. According to the law of conservation of momentum the vanes do work on the gas in the direction of circumferential rotation. The force exerted by the blades has to be balanced by the change in circumferential velocity times the mass of the gas. FIGURE 3 shows the relationship between the velocity vectors on the impeller blade at the inlet in the middle and at the outlet at the outer diameter. The vectors w_1 and w_2 represent the relative velocities and can be calculated by subtracting the circumferential component of the gas velocity c_u from the circumferential blade velocity u. (Kurz 2015.) The Euler's law function for centrifugal compressors is shown in equation 1

$$P = \dot{m} \cdot \Delta h = \dot{m} \cdot (u_2 c_{u2} - u_1 c_{u1}), \quad (1)$$

where P is power in Watts, $\dot{m}$ is mass flow in kilograms per second, Δh is the change in enthalpy in joules per kilogram, u is the circumferential blade velocity at inlet (1) and exit (2) of the impeller in radians per second and c_u is the circumferential component of gas velocity at the inlet (1) and at the exit (2) in meters per second.

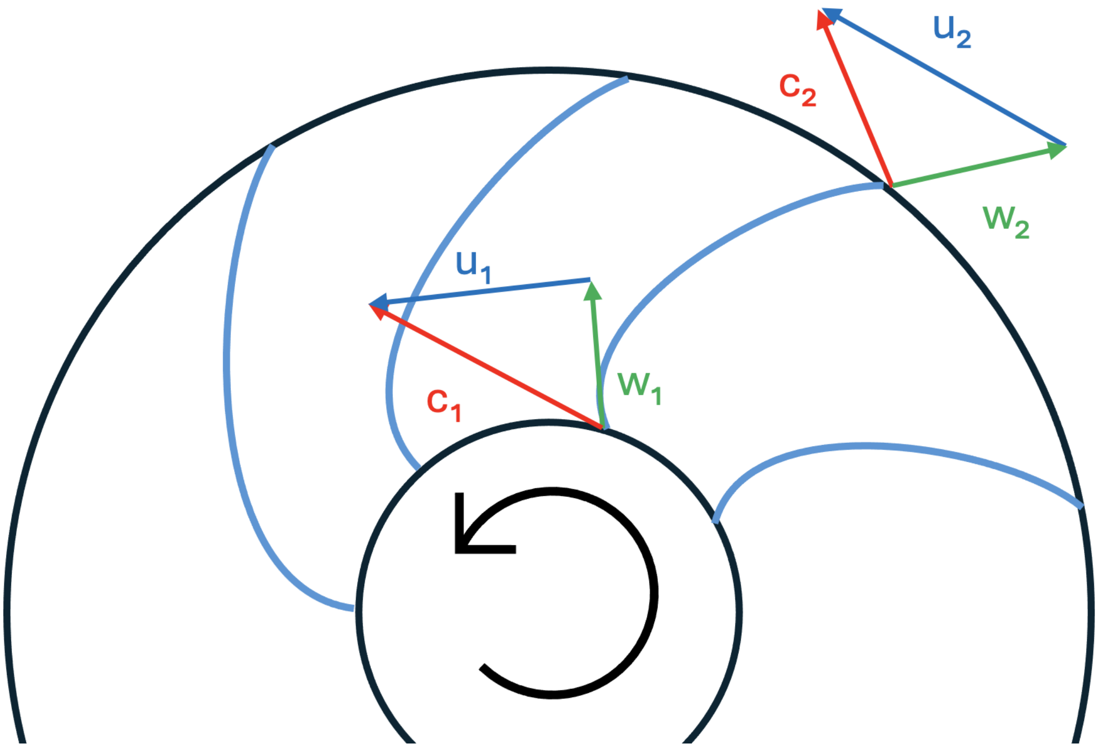


FIGURE 3. Velocity vectors of an impeller at the inlet and outlet. (Lahti 2024).

The velocity vectors shown in FIGURE 3 are affected by the impeller blade's sweep angle. Therefore, the angle affects the impeller's ability to do work on the flowing fluid. The sweep angle can be applied either to the leading edge of the blade or to the trailing edge. When both of these have a swept edge, the impeller is referred to as a combined sweep impeller. A larger forward sweep angle at the leading edge, also known as the inlet, increases the impeller's total pressure ratio and efficiency within its operating range. Forward swept angle means that the blade is curved to the direction of its rotation. The opposite is backward swept blade. When the leading edge of the blade is swept in this manner, the impeller will have decreased pressure ratio and efficiency in high mass flow conditions. In lower mass flow scenarios the efficiency will increase compared to a blade with no sweep angle. Adding sweep angle to the trailing edge of the impeller at the outlet will also have an effect on the performance as well as on the stresses exerted to the impeller. Increasing the outlet backsweep angle to the backward direction increases the total pressure ratio and efficiency of the impeller. With a forward sweep angle, the pressure ratio will be reduced, and the efficiency will also drop in high mass flow operation. (Tian, Hou, Tong, Lin, Ma 2023.)

If the compressor has multiple stages, the process will repeat at each stage. After the last compression phase the gas will enter the discharge system. It might consist of volutes to convert even more of the kinetic energy into pressure or just a collection cavity before the gas exits through a discharge nozzle. (Kurz 2015). A fully assembled Tamturbo turbo compressor is shown in PICTURE 4.

PICTURE 4. Tamturbo's TT325 air compressor (Picture: Lahti 2023).

4.2 Compressor operating efficiency

A centrifugal compressor always has an operating range in which it can perform the most efficiently. The range is different for each compressor model and even within the same model family there can be variations between the units. For this reason, centrifugal compressors should be tested prior to delivery so the running parameters can be individually set for each unit. Air flows too low and too high through the compressor will both lead to decreased efficiency (Kurz 2015).

When the volume of the air flow becomes lower than the stability limit of the compressor, the pressure on the discharge side will become too high for the compressor to overcome. Because of the higher pressure on the discharge side compared to what the impeller is producing, the gas flow will naturally start to flow from the higher pressure zone to the lower pressure zone. Therefore, the airflow will divert to the opposite direction and start to flow backwards from the discharge side to the impeller. This phenomenon is called surge, and it can be detected from strong oscillations of pressure and flow in the system. (Kurz 2015.) Surge can be mitigated by maintaining high enough pressure ratio on the compressor and by using by-pass valves to even out sudden differences in pressure between supply and discharge sides. Recycling valve along with recycling line could also be used (Kurz 2015). The typical operating map for a variable speed driven centrifugal compressor along with surge lines is shown in FIGURE 4.

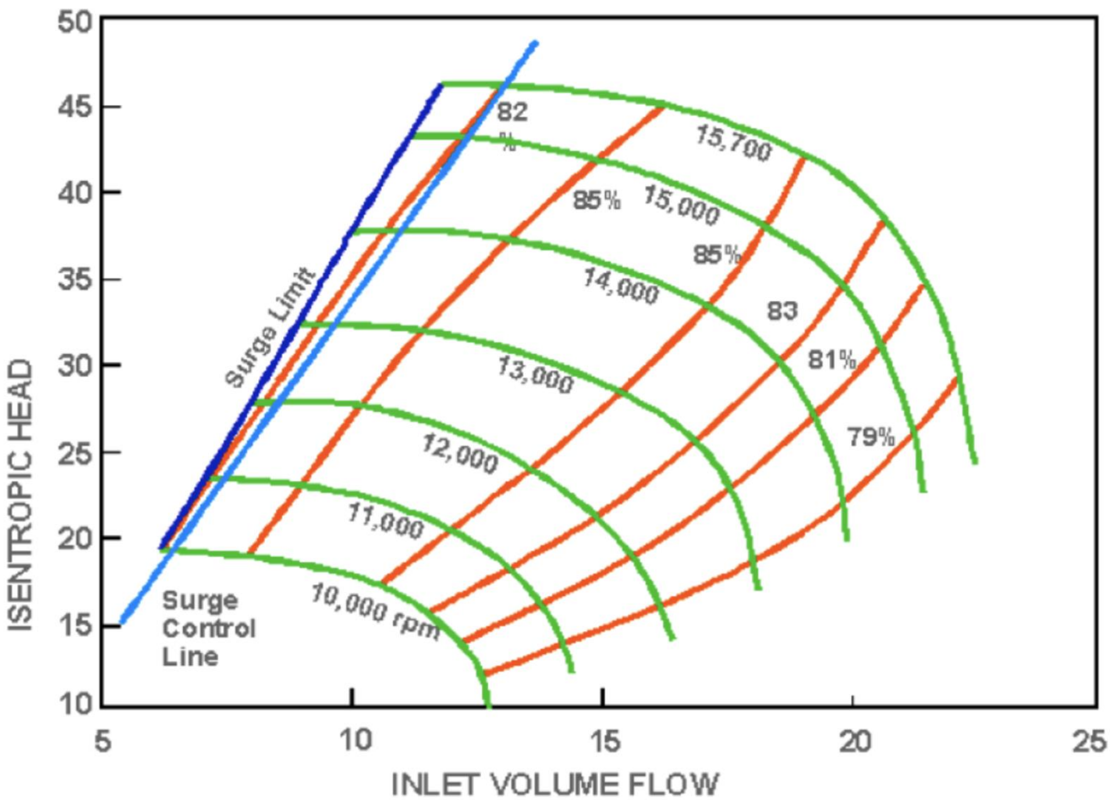


FIGURE 4. Variable speed drive compressor’s typical operating map with surge limit marked. (Rasmussen & Kurz n.d.).

