LAPPEENRANTA UNIVERSITY OF TECHNOLOGY
Faculty of Technology
Industrial Electronics
Master's thesis

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Power efficiency estimation and simulation of different control methods for the rotary screw compressor

Examiners:	Professor	Jero Ahola
	Associate Professor	Antti Kosonen

ABSTRACT

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Nowadays the energy efficiency has become one of the most concerned topics. Compressors are the equipment, which is very common in industry. Moreover, they tend to operate during long cycles and therefore even small decrease in power consumption can significantly reduce electricity costs during the year. And therefore it is important to investigate ways of increasing the energy efficiency of the compressors.

In the thesis rotary screw compressor alongside with different control approaches is described. Simulation models for various control types of rotary screw compressor are developed. Analysis of laboratory equipment is conducted and results are compared with simulation. Suggestions of the real laboratory equipment improvement are given.

Keywords: screw compressor, drive, control method, energy efficiency, simulation

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Abbreviations

AC – alternating current

BJT - Bipolar Junction Transistor

DC – direct current

DTC – direct current control

IGBT - Insulated Gate Bipolar Transistor

kW – kilowatt

MOS - Metal-Oxide Semiconductor

MOSFET - Metal-Oxide Semiconductor Field-Effect Transistor

PWM – pulse-width modulation

RPM – revelations per minute

V/f – voltage divided by frequency

Symbols

c – gas velocity

$C_{\%}$ - capacity in percents

c_{p}- heat of the gas during constant pressure

c_{v}- heat of the gas during constant volume

g-gravitational acceleration

h – gas enthalpy

h_1 - initial gas enthalpy

h_{coolant} – specific enthalpy increase of coolant

h_{gas} - specific enthalpy rise of gas

i_{r} - rotor current

i_{s} - stator current

i_{sq} - stator current in q-axis

i_{Sx} - stator current in x-channel

i_{Sy} - stator current in y-channel

$i_{s\alpha}$ – stator current in real axis

$i_{s\beta}$ - stator current in imagination axis

k – specific ratio

k_R - specific coefficient connected to the magnetizing and rotor inductances

J – moment of inertia

L_m - magnetizing inductance

L_r- rotor coil inductance

L_s - stator coil inductance

$L_s^{'}$ - stator specific inductance

M - gas molar weight

m_{air} - mass of air

$m_{coolant}$ - mass flow rate of coolant

m_{gas} - mass flow rate of the gas

N – number of compression stages

n_v-volume exponent for polytropic process

p – number of pole pairs

$P_{\%}$ - power in percents

p_{air} - pressure of air

p_{gas} - gas pressure

p_{req} – referenced pressure

P_{in} – input power

P_n – nominal power

q – heat flow into the compressor from outside environment

Q – the total volume flow

Q_r - real volume rate

Q_l - volume rate formed by leakages though slippages

q_v - amount of gas (volume), transferred through one revolution

R_s – stator resistance

R_r –rotor resistance

T – torque

T_1 - initial gas temperature

T_a-absolute temperature

T_{air} - temperature of air

T_e - electromagnetic torque

T_L - load torque

T_n – nominal torque

T_{on} - time when reference pressure is reached

T_R - rotor time constant

T_s' - stator time constant

u_r - rotor voltage

u_s - stator voltage

v – volume

V_{air} - volume of air

W – actual compressor work

W_0 – theoretical power input

w_s - isentropic work

y – compressor work input for specific mass

y_i - isentropic head

y_p - polytropic head

y_t - isothermal head

z – elevation

Z_1 – compressibility

η_{is}-isentropic efficiency

η_{v} - volumetric efficiency

ρ -gas density

ρ_{air} - air density

Φ – flux

ψ_{r} - rotor flux linkage

ψ_{rd}- rotor flux linkage in d -axis

ψ_{Rx} - rotor flux linkage in x-channel

$\psi_{r\alpha}$ - rotor flux linkage in real axis

$\psi_{r\beta}$ - rotor flux linkage in imagination axis

ψ_{s} - stator flux linkage

ω - angular velocity

ω_{k} - rotational speed of coordinate system

ω_{m} - rotational speed of the shaft

Introduction

Nowadays different types of gas compressors are widely used in the different types of applications. A screw compressor, which is going to be investigated in the thesis, is the most common compressor used in the industry due to a lot of reasons. It has extremely simple construction that consists of only one level to produce pressured gas. The screw compressor has fewer mowing parts than conventional reciprocating compressors. In modern screw compressor models there are no valves. Also new technologies provide high efficiency for bearings and in some cases there is possibility to develop compressor even with no valves at all [5]. Such simplicity provides many advantages. Firstly, screw compressors are extremely durable and reliable and it leads to decreased maintenance time and cost. Also, wide usage means it is rather easy to find any replacement parts for the compressor. Secondly, power-to-size ratio is rather high. The screw compressor does not require a lot of space for installation; it does not produce high noise and has significantly reduced vibration. Thirdly, decreased amount of steps leads to increased efficiency which is very important since nowadays energy efficiency is a highly concerned issue in Europe. Especially, it is significant if we take into account the fact that screw compressors designed to work with long duty ratios. And the last but the least, screw compressors designed to provide pulsation free pressure. It leads to rather stable output result of applications (for example, output torque).

Like in hydraulic systems, the most common way to control output pressure in air systems was based on valve control. Valve control in general provides enough accuracy. In the most cases, valve control is also accompanied with the direct online startup of an induction motor, which leads to decreased life span of the squirrel cage induction motor, since startup currents of motor 7 or even 9 times higher than the nominal current values that leads to overheating of stator coils. In addition, during startup torque of the induction motor has rather undesired distribution, which can lead to poor output result of application. In the end, valve control is always lossy type of control. Even though not a long time ago there were some significant developments such as digital valve system still the valve control efficiency is under question. In the case of valve control direct output pressure from the compressor itself is constant. At the same time application pressure requirement may be different. Energy consumption will be constant all the time even though it may be regulated according to process. Even more, valve servo drive will also consume some energy during operation. It is not high in comparison with the whole system, but nevertheless compressor is aimed to work in long term conditions total energy consumption may be significant during life time.

Idea of the valve control itself is rather understandable, it is easy for implementation for all kind of industrial processes and basically it does not depend on the motor type or application itself. In most cases compressor application does not have high dynamic requirements. Nowadays though there is trend to replace lossy valve control with other control methods with higher efficiency since nowadays high energy efficiency is the most valuable factor for any system.

This thesis is aimed to define possibilities of improving energy efficiency of real air pressure systems. The examined air pressure systems in general consist of an inverter, an induction motor, a screw compressor and a application (a gas tank in case the thesis). In such system produced pressure required for application is defined by rotational speed of the motor and torque.