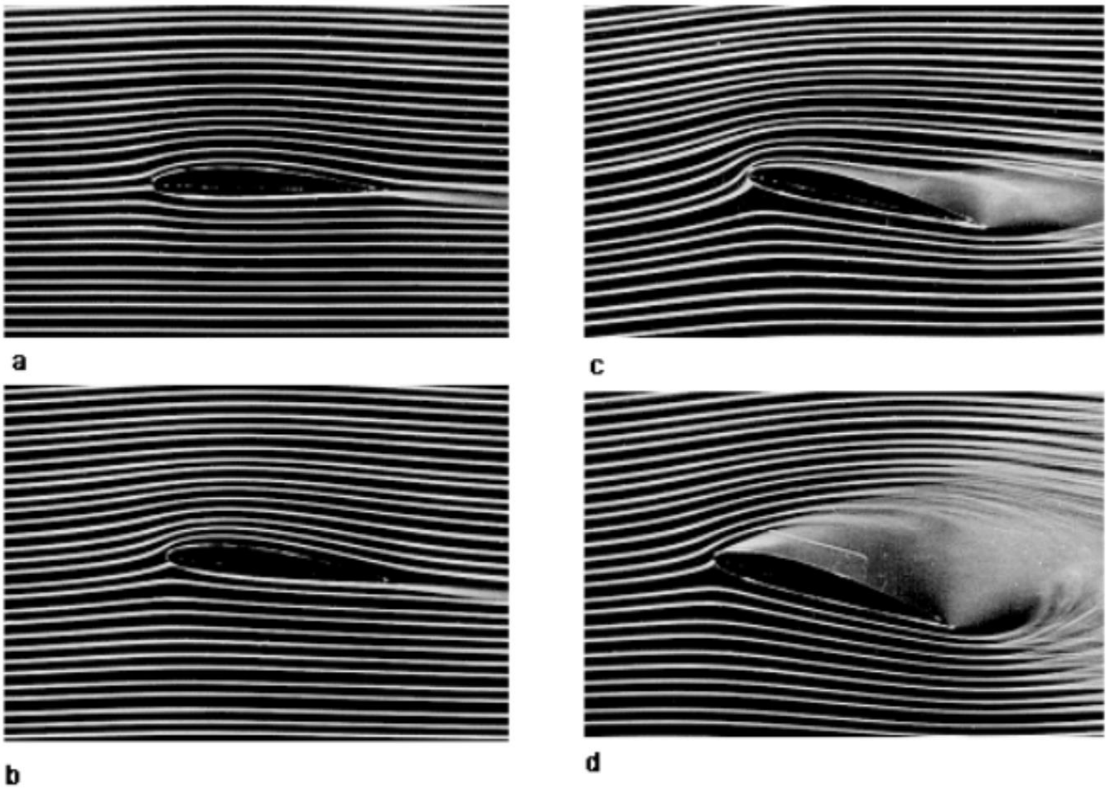
The distance from the surge limit can be determined for any operating point of a compressor. Two of the most often used ways to analyze the point are surge margin and turndown. Surge margin shown in equation 2 is based on the flow margin between the operating point and surge limit at constant speed. Turndown shown in equation 3 is based on the flow margin between the two points at constant head. (Kurz 2015.) In gas compression processes the term head refers to the amount of energy added to the air by the compressor (Compression Service Technology n.d.).

$$SM(\%) = \frac{Q_A - Q_B}{Q_A} \cdot 100 \tag{2}$$

$$Turndown(\%) = \frac{Q_A - Q_C}{Q_A} \cdot 100 \tag{3}$$

In the equations 2 and 3 factors $Q_{A,B,C}$ represent volumetric flows at set measurement points before and after the compression stage in cubic meters per second.

If the compressor runs at a constant speed but the flow through it is reduced, all the aerodynamic components will experience losses. If the air flow is further decreased, the diffusor and impeller inlet will experience stalling, where the air flow is separated from the surface of the airfoil. The phenomenon is shown in PICTURE 5. In the phases ***a*** and ***b*** the airflow is still fully attached. In phase ***c*** it is partially separated and in ***d*** it is fully separated from the airfoil. Stall usually occurs in one compression stage first. (Kurz 2015.)



PICTURE 5. Air flow separation from airfoils at different angles of attack. (a,b) unseparated, (c) partially separated and (d) fully separated. (Nakajama 1988).

The compressor’s efficiency will also drop if flow is increased too much in relation to the pressure output. After a certain point the compressor will stop producing any head at all. The phenomenon would occur on the opposite side of the graph compared to surge limit shown in FIGURE 4. It is called choke, and it is caused by ever increasing friction due to higher internal flow speeds. In addition to frictional losses, an impeller will also experience head reduction when flow is increased due to its backwards bent blades. Another term for choke is stonewall. (Kurz 2015.) FIGURE 5 shows the relationship between flow, head and frictional losses at constant speed.

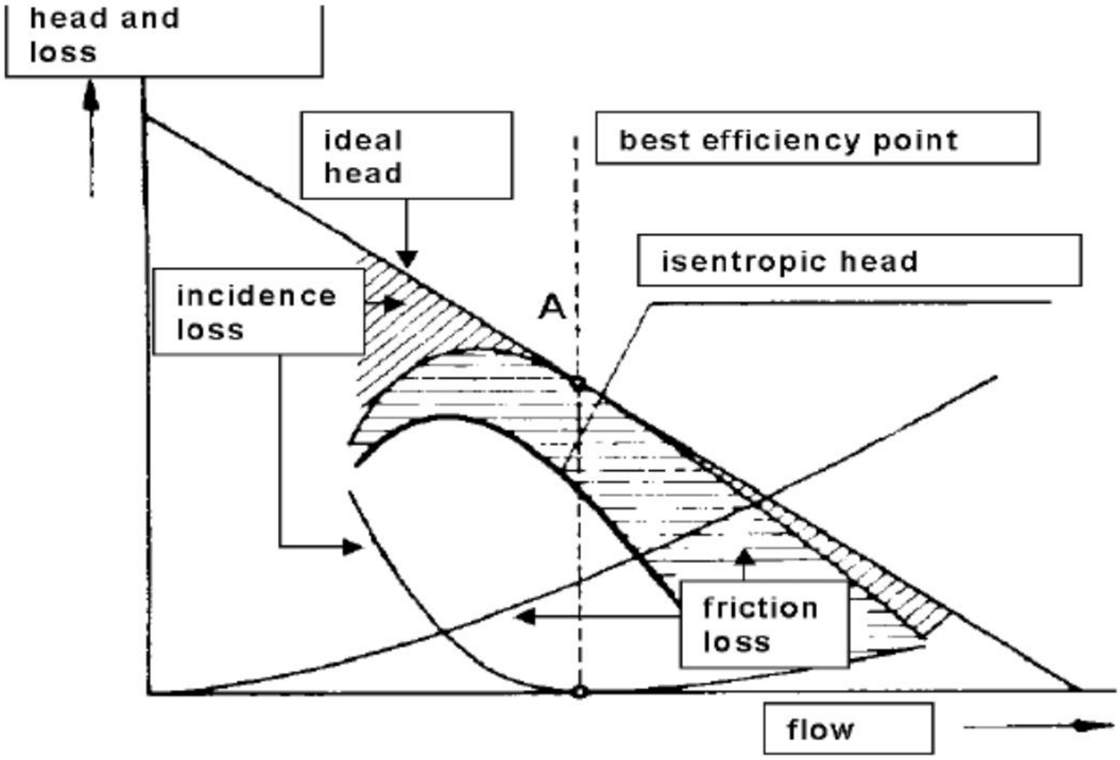


FIGURE 5. Flowrate, head and frictional losses in relation to each other at constant speed. (Kurz 2015).

4.3 Thermodynamics

The operational principles of centrifugal gas compressors, which includes turbo compressors, are based on First and Second Laws of Thermodynamics as well as laws of fluid dynamics. The first law of thermodynamics defines the conservation of energy which states that energy can neither be created nor destroyed but it can only change its form. (Kurz 2015.) Enthalpy relates to this law as it describes the change in the internal energy of a thermodynamic system. It is often used to describe whether a reaction is endothermic or exothermic. (Britannica 2023, a.) When a process occurs at a constant pressure, the heat which is either released or absorbed is equivalent to the change in enthalpy (Hurley & Shamieh n.d.). Enthalpy at a specific point for a gas compression process can be determined by multiplying the mass of the gas with its specific heat capacity and with its change in temperature. The mass can be specified, for example, from the mass flow of the gas entering or leaving the compressor. (Mäkelä, Soininen, Tuomola, Öistämö 2019, 110.)