The inverter (if it exists in the system) transfers electrical energy to the induction motor, then induction motor transforms electrical energy into mechanical and transfers it via the motor shaft to the screw compressor. After that, the screw compressor produces air flow with the desired pressure and transfers it through the pipes to application. It is very important to transfer as much energy as possible from the input (inverter or grid) to the output (application).

Speed control of pressure systems is commonly applied in various areas. Good examples of applications are natural gas transportation, district heating and refrigeration. The output pressure is the most important value in such systems. Some applications can require exact accuracy and time and poor output pressure distribution can lead to undesired consequences. The compressor is connected to the motor shaft directly or indirectly; hence pressure in such systems is proportional to rotational speed of the motor.

To provide fast and accurate control of an induction motor with frequency control is applied. Induction motor is the most common motor type in the industry. It is cheap, does not require a lot of maintenance and also control methods are well-developed for this motor type. In most cases, induction motor provides fast and stable dynamics, enough accuracy. The most important feature of the induction motor is rather high reliability since construction of the induction motor is much simpler in comparison with other motor types.

In general, simplified air pressure system consists of a converter, an induction motor, a screw compressor, and a gas tank. The scheme of the air pressure system is illustrated in Fig. 1.1.

The converter is connected to 3-phase grid to produce the desired electrical power for the motor. The motor converts electrical power of the converter to mechanical power. Then screw compressor receives mechanical power through the shaft of the motor (it is assumed that there is no reducer in the system). The screw compressor produces desired pressure to the application. In this case application is the gas tank.

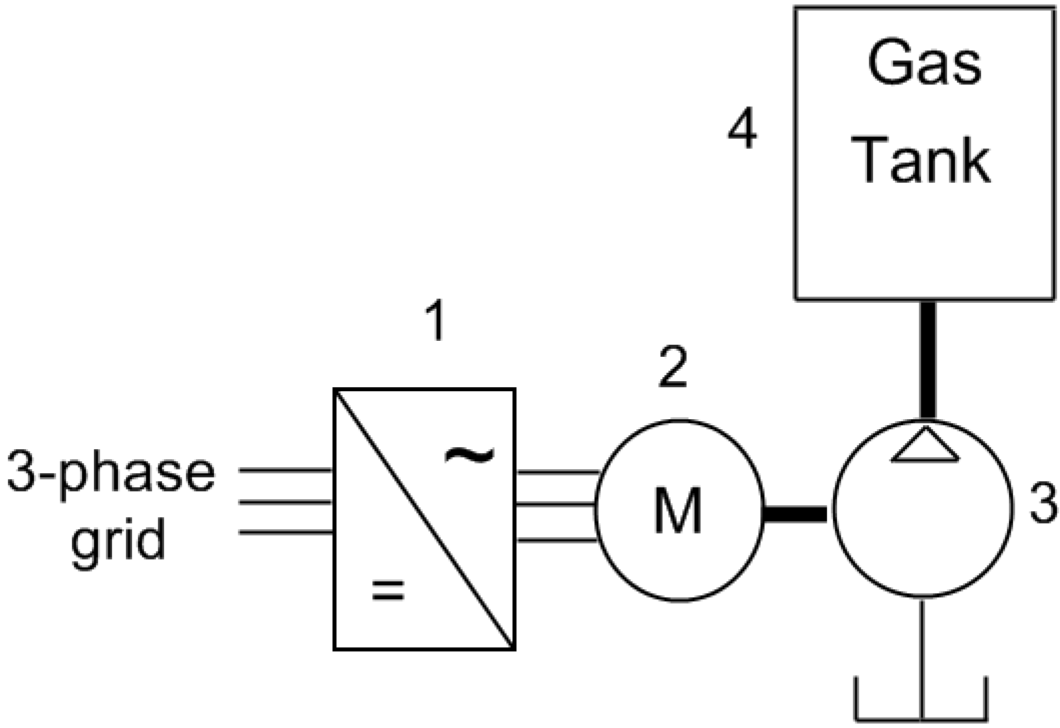

Fig. 1.1 Scheme of the air pressure system: 1) converter; 2) electric engine; 3) screw compressor; 4) gas tank or other application

The compressors are uniform in their structure and control types. However, it is still recommended to analyze real equipment because of non-linearity, which is hard to evaluate using only theoretical approach. Real model simulation can significantly ease the task in defining the ways to adjust control parameters or to find other possibilities to improve efficiency.

According to the simulation, it is also possible to define the main factors, which are affecting the energy efficiency most. So, it is important to build a precise model of the real system. That is why this thesis provides a comparison of real system and simulation.

The thesis follows four main targets. The first target is a comprehensive description of the possible equipment used for rotary screw compressor operations. The second target is a definition of main approaches used for controlling of screw compressors. This is a key target for building a proper simulation model of the compression process. Consequently, the third target is a simulation of the compression processes. During the simulation, different control methods are used, their efficiency estimated and analysis provided. The last target is to verify the simulation with the real laboratory tests. They were done mostly to check the possibilities to increase efficiency of the real systems. Consequently, the recommendations are provided.

Thesis starts from the analysis of the equipment. It is important due to fact that there are several ways to improve the efficiency based on the different parts of the equipment. It is possible to enhance, for instance, the electrical drive part of the compressor, piping or the compressor itself. The theoretical part is crucial for defining the base approaches for improvement and for avoiding inappropriate or too excessive measures.

Additionally, the theoretical part helps to build the simulation model. The model itself is based on the data collected from the real experiments and utilizes the analytical approach. The model consists of two main parts: a motor with or without a converter and a screw compressor with an air reservoir. The screw compressor generates load to the motor and therefore the compressor is core part when calculating the resistive torque and power. In the case of a direct on-line connection to the compressor part are added inlet and outlet valves, which are influencing the produced torque. Their influence are analyzed during the thesis. The torque and speed calculations are essential part of the model, because they allow estimation of the power consumption. Then the power consumption for different control approaches is compared for the same mass flow.

Thesis consists of four chapters.

In chapter one, components and working principle of the screw compressor are described.

In chapter two, the different electrical drive types are described. It is important to analyze the electrical part of the compression system since the variable frequency control is implemented completely through it.

In chapter three, there is description of screw compressor thermodynamic that provides the preliminary analysis of the compressor behavior. Compressor capacity control is described to provide a proper simulation.

In chapter four, Matlab Simulink models of the system are developed. In this thesis, two electric systems are analyzed: an air pressure system with direct on-line startup and the system based on the variable frequency control. It is assumed that the best way of improving efficiency of the air pressure system is to replace the valve control by frequency control of the screw compressor. Efficiency comparison of two systems is also held. The laboratory equipment is described and laboratory results are being compared with the simulation results. Additionally the recommendations of systems improvement are presented.