The Second Law of Thermodynamics explains the concept of entropy which is associated with the loss of ability to do work. The basic principles are that 100 % of energy cannot be transformed to work and that entropy can be created but never destroyed. (Tom Benson 2021.) The fundamental ideas of entropy and enthalpy allow to calculate the efficiency of a compressor according to equation 5 by comparing the theoretical optimal, the work required for the isentropic compression, with the actual amount of work required for the compression.

$$\eta_s = \frac{W_i}{W_a}, \tag{5}$$

where W_i is the work done in isentropic compression in joules and W_a is the actual amount of work done in the real compression in joules (Kruz 2015.) The amount of work required for the isentropic compression is calculated according to equation 6

$$W_i = m \cdot c_p \cdot T_{air\ isentropic\ outlet} - T_{air\ inlet}, \tag{6}$$

where *m* is the mass of the gas in kilograms, c_p is the specific heat of the gas in joules per Kelvin and *T* is the temperature at the outlet of the isentropic compressor and at the inlet in Kelvin. The air temperature at the isentropic outlet point is calculated according to equation 7

$$T_{air\ isentropic\ outlet} = T_{air\ inlet} \frac{P_{air\ oulet}}{P_{air\ inlet}}^{\frac{\gamma-1}{\gamma}}, \tag{7}$$

where *P* is the pressure in Pascals at the specified points and γ is the isentropic exponent. The amount of work required for the actual compression is calculated according to equation 8

$$W_a = m \cdot c_p \cdot T_{air\ outlet} - T_{air\ inlet}. \tag{8}$$

The theory of the isentropic compression and expansion can be used to demonstrate the behaviour of temperature and pressure in an imaginary process. The process involves compressing air with a compressor after which the thermal energy is collected with a heat exchanger. The pressurized air is stored in a tank where it sits in its original room temperature in elevated pressure. From the tank the air flows to a turbine where it expands lowering its pressure back to that of the ambient surroundings. While the air expands its temperature also drops below 0°C. Therefore, it can be used for cooling by passing it through a heat exchanger where its temperature will once again rise back to room temperature without affecting the pressure. This case study is shown in FIGURE 6 and in FIGURE 7 and the calculations for the changes in temperature and pressure are done according to equation 7. The processes are assumed to have no losses to illustrate the thermodynamic events in a simplified manner. The isentropic exponent $k = 1{,}4$ in the calculations as that is the approximate value for dry air (Hall 2021). The purpose of this example is to show the potential use cases for the potential energy stored in the air but also the various processes where the temperature changes occurring in compression and expansion can be taken an advantage of. The electrical power that is initially fed into the compressor also lends itself for heating and cooling purposes in addition to the compression, allowing for cost savings in those categories. This process also demonstrates that 100% of the power consumed in the compression can be used as thermal energy. When manufacturers state that air compressors waste over 90% of power as heat from cooling and that compressors are able to capture less than 10% of the energy into the pressurized air, that is not fully accurate considering the physics shown in this process. The energy balance and efficiency of an air compressor should be considered separately. All the energy put into the compressor also exits it and if the pressurized air does not have velocity, all the energy will be as heat. (Uwe Kaiser n.d.) This theory is based on the first law of thermodynamics. When a gas is contained in an enclosed container and work is done on the gas to increase its pressure, its internal energy will increase. For example, decreasing the volume of the container by pressing down with a piston does work on the gas, which is equal to the force pressing down the piston multiplied by the distance which it travels. When the volume of the container decreases the gas molecules inside will move closer to each other and therefore gain ever increasing velocity and hit the walls of the container with higher momentum. This will apply force on the walls, which creates the force of pressure. The kinetic energy in the microscopic level, meaning in the gas molecules, translates to thermal energy on a macroscopic level. When the gas is brought to state of zero velocity, the air molecules will also have zero net motion. Therefore, all of the internal energy stored will be as thermal energy which will be reflected in the temperature of the gas. (David SantoPietro n.d.)

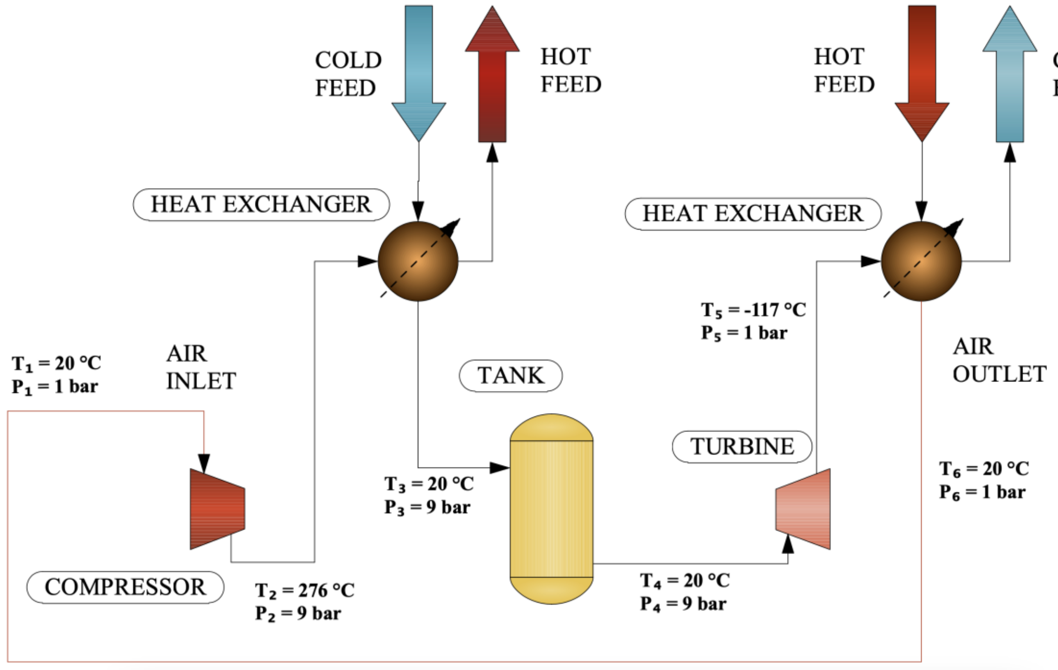


FIGURE 6. Flow diagram of a compression and expansion process. (Lahti 2024).

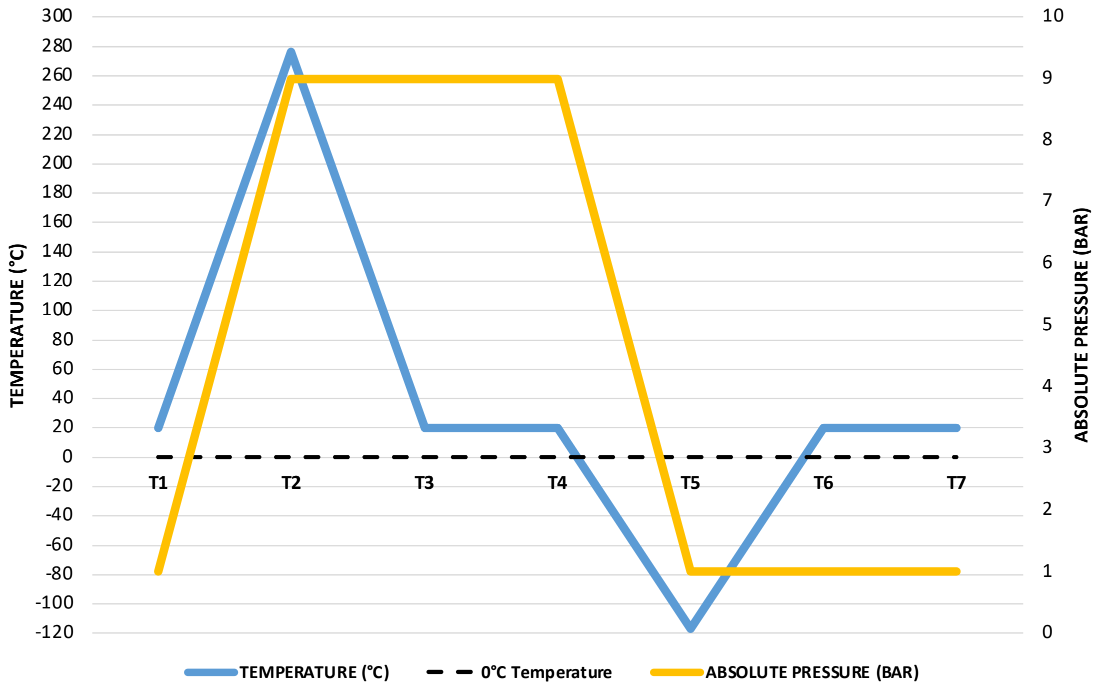


FIGURE 7. Temperature vs. absolute pressure diagram of the compression and expansion process shown in FIGURE 6. (Lahti 2024).

5 ENERGY

5.1 Heat recovery

When considering heat recovery systems in gas compressors, understanding the energy balance of the unit helps to consider all the possible sources of energy the system presents. It also allows to calculate potential savings from the recovered energy and asses the efficiency of the compressor. Energy balance in essence shows all the energy streams going into the compressor and the streams coming out of it and their relative magnitudes. The energy balance model of a high-speed turbo compressor is shown in FIGURE 8. Each energy stream has been broken down into components to show the factors affecting them.

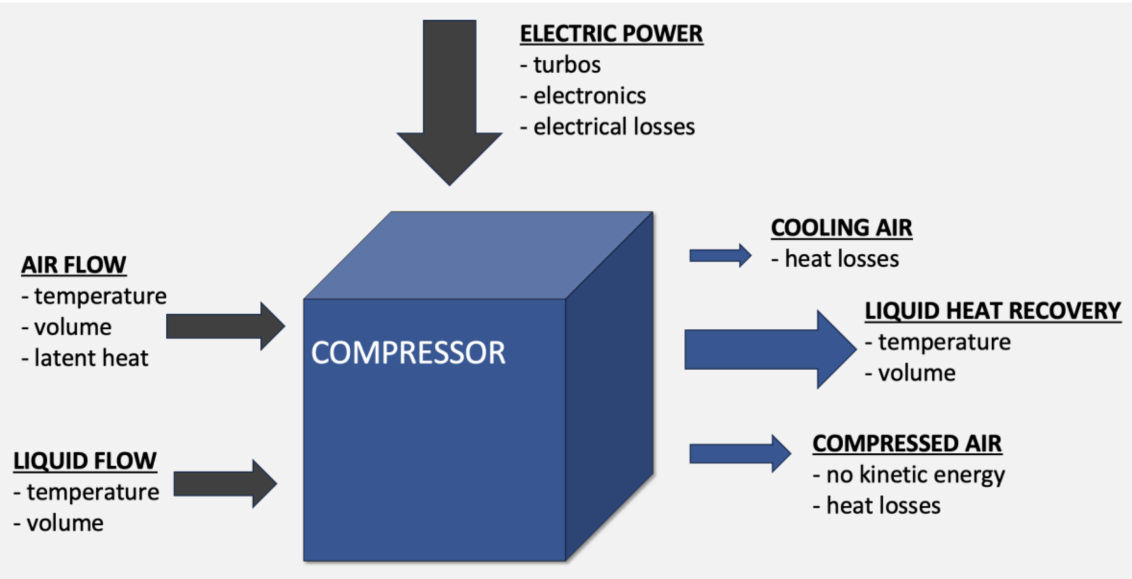


FIGURE 8. Energy balance of a turbo compressor. (Lahti 2024).

Electric power is distributed to each turbo stage as well as to all the electrics in the machine. Some of it is lost as electrical losses in the motors, frequency converters and other electronics. The lost thermal energy that radiates out from these components without added cooling is considered natural air cooling. In order to maintain suitable conditions inside the compressor housing, forced air cooling is also required. That means using additional blowers to generate air flow through the machine and therefore boost the heat dissipation.

Inlet air flow to the impellers brings in thermal energy as well as humidity which allows for collection of latent heat. Liquid flow enters the compressor with cold temperature and its purpose is to collect thermal energy from the compression process and exit as hot liquid. In cases where the incoming air has a higher temperature than incoming liquid, the temperature difference can be collected into the liquid. Compressed air exits the compressor having thermal energy as it will be warmer than the air at the inlet. The thermal energy in the air flow is considered to be lost energy as it cannot be recovered any further. For this model the pressurized air is considered to have no kinetic energy and to be in a stationary state in a storage tank, for example. This is to make the energy balance simpler, as now the only form of energy in the air is heat. Pressure itself is not energy as it describes the force exerted in a certain unit area. Changes in pressure can allow for transformation of energy. (Anne Helmenstine 2022.)

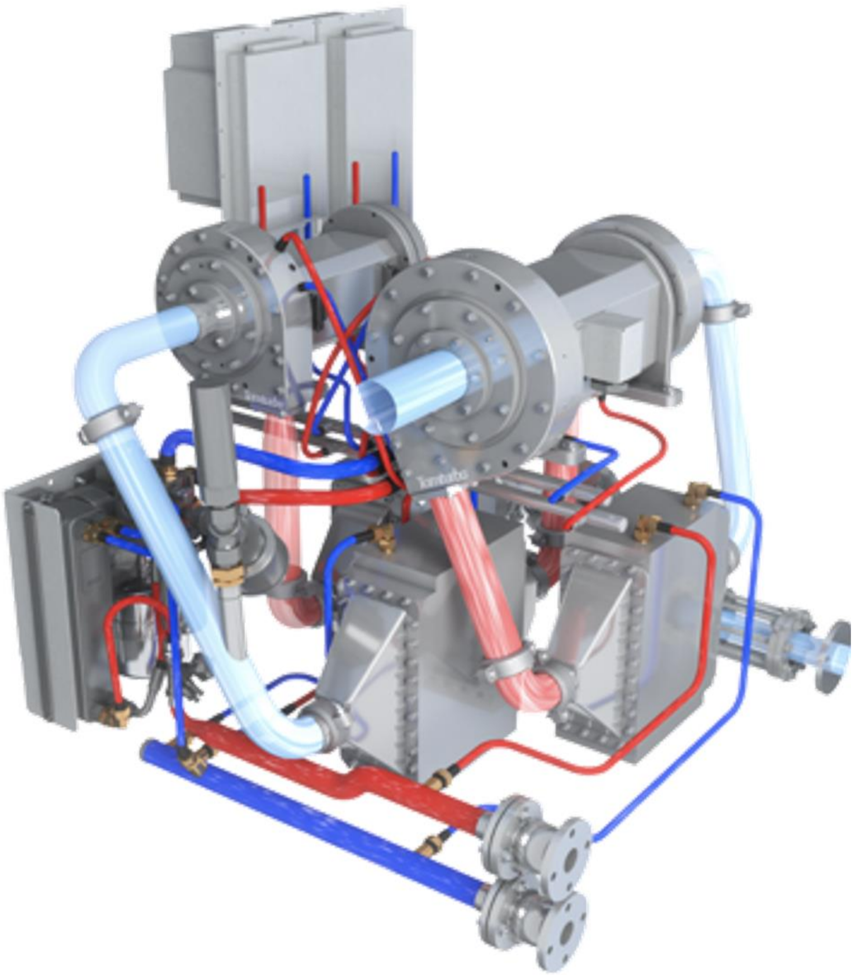
Robert Boyle’s law describes the relationship between the volume and pressure of a gas in a constant temperature environment. It tells that when the space containing air reduces, its pressure must increase as long as the temperature stays the same. Jacques Charles’ law describes the behaviour of gas when it is heated or compressed. It says that gas’ volume divided by its temperature equals a constant, which in this case is pressure. According to this law increasing temperature will also require an increase in volume to maintain the same pressure. (vmacair 2023.) The combination of Boyle’s and Charles’ laws describes the relationship between air’s pressure, volume and temperature related to each other as shown in equation 9. (vmacair 2023.)

$$\frac{P \times V}{T} = k, \tag{9}$$

where *P* equals pressure in Pascals, *V* equals volume in cubic meters, *T* equals temperature in Kelvin and *k* equals a constant.

Alongside the pressurized air that a customer gets from an air compressor, recovered thermal energy can be just as important for them. It is essentially free energy created as a by-product from the compression process. Efficient storage and use of the heat will create massive savings in heating costs for the customer compared to using natural gas, for example. (Tamturbo 2023, c.)

The majority of the heat comes from the pressurization of the gas. However other components of the compressor are also a source of thermal energy. As shown in PICTURE 6, Tamturbo's compressors also collect excess heat from the electrical cabinets, turbo heads and frequency converters that are energy intensive components used to run the compressor. (Tamturbo 2022.)



PICTURE 6. Hot and cold circulations of TT145 Boost Heat Recovery (BHR) compressor. Heat is recovered from compressed air, motors and electric cabinet. (Tamturbo 2022, modified).

The thermal energy in many cases is used in the same facility where it is created but it is also possible to transport the heat via a pipeline to a nearby building, where it can be used, for example, as a district heating source. This was demonstrated in Northern Ireland where a packaging plant called Greiner Packaging used its excess heat recovered from its air compressors to contribute to the heating system of a nearby secondary school. The school had 600 students and used about 47 000 euros annually for heating expenses. After Greiner Packaging began to distribute their excess thermal energy to the school, in addition to savings in heating costs, they were also able to reduce their carbon footprint by 200 metric tons. (Plant & Works Engineering 2020.)