In chapter five, all the results are summarized.

1 Compressor description

1.1 Rotary screw compressor

Rotary screw compressors have been well-known and widely used in different applications for more than 50 years already. [5] The rotary screw compressor become rather popular in the industry due to following reasons:

Reduced maintenance. The screw compressor itself is very simple device, which has actually only two working rotating parts. It has no pistons, valves, etc. that would require often maintenance. Therefore, maintenance costs and time are significantly reduced.

Compression ratio. Screw compressors can provide from 2 to 20 compression ratio on a single level while providing high volumetric efficiency. In the comparison with reciprocating machines, screw compressors actually produce lower pressure, but at the same time reciprocating machines need more compression stages each of them with limitation of four per level. That leads to more sophisticated structure and also to requirement of inner-cooling for reciprocating machines. In the end, it leads to lower costs and higher efficiency for the screw compressors.

Small size. Having only one level of compression leads to small size of a compressor. Also it means that volumetric power density of the screw compressor is rather high. [3]

Wide operation range. The screw compressor can work with a wide range variety without requirement of any changes in the machine. It makes the compressor type especially suitable for operations, where the flow rates and other parameters are changed within a high range and speed.

1.2 Screw compressor working principle

The basic geometry of a rotary screw compressor is presented in Fig. 1.2.

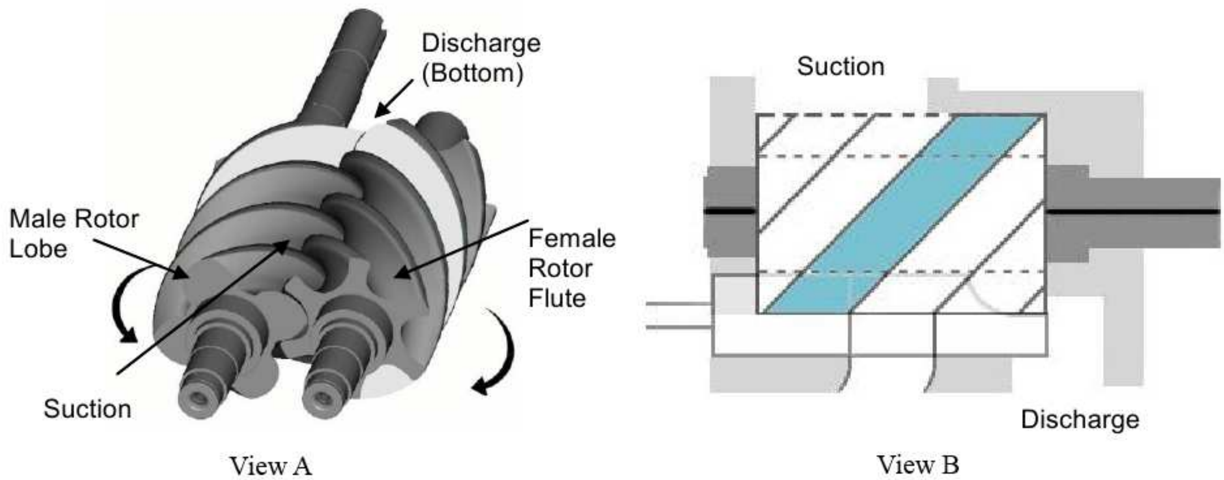

Fig 1.2 Rotary screw compressor a) male and female rotors b) cross-sectional view of the compressor [1]

The rotary screw compressor consists of two rotors: a male and female rotor. As the both rotors turn in outside direction, male rotor lobe disconnects from female rotor flute thereby forming an open area for gas to come. When lobes of the rotors connect to each other, gas will be closed and compression will occur. Enlightened area in Fig. 1.2 shows gas being trapped and moved within the volume. The same area also is shown on the cross-sectional side view of the compressor (view B). As it can be seen from the view B, there are two ports: the suction port, through which gas proceeds inside the compressor and the discharge port. Actual compression occurs on the bottom of the compressor when gas discharges through the discharge port with much more higher pressure.

Fig. 1.3 represents the typical structure of a screw compressor. In general, the structure of the screw compressor is simple: it has two rotating rotors and only one of them connected to the motor shaft. During the startup, the discharge port (1) is opened and the inlet port (2) is closed. When the motor reaches certain speed, the discharge port (1) is closed and the inlet port (2) become opened. After this, the supply process of the pressured air starts and therefore air goes through the inlet port (2). Air should be filtered first in the air filter (3). Actual pressuring starts in the oil storage (6). Oil from the oil storage (6) reaches radiator (11) through the pipe (7). Then oil goes to radiator (11), from which it reaches the compressor (4) through the pipe (8) and oil filter (9). Compressor (4) oil is got mixed with air. Then mixture of the air and oil returns to the oil storage (6), where division of air from oil starts. The division finishes in the air-oil separator. After that the pressured air through the pipe (15) reaches the outlet port (16).

Such implementation of the screw compressor increases the lifespan significantly. Firstly, it contains air filter, which cleans the air from dust and small debris. Secondly, the usage of oil in the system decreases friction in the compressor. It is very important, since for efficient work of the compressor, two rotors should fit each other.

In the screw compressor while both rotors unmeshing with each other there will appear free volume vacated by the male rotor. Then this vacated volume will be filled by suction gas and the volume will be increasing while both rotors disconnecting with each other. When both rotors finish the unmeshing procedure, flute reaches the edge of the suction port and in this exact moment the maximum volume is reached. After that the actual compression process occurs. Both male and female rotors start meshing in the other and of the compressor and therefore decreasing volume available for gas.