5.2 Latent heat

Latent heat is the energy absorbed or released by a medium during a change in in its phase that occurs at a constant temperature and pressure. When a medium, such as water, melts into a liquid or freezes into a solid, the latent heat associated with the process is called the heat of fusion. When a solid or a liquid vaporizes or a vapour condenses, the associated term is heat of vaporization. Latent heat is displayed as the amount of heat, in joules or calories, per mole or unit mass of the medium undergoing change of phase. (Britannica 2023, b.)

Latent heat is produced from the work that is required to overcome the forces holding together atoms or molecules in a material. A solid is held together by the attraction forces between the individual atoms which constantly vibrate slightly at their positions. As temperature rises, the vibrating motion also increases eventually leading to the point where the attractive forces no longer maintain the structure together. This is the melting point of a substance. For the material to completely transition into liquid state more latent heat needs to be added. And then even more so when entering a gaseous state. When going from a solid state towards the boiling point, heat is absorbed into the material and as the direction reverses from vapor towards liquid and solid the stored thermal energy gets released into the surrounding atmosphere. (Britannica 2023, b.) Latent heat can be calculated according to equation 10. However, there are also various tables available which allow for calculations of latent heat at specified temperature or pressure (Engineering toolbox n.d.).

$$Q = mL, \tag{10}$$

where m is the mass of fluid in kilograms and L is the specific latent heat in joules per gram.

Latent heat is often overlooked when considering heat recovery in air compression. As the water vapour in the incoming air is condensed into liquid phase it releases thermal energy which can be collected and used elsewhere in another process. Collection of latent heat also allows the percentage of energy recovered from the compressor compared to the supplied electrical power to exceed 100%. (Deepak Vetal 2016.)

5.3 Heat exchange

Heat exchangers usually have a tube or a plate structure which is made out of thermally conducting material. The purpose of heat exchangers is to transfer heat from one substance to another. For any calculations regarding heat exchangers, it can be assumed that no heat energy is lost in the transfer process. (Dean A. Bartlett 1996.)

In order to be able to select a proper heat exchanger for a specific system, basic understanding of thermodynamic and transfer properties of the fluids is required. The main factors affecting heat transfer properties of a fluid are its density, specific heat as well as thermal conductivity and dynamic viscosity. Density is the fluids mass per unit of volume, and it is measured as $\frac{kg}{m^3}$. Specific heat (c or c_p for gases at constant pressure) represents how much heat is required to raise the temperature of a unit of fluid mass by one degree. The unit for specific heat is $\frac{J}{kg\,°C}$. Thermal conductivity is the fluid’s ability to conduct or transfer heat and its unit is $\frac{W}{m\,°C}$. Dynamic viscosity measures the fluid’s resistance to flow. A highly viscous fluid causes a high pressure loss in the exchanger, especially along the edges. The SI unit for dynamic viscosity is Pascal-second ($Pa \cdot s$). (Bartlett 1966.)

The flow inside a heat exchanger can either be laminar or turbulent. Turbulent flow usually causes higher pressure loss but also enables heat to transfer more effectively due to the mixing of the fluid. Whether the fluid is in laminar or turbulent flow can be determined from its Reynolds number as shown in equation 11. Heat exchangers with Reynolds number less than 2000 have fully laminar flow. If the number is greater than 6000, the flow is fully turbulent. (Barteltt 1996.)

$$Re = \frac{\rho \cdot v \cdot D}{\eta}, \tag{11}$$

where ρ is the fluid's density in kilograms per cubic meter, v is the velocity of the fluid's flow in meters per second, D is the inner diameter of the round tube in which the fluid flows in meters and η is the dynamic viscosity of the fluid in pascal seconds.

Heat exchanger approach temperature is the smallest difference between cold and hot streams. For example, if 80°C fluid is being heated to 100°C by using 105°C liquid in the hot side, the approach temperature is: 105°C – 100°C = 5°C. In counter-current heat exchangers the approach temperature can be held low which means less utility medium is required or the medium can have temperature closer to that of the process medium. In both instances operating costs can be lowered. Counter-current heat exchangers also allow the process medium to be heated to a higher exiting temperature than that of the exit temperature of the utility medium as displayed in FIGURE 9. (Alfa Laval n.d.)

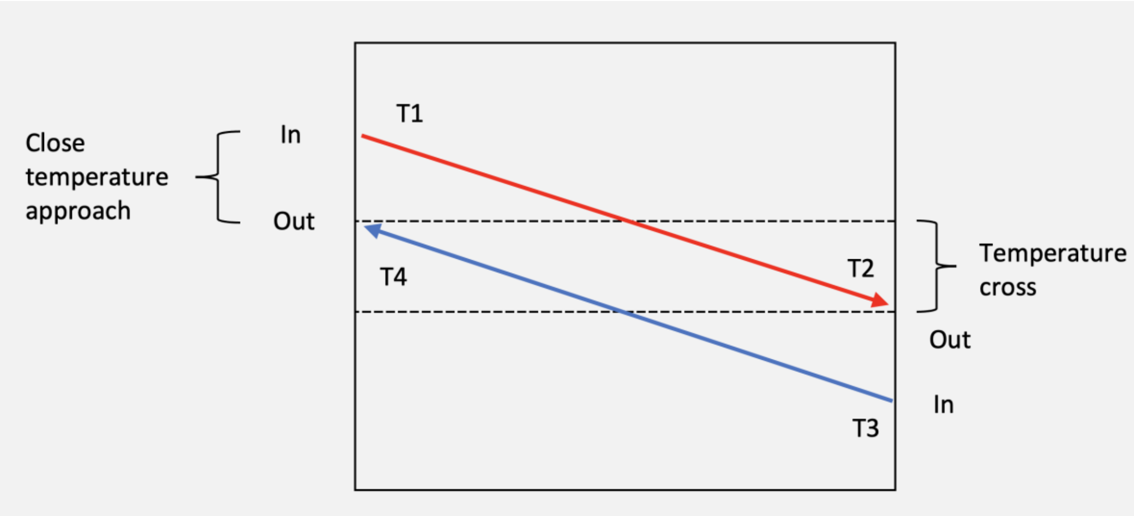


FIGURE 9. Compact heat exchanger with counter-current flow operation. (Lahti 2024).

When considering a heat exchanger for a specific system it's important to consider the required effectiveness values but increasing the size of the exchanger too much is not cost effective. The exchanger's size approaches infinity asymptotically as its effectiveness approaches 1 which means that significantly more surface area is required to raise the effectiveness from 0,8 to 0,9 than from 0,7 to 0,8. Rather than upsizing the exchanger, the effectiveness can also be improved by using a more powerful pump. Higher flow velocity will cause more turbulence to the fluid, thus improving heat transfer qualities. However, it will also lead to a higher pressure drop. (Bartlett 1996.) The temperature of the outlet liquid also greatly affects the efficiency of the heat exchangers. Therefore solely altering the flow characteristics will not allow for high efficiency. When trying to achieve high liquid temperatures, the internal structure of the heat exchangers has to be designed accordingly. The channelling has to be routed in a way that allows to have enough passes in the internal elements where the air and liquid flows meet, thus allowing the liquid temperature to reach the higher levels.

The latent heat is also collected with the heat exchangers. As the humidity in the air flow condenses to the inner surfaces, it cools down and the water will be separated from the air with condense separators. Eventually the liquid will fall to the bottom of the heat exchanger from which it will be directed out via condense valves which operate with surface level sensors. (Yao, Chen, Chen, Gong 2020.)

5.4 Boost heat recovery

Typically, air compressors have heat exchangers to cool down the air compressed by one or more compression stages. In multi-stage compressors the heat exchangers are usually positioned after each compression stage. They are needed because otherwise the temperature of the compressed air would rise too high which would significantly reduce the efficiency due to the reduction in density. The hot temperatures would also cause issues with the internal temperature of the machine. In standard heat recovery compressors the heat exchangers are often coupled in parallel, and the coolant liquid is also fed in parallel into each heat exchanger. In this configuration the temperature of the coolant entering each heat exchanger is the same. In this configuration the compressor can be optimised for cooling and therefore the conditions can be maintained suitable for efficient compression. However, in this configuration hot liquid over 65°C cannot be recovered as the configuration does not allow for it. The boost heat recovery option was developed to solve this issue. It allows for dynamic rearrangement of the flow order of the coolant according to its inlet temperature and also allows to use the compression stages as heat boosters to achieve liquid temperatures up to 90°C. If the boost in temperature rise is done at the machine with the compression stages, it will slightly reduce the compression efficiency as the air entering the next turbo head will be hotter. However, the benefit in achieving hotter liquid temperature outweighs the negative impact on compression. The reduction in efficiency could be mitigated by using intermediate coolers after the booster heat exchangers or by placing the booster outside the machine after the last compression stage. (Tamturbo 2021.) One of the possible configurations that could be used in a BHR compressor is shown in FIGURE 10. This figure is an illustration of one of the possible scenarios. In real world cases the flow order may differ and can be changed. When placing the heat exchangers in series, like in the figure, the coolant or part of it from one of the heat exchangers contributes to the input of another heat exchanger. This way the heat recovery and compression of gas can be optimised individually for each compression stage. The flow rate of the coolant liquid can also be changed in order to achieve a specific temperature in the coolant output and to keep the temperature constant regardless of the compressor's operating point or the ambient conditions.