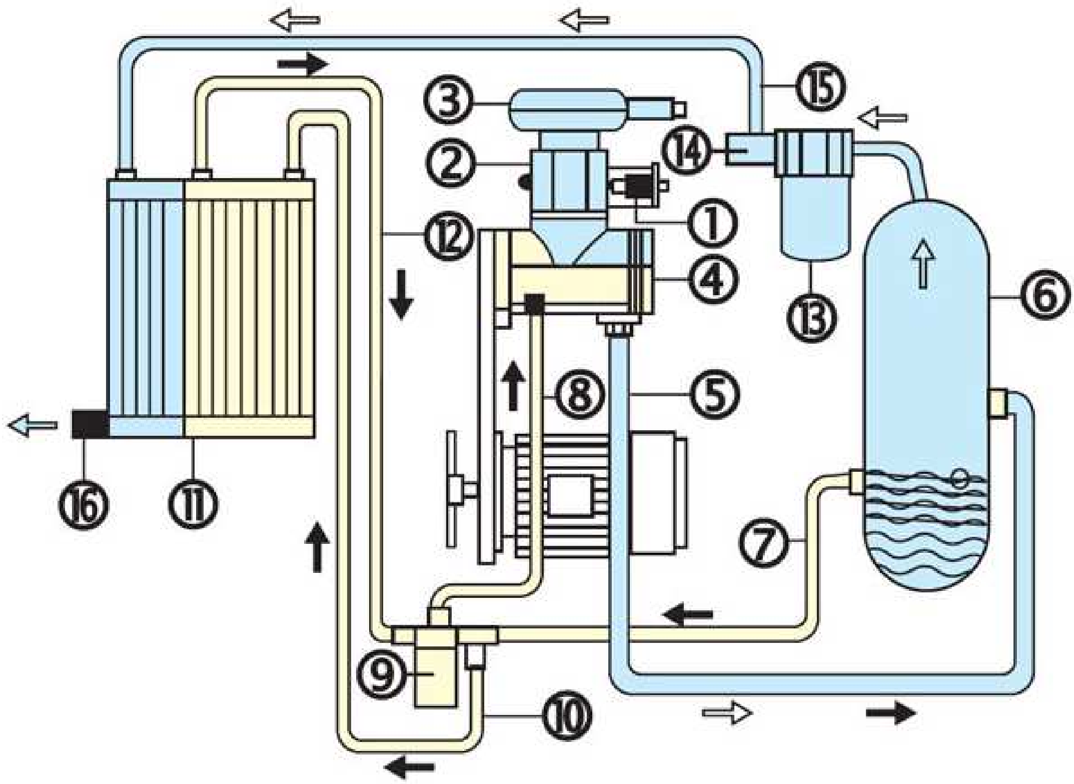

Fig. 1.3 Structure of the screw compressor [2]

1.3 Screw compressor construction

In Fig. 1.4, constructive plan of the screw compressor with a cooled rotor is presented [2]. The main rotor (2) is connected to the rotating mechanism (e.g. an electrical motor or an engine). It has convex and wide teeth while secondary rotor has concave and thin teeth. Torque from rotating mechanism is transferred to the lead rotor through the direct connection or if it is required through a reducer or a booster. The biggest part of torque is transferred to the male rotor while the female rotor depending on the type usually experiencing only a small part of torque. One of the feature of the screw compressor is that there is no direct contact between the male and female rotor. Torque is provided through the oil film in the case of oil injected compressors.

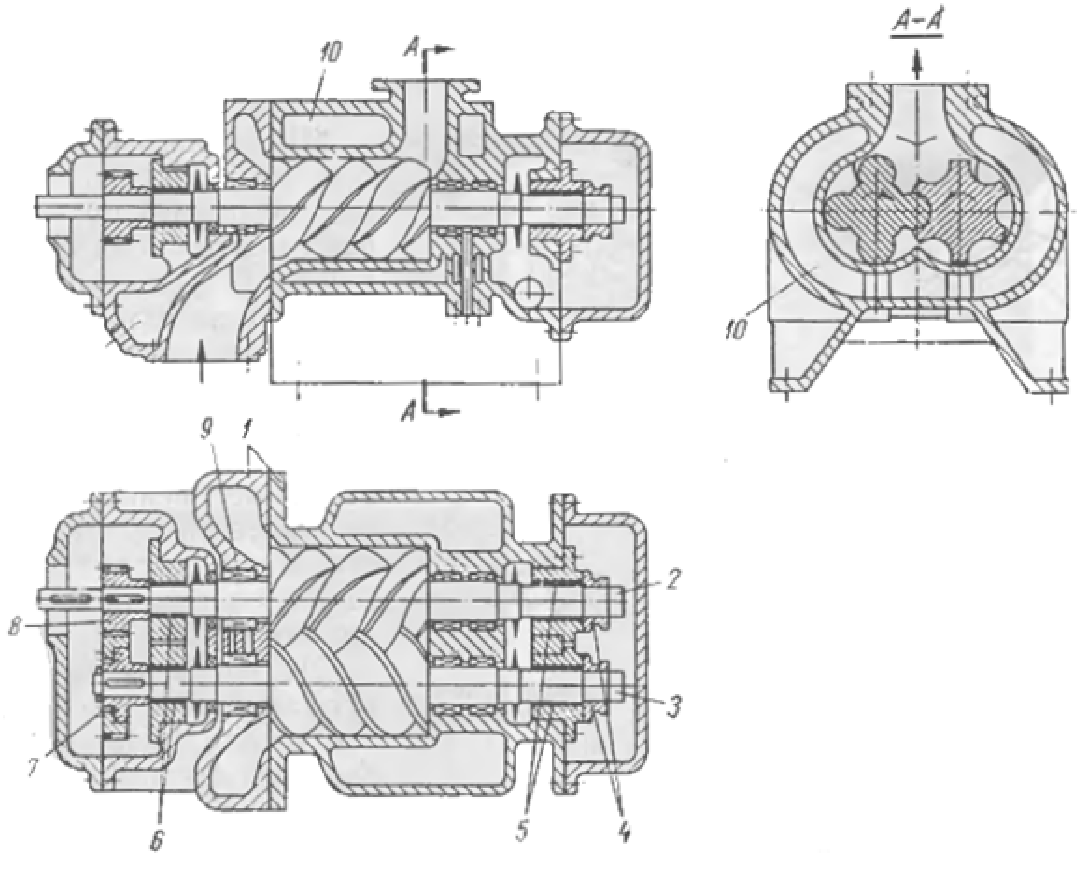

Fig. 1.4 Screw compressor constructive plan [6]

The diameter and length of rotors define the output pressure and the total gas capacity. In general, the length of the rotor determines output pressure. In this case, the longer rotor has higher output pressure in comparison with the shorter rotor. The diameter of the rotor assigns to output gas capacity.

In the ideal case, the distance between teeth of two rotating rotors should be equal to zero. At the same time, teeth apex that creates cylindrical surface during rotation must have theoretical distance with the compressor body also close to zero. However, both of the distances should have values that can provide safety for both screws without damaging the compressor body or rotors. The most important, distances play a significant role in defining the efficiency of the whole compressor. In the case of the screw compressor with lubricant there is might tangency between two rotors but still no connection with the body of the compressor.

In Fig. 1.4 under number 1 the casing is presented. The casing consists of different parts and has inside boring to fit both rotors. Casing includes internal chamber (10) that provides circulating cooling liquid or if pressure is not high it has ribs for external air cooling.

The main target of cogwheels (7 and 8) is to synchronize both rotors and do not let them to interfere with each other. On the rotor shaft there are might be additional details such as sleeves, sealing, oil rings, etc.