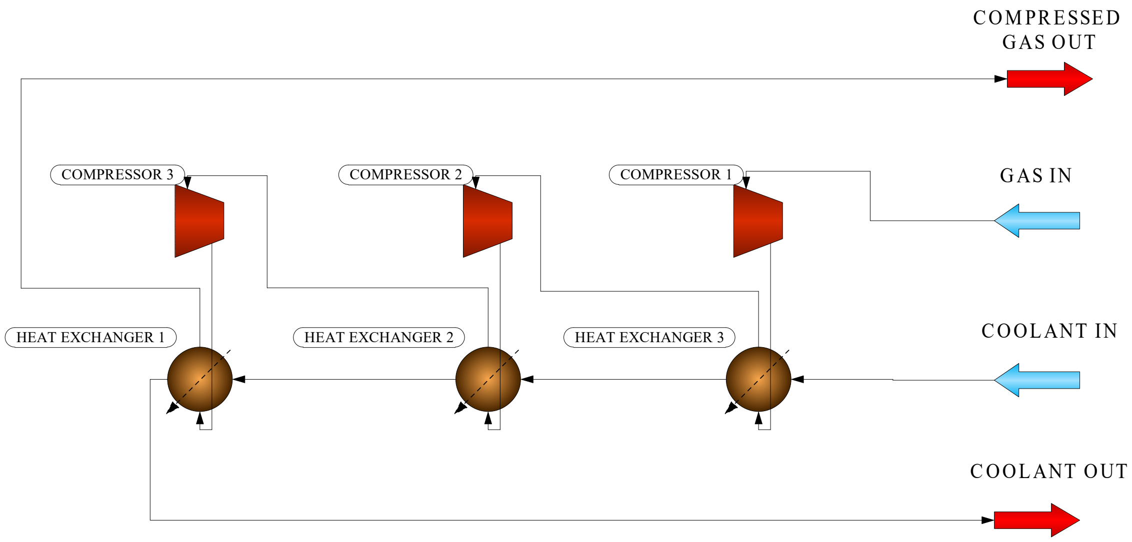


FIGURE 10. Illustration of the differing order of gas and liquid flow circulations in a boost heat recovery compressor. (Lahti 2024).

It is important to recognise that different industrial processes require different temperatures which places different values on the liquid temperature that the compressor is able to provide. If the temperature is fixed, it might have great value to a specific unit process in a certain industry but also be difficult to take advantage of in another application. Being able to choose the desired temperature and also change it regardless of the conditions or the running state add a lot of value to the boost heat recovery system.

6 CALCULATION TOOL

6.1 Theory

The purpose of the calculation tool was to make it easy and convenient to determine the effects of external factors on the thermal energy balance of a centrifugal air compressor. The energy balance is described in FIGURE 8. Apart from producing pressurized air, the idea behind Tamturbo's compressors is to collect as much heat from the inlet air, compression stages and electronics as possible. The calculation tool is based on existing testing data as there were already extensive data sets collected. In order for the calculator to be easy to operate, the data sets had to be brought together and simplified. The calculator was made simple enough to use that it would not need comprehensive knowledge of the compressor's operation or the physics behind it in order for the user to be able to do calculations. The basic parameters are taken from the compressor's data sheet and from the local weather forecast at the site where the compressor is located. The results show how much of the energy that is being put into the compressor can be collected in specific conditions and with selected compressor parameters.

The tool was made using Microsoft Excel and it was divided into two parts. The main user interface for the calculator only contains the essential variables required to do the calculations. The results are presented in numbers as well as in visual format. The other side contains all of the parameters and equations. The user does not need to interact with this document. Once the user inputs the desired variables, the calculations are done in the background and the user interface side only shows the relevant results. There is a possibility to do four simultaneous scenarios with different variables and then easily compare the results between them. The multiple scenarios allow for example to study the effects of changing a single variable on the system or to see the performance differences in winter and summer climates at the same site. The chart with all the parameters required for the calculations is shown in appendix 1. The variables are divided into their corresponding sections. *Site conditions* are present at the facility where the compressor is located. *Outlet conditions* are determined by the properties of the air flow which exits the compressor. The data sheet of the specific compressor model provides the information about the electrical power input to the compressor and also the percentage of that power recovered into the liquid flow as well as the mass flows of inlet air and liquid condensate. Parameters in the *inlet* section are calculated based on the current climate conditions at the site, such as temperature, air pressure and relative humidity. The *outlet* section contains calculated values for air temperature, humidity and volumetric flows. It also contains heat of vaporization values which is necessary for calculating the energy stored in latent heat. The heat of vaporization is derived from table values shown in TABLE 2 and they were put into a graph shown in FIGURE 11 to form a trendline. The equation of the trendline is also shown in the same figure. The trendline's R^2 value is 0,9996 meaning it follows closely the table values and is reasonably accurate to use in these calculations.

TABLE 2. Heat of vaporization values in the pressure range of 0,5–15 bar. Vapor pressure value 13 is interpolated to form a more accurate trendline.

Vapor pressure (bara)	**Heat of vaporization** $\left(\frac{kJ}{kg}\right)$
0,474	2308,000
0,702	2282,500
0,878	2266,900
1,014	2256,400
1,434	2229,600
1,987	2202,100
3,615	2144,300
6,182	2082,000
10,028	2014,200
13,000	1963,471
15,549	1939,700

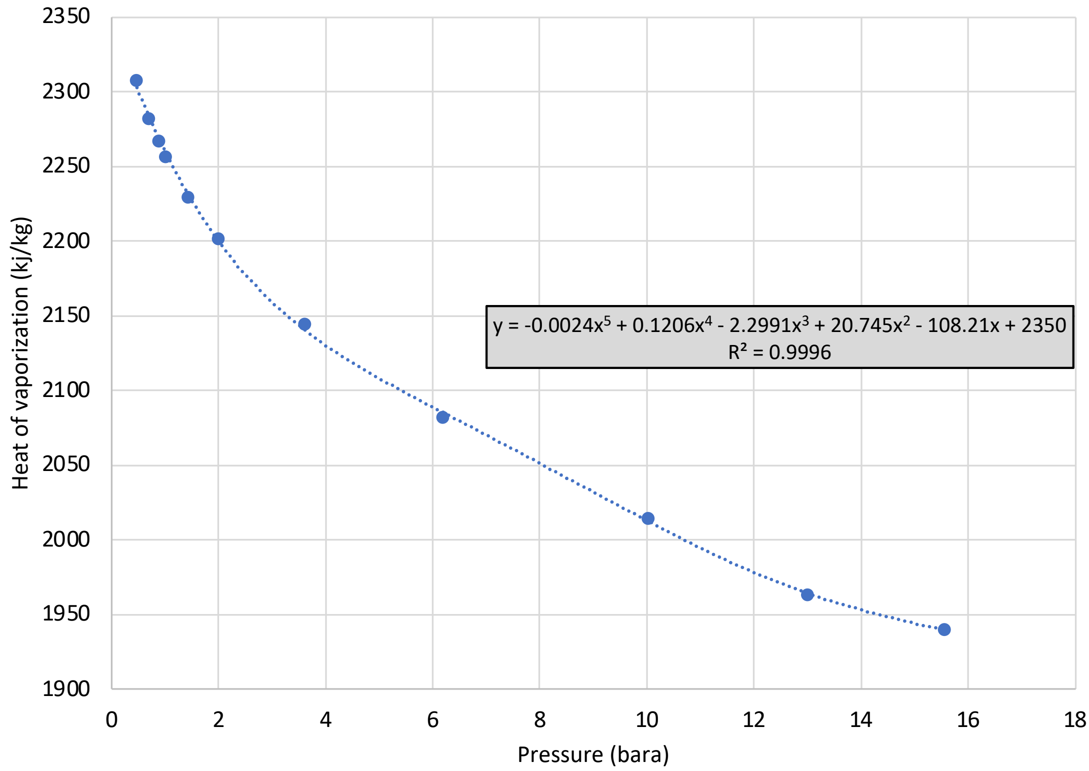


FIGURE 11. Heat of vaporization trendline and equation in the pressure range of 0,5–15 bar.

The margin of error calculations for the heat of vaporization values were calculated at four different points. The calculated values were compared to table values and the percentage of error in each case is shown in TABLE 3. The error for latent heat calculations is shown in TABLE 4. The comparable values were calculated according to a latent heat flow formula derived from equation 10. The errors were determined to be small enough for the calculation tool to be valid for its intended purpose.

TABLE 3. Error calculations for heat of vaporization values at four different pressure points.

Error review of heat of vaporization trendline equation			
Pressure	Heat of vaporization (calculated)	Heat of vaporization (from table)	**Error**
(bara)	$\left(\frac{kJ}{kg}\right)$	$\left(\frac{kJ}{kg}\right)$	(%)
10,028	2012,317	2014,200	**0,093**
6,182	2085,445	2082,000	**-0,165**
3,615	2140,395	2144,300	**0,182**
1,987	2200,638	2202,100	**0,066**

TABLE 4. Error calculations of latent heat calculations of the calculation tool at four different temperature and humidity points.

Error review of latent heat calculations at 6,1823 bara				
Temperature	Relative humidity	Calculator	Formula (latent heat flow)	**Error**
(°C)	(%)	(kW)	(kW)	(%)
39	87	48.050	49.878	**3.663**
27	90	8.800	8.965	**1.832**
32	83	18.469	18.899	**2.274**
23	100	4.238	4.308	**1.627**

6.2 Example use case

The example case for using the calculator demonstrates the conditions at two separate locations, Tampere, Finland and Houston, Texas in United States. The example simulates summer and winter climates in these locations and the changes that occur in heat recovery qualities of a TT325 Boost Heat Recovery compressor. The compressor’s running parameters are held constant to emphasise the effects from climate conditions that change depending on the season and geographic location. It is assumed that the compressor units are located at the same altitude relative to sea level and that the inlet air temperature is the same as the outside ambient air temperature. In real world scenarios there could be intermediate heaters that heat up the air coming from outside before it would enter the compressor. It is also assumed that no rain in liquid or snow form enters the compressor. After assuming all the non-essential parameters constant between the units, the only variables affecting the calculations are air temperature, air pressure and relative humidity. These are variables that the user of the tool would get from the local weather forecast when the location of the compressor is known. The customer site conditions are shown in TABLE 5. The parameters that are collected from the compressor’s data sheet are shown in TABLE 6. These are same across all the scenarios for comparison reasons. The results are shown in TABLE 7. The scenarios from 1 to 4 used in tables 5,6 and 7 are as follows:

1. Houston summer
2. Tampere summer
3. Houston winter
4. Tampere winter

Summer in this example corresponds to average conditions in July and winter respectively corresponds to January. The main variations in conditions between the locations were air temperature and relative humidity. The differences in these values demonstrate their effects on heat recovery qualities of the compressors.

TABLE 5. Parameters corresponding to the customer site conditions. The scenarios are 1 = Houston summer, 2 = Tampere summer, 3 = Houston winter and 4 = Tampere winter.

Site conditions					
SCENARIO		**1**	**2**	**3**	**4**
Ambient air temperature	°C	**39**	**23**	**18**	**-9**
Ambient air pressure	bara	**1.01**	**1.01**	**1.02**	**0.984**
Relative humidity	%	**93%**	**70%**	**52%**	**99%**
Inlet liquid temperature	°C	**10**	**10**	**10**	**10**

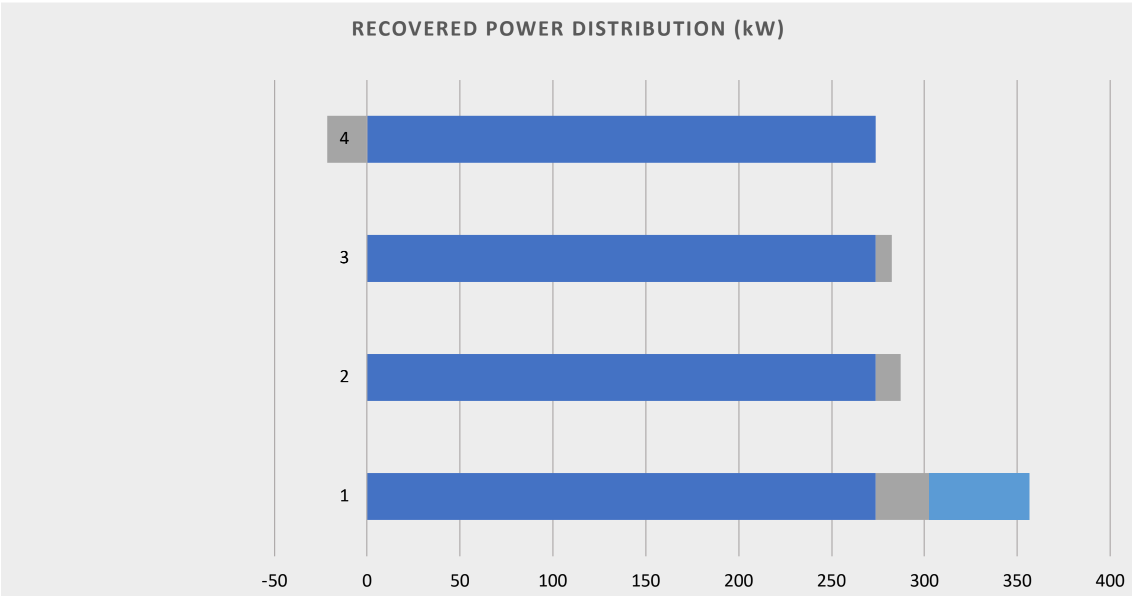
TABLE 6. Parameters collected from the compressor's data sheet. The scenarios are 1 = Houston summer, 2 = Tampere summer, 3 = Houston winter and 4 = Tampere winter.

Data sheet parameters					
SCENARIO		**1**	**2**	**3**	**4**
Electric power input	kW	**351**	**351**	**351**	**351**
Inlet air volumetric flow	m^3/min	**52.1**	**52.1**	**52.1**	**52.1**
Outlet air temperature	°C	**55**	**55**	**55**	**55**
Outlet air pressure	barg	**5.1823**	**5.1823**	**5.1823**	**5.1823**
Recovered liquid heat	%	**78%**	**78%**	**78%**	**78%**

TABLE 7. Heat recovery results based on climate conditions and compressor parameters. The scenarios are 1 = Houston summer, 2 = Tampere summer, 3 = Houston winter and 4 = Tampere winter.

Recovered power					
SCENARIO		**1**	**2**	**3**	**4**
Direct recovery to liquid	kW	**273.8**	**273.8**	**273.8**	**273.8**
Inlet air heat transfer to liquid	kW	**28.6**	**13.5**	**8.5**	**-21.5**
Latent heat transfer to liquid	kW	**54.0**	**0.0**	**0.0**	**0.0**
Total heat recovery to liquid	kW	**356.4**	**287.3**	**282.3**	**252.2**
Total vs. electricity input	%	**102%**	**82%**	**80%**	**72%**
Remaining power	kW	**0.0**	**63.7**	**68.7**	**98.8**
(Electric input power - Recovered energy)					

The results are displayed according to the energy balance model shown in FIGURE 8. The most energy is recovered via the liquid circulation which collects heat from the turbo heads and heat exchangers as well as the electronics of the compressor, such as frequency converters and motors. This result is equal among all the scenarios as the compressor parameters of input electrical power and recovered liquid power percentage are the same. The power from the air flow is affected by the inlet air temperature. It determines how much additional thermal energy can be bound into the liquid flow in addition to the temperature rise occurring in the compression of air. Naturally in warmer climates this number is higher. In the scenario 4, which corresponds to conditions in Tampere at winter, the result for recovered power from air flow is negative. This happens because the temperature of the air flow is lower at -9°C than the temperature of inlet liquid at +10°C. The delta between the temperatures is considered lost energy. However, the results do not show that as cold air has higher density than hot air, it will be more efficient in the compression process. Latent heat power is recovered as the humidity in the inlet air condenses into liquid and releases thermal energy in the process. Hotter, more humid air will produce more latent heat. The results show that in Houston at summer times it is possible to reach power recovery percentages over 100 % when latent heat is factored in. The total percentage is compared to the amount of electrical power consumed by the compressor. The remaining power shows the total amount of heat energy recovered versus the electric power consumed. As is shown in the energy balance model in FIGURE 8, in reality there are also minor heat losses from the cooling air and thermal radiation off of the compressor as well as the heat carried with the compressed air. The results for the distribution of recovered power are shown in visual form in FIGURE 12. The percentages of the recovered power in relation to the electrical input for each scenario are shown in FIGURE 13. The amount of total recovered power in comparison to remaining power is shown in FIGURE 14.



kilowatts	1	2	3	4
Direct recovery to liquid	273.8	273.8	273.8	273.8
Inlet air heat transfer to liquid	28.6	13.5	8.5	-21.5
Latent heat transfer to liquid	54.0	0.0	0.0	0.0

FIGURE 12. Distribution of recovered heat divided into liquid and air flows as well as latent heat. The scenarios on the vertical axis are 1 = Houston summer, 2 = Tampere summer, 3 = Houston winter and 4 = Tampere winter.