In the most cases, there are two types of bearings (7, 8) which are sliding bearings or rolling bearings. Axial stress, which interacts with both rotors, is perceived by thrust bearings (4). Near the shaft in some cases there are seals (9). Rotating speed of the compressor is assumed equal to the speed of the lead rotor.

It is clear that the higher impermeability of chambers with pressured air that are formed during rotor rotation, the higher economical efficiency and higher output pressure. However, at the same time the compressor provides higher requirements to injection pressure alongside with equal internal pressure.

Processes of suction, compression and extrusion follow after each other for every air chamber. Due to rotor high rotation speed there is rather low output pressure deviation [6]. For example, with the speed of 30 000 rpm and amount of chambers equal to four the amount of air portion is equal to 120 000 per minute. In case of huge compressors with speed up to 3 000 RPM amount of air portions per minute will be equal to 12 000. [6]

In the screw compressor, there are no valves or any other distribution devices. There might not be any lubricant for the rotors. The screw compressors with cooling liquid or with internal lubricant (oil, etc.) usually do not have casing. Excessive heat inside the compressors is dissipated by lubricant or cooling liquid.

The screw compressors are widespread in different types of applications, for instance, in natural gas applications, refrigeration, fuel gas compression, and vapor recovery systems. [7]

Screw compressors can operate using wide variety of working fluids that consist of gases, vapours or mixtures that can change their phase during operation. Screw machines can use different kind of lubricants or can work even without any internal lubrication. Thus, there are two main types of screw machines: oil free and oil-injected [6].

The first group is so-called machines of dry compression or more known as the oil free machines. Output gas from such machines does not contain any lubricant or product of equipment deterioration. Inside of the working space of such compressors there are no lubricants or cooling liquid. Cooling of such machines can be achieved by using external blowing or by different kinds of external liquids, which stay in contact with the casing.

Analogically, the second group is machines of wet compression or oil injected machines. In case of the group output gas is mixed with lubricant as a result. The second group can be also divided by amount of used lubricant. There are oil-filled compressors and compressors where the amount of lubricant is rather small. In the case of the second group special equipment such as filters, pumps, separators that can separate lubricant is required. Multi-stage separators detach lubricant from the output gas and move gas to the aftercooler.

Lubricant should be also cooled in order to return to the compressor. The main difference between two groups is the case construction. Also the first group can provide higher output pressure and such compressors have more simplistic construction (e.g. they do not require cogwheels or seals). However, friction between oil and rotors appears and therefore mechanical losses are much higher. Also some energy required to deliver lubricant and to detach it from output gas. At the same time, oil injected allow to produce output pressure around 14 atmospheres for only a single step. To produce such pressure using dry compressors there would be required additional fridge to decrease gas temperature. Oil injected and oil free compressors are presented in Fig. 1.5.

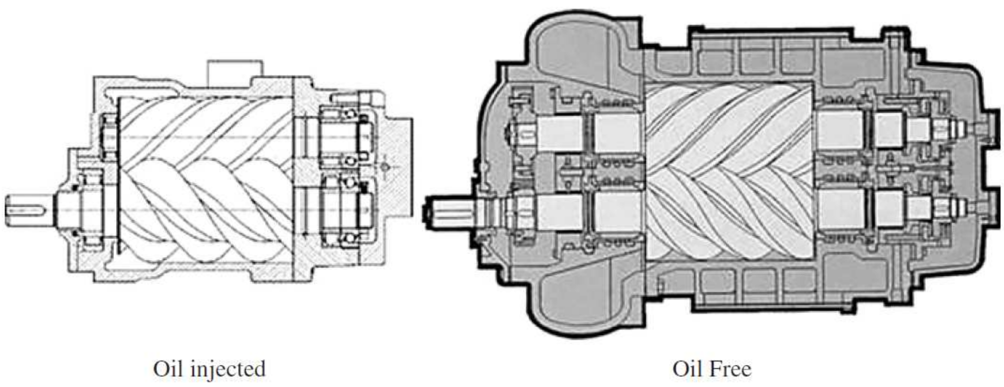

Fig. 1.5 Oil-injected and oil free compressors [5]

According to Fig. 1.5, it is clear that the oil injected machines have generally much smaller casing or no casing at all. It leads to smaller price per kilogram, simpler construction, cheaper manufacturing, and relatively higher efficiency. In order to prevent lubricant access to the working chambers, the oil free machines require seals installed on the shaft between the bearings and working chambers. Production of effective seals nowadays is still problematic for manufacturers and, however, it follows the fact that oil free machines considerably more costly to manufacture in comparison with the oil injected machines. In spite of the fact that oil free compressors are less efficient, the oil free output gas can be achieved almost exclusively by the screw compressors. Also for different gas should be used different lubricant. Improper lubricant may have unacceptable viscosity during the start-up or speed change. In addition, the oil injected compressors usually have secluded oil system and therefore oil amount should be minimized. Any oil leakages can lead the system to a fail if leakage is not detected properly.

There are also compressors with more than two rotors [6]. However, the efficiency of the middle rotors is rather reduced due to low chamber gas filling. In addition, the construction is rather hard for implementation. Such compressors found their usage only in some special technological processes. It is expected that such machines can be used as engines in the future. There might be difference in rotors' position, suction port or discharge port positions. However, the rotor geometry and their construction is not depended on the amount of the rotors.

Effective work of the screw compressor is defined by rotor design. Nowadays, technology allows to reduce internal leakages significantly and also to make clearance as tight as possible. The main requirement to the rotor is the maximized internal flow area alongside with reduced both internal leakages and friction. The higher flow and smaller leakages both improve volumetric efficiency of the compressor. It also leads to increased adiabatic efficiency since mechanical power of the electrical engine is used more efficiently to compress the gas. Most of the screw compressors are still manufactured with four lobes on the main rotor and six lobes on the gate rotor, which leads to the same outer diameter of both rotors. However, in order to produce higher output pressure ratio, different rotor designs appeared.

The most common rotor types of screw compressors with commercial name or patent are presented in Fig. 1.6. The first group consists of rotors with four lobes in main rotor and six lobes in the gate rotor. This is the most common rotor types because they are the most universal and can fit most of the applications. The largest group of the rotors is type of 5/6 that nowadays becomes more popular because it provides large displacement with large discharge ports alongside with decent load characteristic.

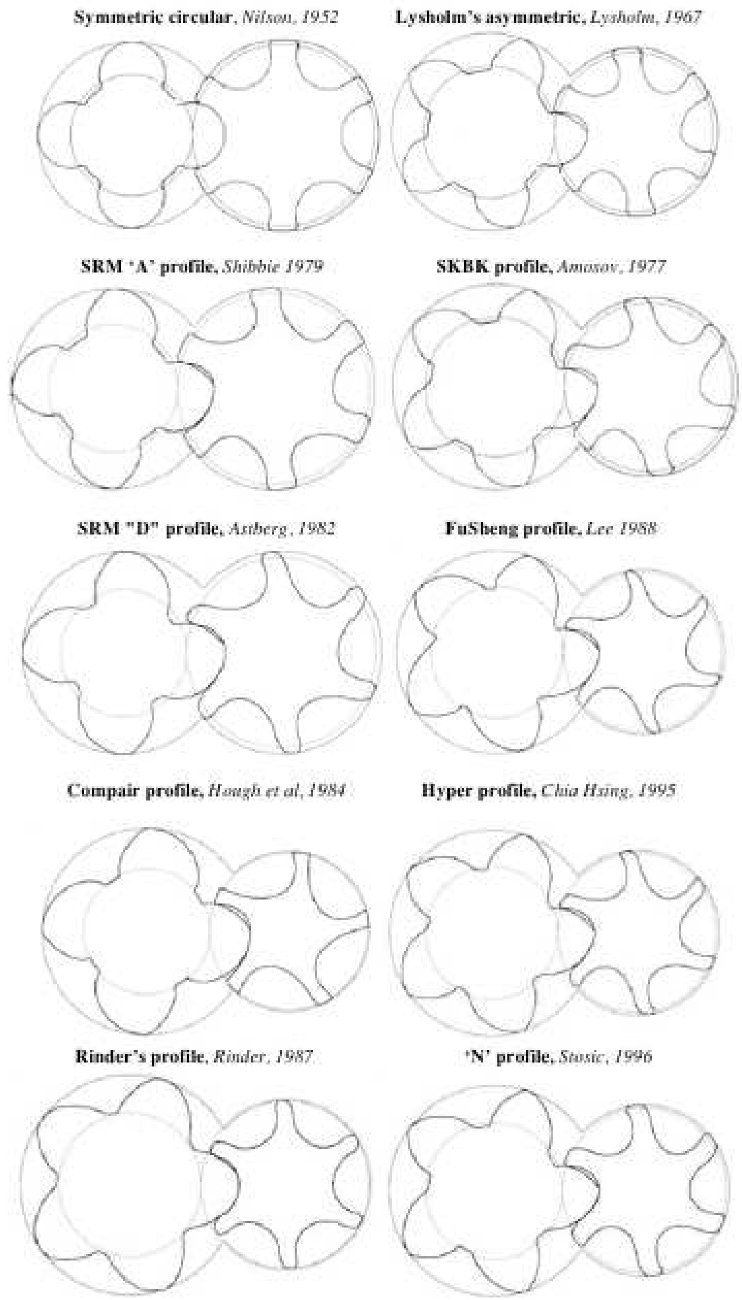

Fig. 1.6 The most common screw compressor rotor types [5]

Teeth profile can be also different, which are troichoid, circular and elliptical. However, there are certain requirements to the contact line between the teeth on both rotors. Firstly, the contact line between the teeth should be continuous. The requirement can be reached by any teeth profile. Secondly, air chambers should be sealed. In other case, it would be possible for gas to get direct access from the injection chamber to the suction chamber. The second requirement in real cases is not always achieved but any holes should be minimized. Thus, the teeth profile, teeth lean angle, and their length are the most important parts of the compressors and thereby teeth should be manufactured with high accuracy.

Unfortunately, there is no common rotor construction for all applications at once. For different applications, such as refrigeration or compression, specific compressor design is required. For example, huge compressors require high efficiency since they are mostly working in the factories or manufacturing companies with long runs. In housing applications, the most important value is size and therefore efficiency is not of importance.

Bearings of the screw compressors are characterized by high rotational speed. The usage of the roller bearing can simplify construction alongside with decreasing price of small screw compressors. Medium and huge performance compressors have significant bearing reaction around of a couple of tons with injection pressure around four to five atmospheres. In this case, the slider bearing are used in machines. However, the slider bearings for operational needs require compulsory lubricant injection. In some cases it may lead to installation of an additional lubricant pump. However, there is a drawback: viscosity of lubricant can change significantly and it may lead to an additional rotor displacement from the geometrical center during high speed changes.

1.4 Screw compressor losses

1.4.1 Gas leakages

Screw compressors like any other compressor type have leakages that affect the total compressor performance. Gas leakages can impact the process by changing mass flow rate and therefore affect both thermodynamic and volumetric efficiencies.

In general, gas leakages can be divided into internal and external gas leakages. External gas leakages appear through the seals on the shaft ending of each rotor. From the discharge port, gas leaks to the quarters where the compressor is situated. The suction port in addition may have increased gas income that also can affect output pressure.

Internal leakages can be divided into two types that both affect compressor parameters. [5] The first type includes leakages between both rotors in the sucking period that affects compression coefficient, efficiency and compressor power. It appears form the chambers with high pressure and then proceed back to the suction chamber if the rotor clearance is large enough. It significantly decreases amount of gas received from the source. Moreover, leakages have higher temperature than the received gas and it leads to decreased gas density in the chambers. In the end, the compressor requires more power to produce the required output pressure.

Another type is leakages between chambers during the compression period. They do not have high influence on the compression coefficient, but have rather huge impact on the efficiency and power. They indirectly increase leakages of the first type. Also they change amount of gas that stays in the isolated chamber same as the leakages of the first type and therefore increase the compressor power consumption required to compress the gas. The rotor lobe has two edges both of which face different pressure zones. The leakage can occur from the discharge pressure zone (leading edge) to the suction pressure zone (trailing edge). During the compression, gas leaks from one chamber to another to increase its pressure from the preceding chamber. It leads to the constant flow loss. Accordingly, because of the lowered pressure inside chambers compressor requires more power to produce required output pressure. In addition, both leakage types do not compensate each other.

Basic concept of calculating leakages follows idea that gas flow through the holes (clearance). The process should be assumed to be adiabatic. For calculation, simplification gas estimated having constant temperature rather than enthalpy.

All sealing lines in the compressor form their own leakage gas behavior. There are four different sealing lines in the compressor: sealing line between the rotors and casing, sealing line between casing and both gate and main reverse tip, lobe sealing lines between rotors, sealing line formed by the housing, and rotor discharge front.

1.4.2 Passage and port losses.

The main difference between the screw compressors and reciprocating compressors concludes in the fact of unreliability of discharge and suction ports of the screw compressor in regulation of gas flow.

Therefore, approach used for calculating valve losses in reciprocating compressors cannot be applied for the screw compressors. In that case, group of alternate factors should be taken into account.