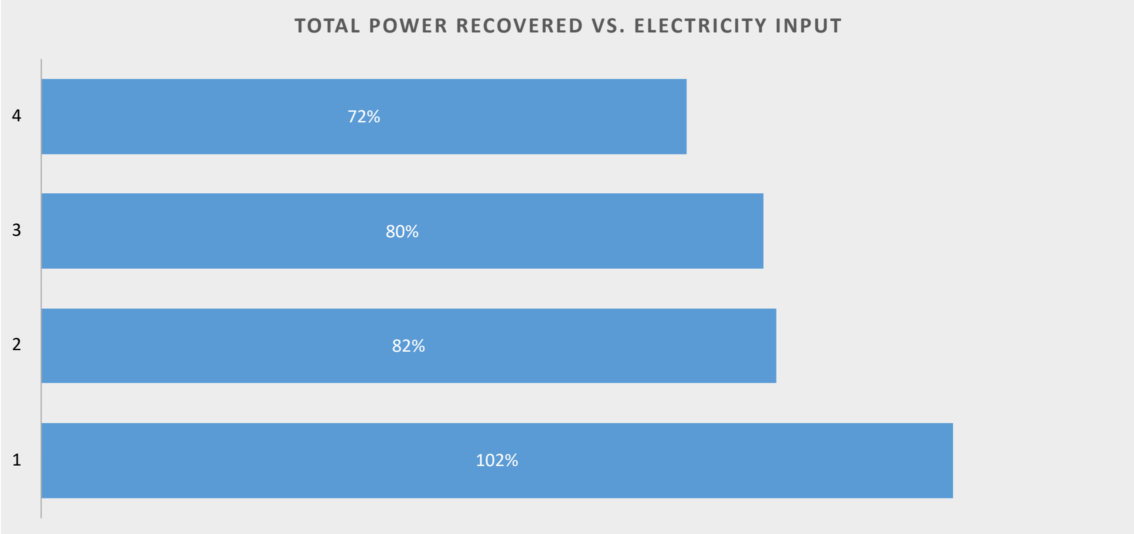
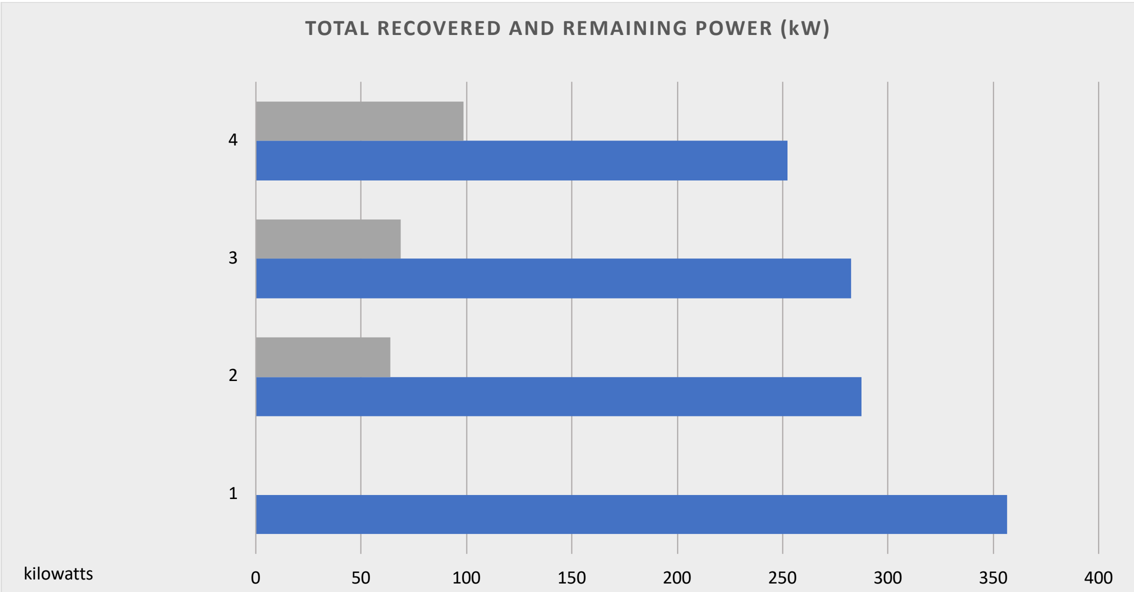


FIGURE 13. The percentage of the total recovered heat combined from all the streams in comparison to the electric power consumed by the compressor. The scenarios on the vertical axis are 1 = Houston summer, 2 = Tampere summer, 3 = Houston winter and 4 = Tampere winter.



	1	2	3	4
Remaining power	0.0	63.7	68.7	98.8
Total heat recovery to liquid	356.4	287.3	282.3	252.2

FIGURE 14. Total heat recovery to liquid in comparison to the theoretical remaining power which describes the difference of recovered power compared to supplied electric power. The scenarios on the vertical axis are 1 = Houston summer, 2 = Tampere summer, 3 = Houston winter and 4 = Tampere winter.

7 CONCLUSIONS

The results of the example use case show that in climates where air temperature and relative humidity are high, the amount of heat that can be recovered from the compressor is at its highest. The variations in ambient air pressure are typically low so it has only marginal effects on these calculations. Inlet liquid temperature was held constant between the scenarios in order to emphasise the sole effects of climate conditions on the results. Cooler temperature of the inlet liquid would have further increased the recovery percentage as the delta between it and inlet air temperature would have been higher. This would allow more thermal energy from the air flow to be bound into the liquid. The outlet pressure of compressed air also has an effect on the heat recovery as the more the air is compressed, the more its temperature will rise. In boost heat recovery compressors, the changes in the compressor’s pressure ratio are compensated by adjusting the volumetric flow rate of the liquid circulation as well as the order of rotation in terms of heat exchangers. Therefore, the pressure output does not have a major impact on the outlet liquid temperature on those compressor models.

The calculation tool presented in this thesis considers the compressor as a single unit rather than splitting it into multiple stages. The calculator could be expanded to examine the results after each heat exchanger to get more detailed data points. However, for the purpose of making the calculator easy and quick to use, it was decided to combine all the stages into one. It would also be possible to include cost savings calculator with the tool and compared the savings in energy costs by using the recovered power form the compressor in comparison to prices of natural gas or electricity, for example. The information in this thesis also lends itself to be used for educational purposes, for example as training material for new employees.

This calculator only focuses on the energy recovery aspect of compression operation. The whole other side is the compression process itself and factors affecting that. It is important to note that some factors, such as hot inlet air with high relative humidity, provide great results in the recovery of thermal energy but at the same time those are bad qualities in air compression. Inlet air with high temperature and humidity allow for a lot of heat to be transferred to the liquid flow and provide a lot of latent heat. However, both of these attributes lower the density of the air, making it less efficient in the compression process. Hot air also stresses the electrical components inside the compressor.

It is difficult to reference the calculation results to any existing study as they are highly dependent on multiple factors, including the location and climate at the site. As a reference, Atlas Copco states that their heat recovery units can recover anywhere from 80 % to 105 % of the consumed electrical power depending on the customer’s site conditions (Atlas Copco 2017). Ingersoll Rand states that its customer in a food and beverage plant in Netherlands was able to recover 70 % of the 320 kilowatts consumed. Ingersoll Rand also says it is possible to recover over 80 % of the consumed energy without affecting the performance of the compressor. (3BLmedia 2019.) These numbers, while subject to many variables, are in range of what is assumed in terms of the results from the calculator.

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APPENDICES

Appendix 1. All parameters of the calculation tool

Site conditions	Air Temp	°C	39
	Air Pressure	bara	1.01
	Humidity relative	%	87%
	Liquid inlet T	°C	10
Outlet conditions	Air temperature at outlet	°C	55
	Air pressure at outlet	bara	6.1823
From data sheet	Power input	kW	351
	Inlet flow rate	m³/min	52.1
	Air inlet mass flow	kg/s	0.979
	Recovered liquid heat	%	78%
Inlet	Air density (Dry air)	kg/m³	1.1272
	T (Kelvins)	K	312.15
	Water vapor saturation pressure [p_{ws}]	bara	0.070
	Water vapor partial pressure [p_w]	bara	0.061
	Hum absolut inlet	g/kg Dry Air	39.743
	Water vapor	kg/s	0.039
Outlet	Air outlet mass flow	kg/s	0.9788
	Air temperature at outlet	K	328.150
	Water vapor saturation pressure [p_{ws}]	bara	0.157
	Water vapor partial pressure [p_w]	bara	0.157
	Relative humidity	%	100%
	Humidity absolut calculated	g/kg Dry Air	16.202
	Humidity absolut outlet	g/kg Dry Air	16.202
	Water liquid	kg/s	0.023
	Heat of vaporization	kJ/kg	2085.445
	Density of dry air	kg/m³	6.563
	Density of moist air	kg/m³	6.500
	Volume of Air Dry at outlet pressure	m³/min	8.948
	Volume of Air Moist at outlet pressure	m³/min	9.393
	Volume of Water Vapor	m³/min	0.446
Energy recovery	Air specific heat (isobaric)	kJ/(kg K)	1.006
	Inlet air vs. inlet liquid ΔT	°C	29
	Inlet air recovery power	kW	28.554
	Vapor to liquid power	kW	48.050
	Liquid recovery power	kW	273.78
Total	Recovered power total (liquid+inlet air+latent)	kW	350.385
	Recovered power total vs input electrical power	%	100%