In most screw compressors working gas is moved through the ports placed inside the compressor housing. The size requirement for the inlet port is to minimize entrance flow losses. Same rule can be applied for the discharge port exit losses. At the same time, the location of the discharge port should be considered as the most critical factor. Pressure is increased during steady decreasing of the gas volume inside the compression chambers. In the case of the screw compressor, where there are not any discharge valves, the process of compression proceeds until the point of uncovering the discharge port. Therefore the static location of the discharge port allows having constant output volume ratio. Because of that, the incorrect port position can decrease the overall compressor efficiency.

The isentropic volume exponent of the gas defines the overall time to reach the desired pressure. If the discharge port located too early, the overall volumetric efficiency will be decreased due to gas leakage back into the groove. Consequently, if discharge port is placed too late, overcompression may occur causing additional power losses.

1.4.3 Heat transferring effects

Screw compressors provide ability to have a compression ratio up to 20 only at single stage. This is possible because of the coolant injection inside the chamber during the process that prevents penalty of significant discharge temperatures. The compression relationship is presented below [5]:

$$P_{\text{in}} = m_{\text{gas}} h_{\text{gas}} + m_{\text{coolant}} h_{\text{coolant}} \tag{1}$$

where m_{gas} is the mass flow rate of the gas, h_{coolant} the specific enthalpy increase of the coolant, m_{coolant} the mass flow rate of the coolant, and h_{gas} the specific enthalpy rise of the gas, P_{in} – input power.

However, coolant injection has two-sided effect. The injected coolant decreases the effective volume left for the gas. From theoretical point of view, it might lead to reduced capacity. On practice though, the coolant assists in lowering of the discharge temperature, which leads to the almost isothermal compression thereby making higher isentropic efficiency.

1.4.4 Pulsation effect

Pulsation effect of the screw compressor has much higher frequencies in comparison with reciprocating compressors, therefore proper diameters and lengths of pipes should be calculated and installed. In general, compression occurs up to 12 times per revolution typically at speed of 3600 rpm in case of the screw compressors [5]. Consequently the higher frequencies lead to the higher their influence on the performance.

1.4.5 Friction

The screw compressor like any other compressor type has sliding parts because of friction occurs. Additional power is required to overcome the friction. Moreover, any part of the compressor affected by friction has high temperature and therefore it heats the gas inside groove. [5]

2 Electrical drives

Analysis of the world experience in manufacturing of new equipment alongside with modernizing the old one show the high dynamic in the development of electric drives and control systems. The reason behind that is the aspiration to achieve high performance and quality with increasing level of manufacturing automation. There are different tends in electric drives development such as: high usage of electric drives in the industry, replacement or modernization of uncontrolled electric drives in the energy consuming equipment with controlled electric drives in order to achieve high energy efficiency, usage of modular systems with embedded controllers and implementation of cascade control systems, etc. Before 1950s, all energy consuming applications required DC drives. AC motors were not capable to provide précised speed since they were operating at almost synchronized speed with frequency of the power source. In addition, almost all control methods for AC drives used energy consuming approaches with resistors, etc. Control flexibility of general-purpose AC drives in comparison with DC drives used to be limited and AC drives were used only in applications with gradual speed change, such as pumps, fans, etc. [17] [8]

An electrical drive is assumed to be electromechanical system that includes energy converters, electromechanical and mechanical transformers, control systems for converting and transferring electrical energy from the power source to the mechanical load in order to control different processes. The common structure of any electric drive is shown in Fig. 2.1. Some parts of the scheme might be optional (for instance, sensor signals, etc).

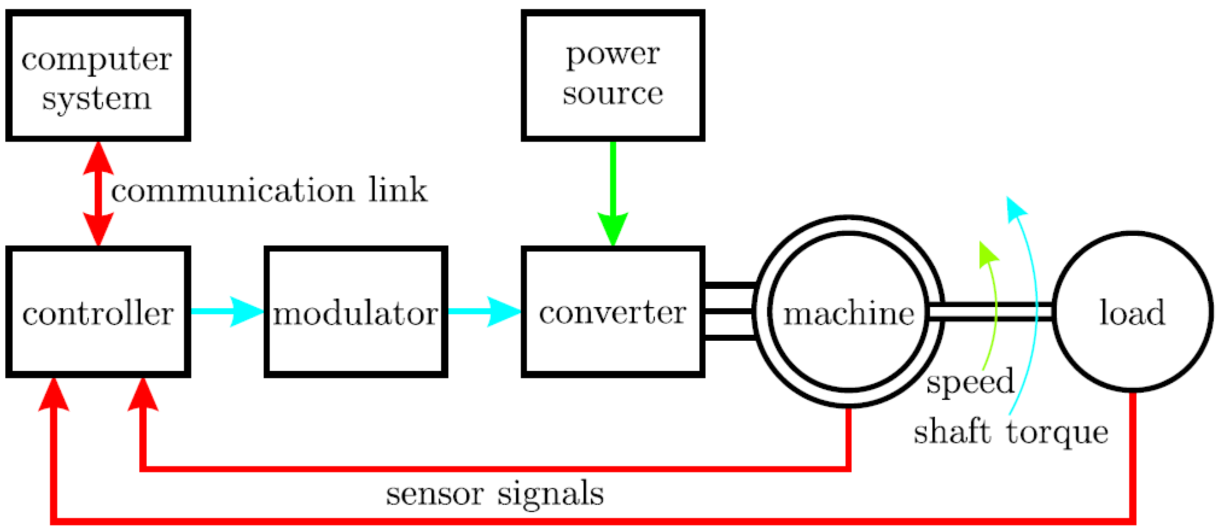

Fig. 2.1 Common electric drive structure [4]

In general, the converter is controlled directly by a controller or through the modulator. The controller should be also able to communicate with high-level computer systems through the communication link. It is important because electric drives operate in connection to each process and it is required to estimate the operation process of the whole system on the high level. Different control approach requires different sensor signals and therefore different control algorithms should be implemented.

Different drive application (in most cases a mechanical system) has different requirements to be fulfilled. As it is shown in Fig. 2.2 electric drives have four operating regions with wide power range [17] [14]. The vast majority of the electric drives require an operation range only in the second and the first quadrants [9]. Energy transmission in this scenario is one-sided leading to fact that electrical energy is just being transferred through the electric drive parts to the mechanical load [8]. However, sometimes it is required also to convert mechanical energy from the load to electrical energy (regeneration capability) and it leads to the requirement to operate in all four quadrants. Electric drives that allow two-direction conversion are called as bi-directional. Nowadays, electric drives use power electronic equipment to control the energy conversion in both directions.

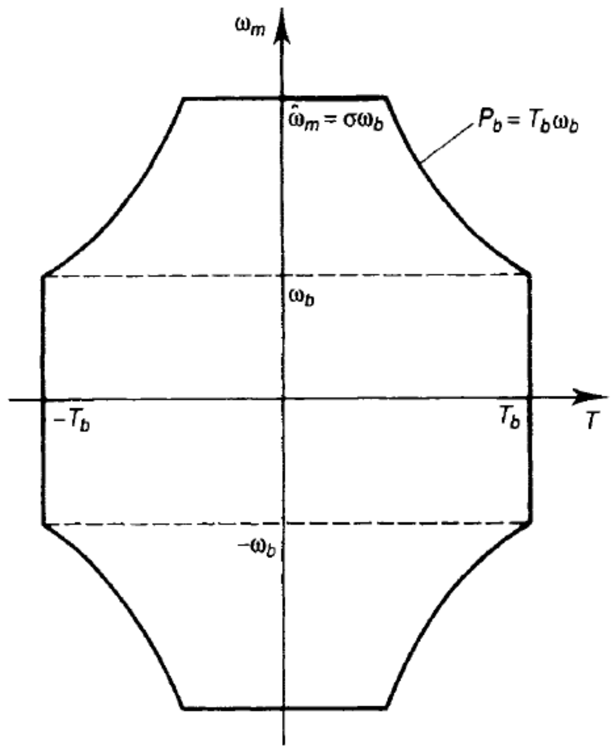

Fig. 2.2 Control range of electric drive [17]

An electrical machine is the main electro-mechanical converter in the system. It should be controlled in order to fulfill the process requirement. There are many types of electrical machines but the most commonly used are asynchronous machines, permanent magnet synchronous machines and switched reluctance machines. However, brushed DC machines are still very common especially in the small applications. [11]

2.1 Induction motor

Electric drives typically use electrical motor types as an induction machine (an asynchronous motor), a synchronous machine (with or without permanent magnets) and switched reluctance machines. The most common machine type in the industry is the induction machine, which is shown in Fig. 2.3. It is related with the fact that the induction motor has a lot of features over other machine types. The only drawback used to be a requirement of sophisticated control algorithms and equipment. However, appearance of digital embedded systems that have low prices alongside with enough performance for an electric drive to implement field-oriented control skyrocketed induction motor electric drives in the industry. As a result, the induction motor allows the electric drive to be a brushless system with low maintenance cost without any position sensors and dynamic performance higher than in the case of the DC motor. [12]

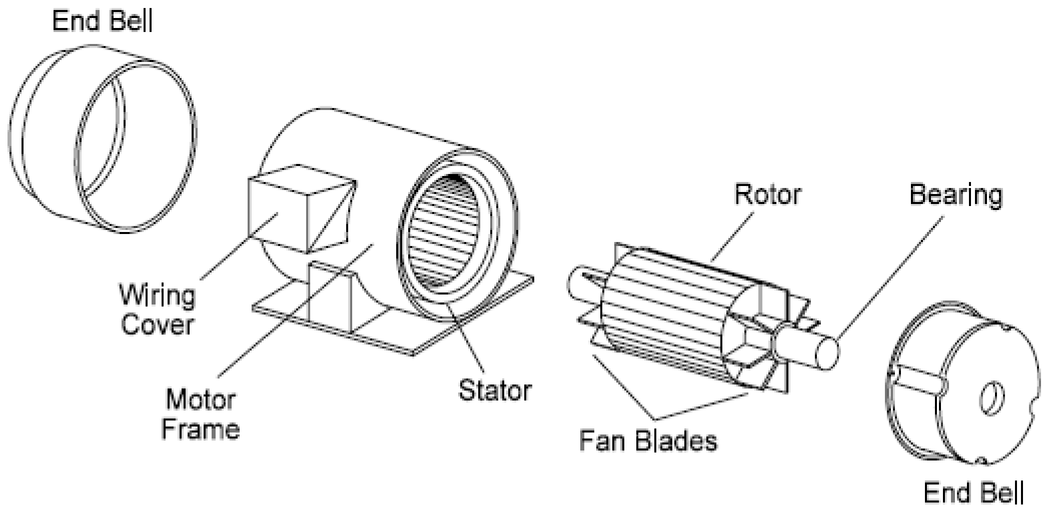

Fig. 2.3 Completed structure of induction motor [8]

Most of the induction motors are connected to three-phase inverters that produce in ideal cases sinusoidal voltages and currents with controlled amplitude and frequency. As a result, flux density produced inside the motor has sinusoidal distribution around the air gap and it rotates with angular velocity equal to the stator currents frequency. [11]

The most common industry induction motors have a squirrel cage rotor where conductors have short circuits between each other through the aluminum end rings.

In the past, most induction motors were designed in a way to have as low initial cost as possible. Nowadays, the performance and energy efficiency both have the highest priority. It is related to the fact that the induction motors (especially with large size) are planned to work for long periods around anniversaries. Losses in both rotor and stator windings can be estimated as 20–30% of the total losses. In laminated iron of yoke and teeth, there are also core losses that should be considered around 15–20%. Stray losses or eddy current losses assumed to be around 10%. [18] However, the larger motor the larger stray losses since they are proportional to square of current inside the stator. Friction losses are estimated around 5–10%. It is clear that to increase the efficiency there are more copper required to decrease the total winding resistance alongside with the temperature. Higher quality iron with lower flux density should also be used to shrink the total eddy current loss. Stray losses might be decreased by increasing the air gap. However, such machines require higher axial length. [8]

In comparison with commutation machines, the induction motors have much lower maintenance cost and lower price. They can also be built with closed casing if it is required to work in dirty or dangerously explosive conditions.

The biggest majority of the induction motors works in constant speed applications. However, constant speed applications tend to be reconsidered as low energy efficient and as processes with low flexibility with poor output quality production. Especially, it is related to quite common industry processes, such as pumps, fans, coolings, etc.

In spite of the fact that variable speed drives with highly effective motors increase total cost of the system, the cost might be compensated by the decreased energy consumption and reduced heat losses.

2.2 Electric drive types

The most typical variable speed drives are presented in Fig. 2.4. From Fig. 2.4 it is clear that all drives can be divided into three main groups, which are mechanical, hydraulic and electric drives. Each group has drawbacks and advantages.

Historically development of electric drives slowed by the complexity of the control systems even for DC-drives and were used only with high priority applications. Eventually, it led to wide spread of mechanical drives due to their low cost and overall simplicity.

Hydraulic drives allow system to have a soft-start without implementation of different control types like in case of electric drives. They mostly used in transportation and conveyor applications because of high starting torque. [8] The basic principle is to transfer torque from a prime mover through the hydraulic pump to the hydraulic engine. The speed is regulated through the controlling pressure ratio or fluid flow.

In electric drives different approach is used, which is focused on the speed control of the engine directly. Electric drives that control DC machine are called as DC drives. Analogically, electric drives with AC machine are called AC drives.

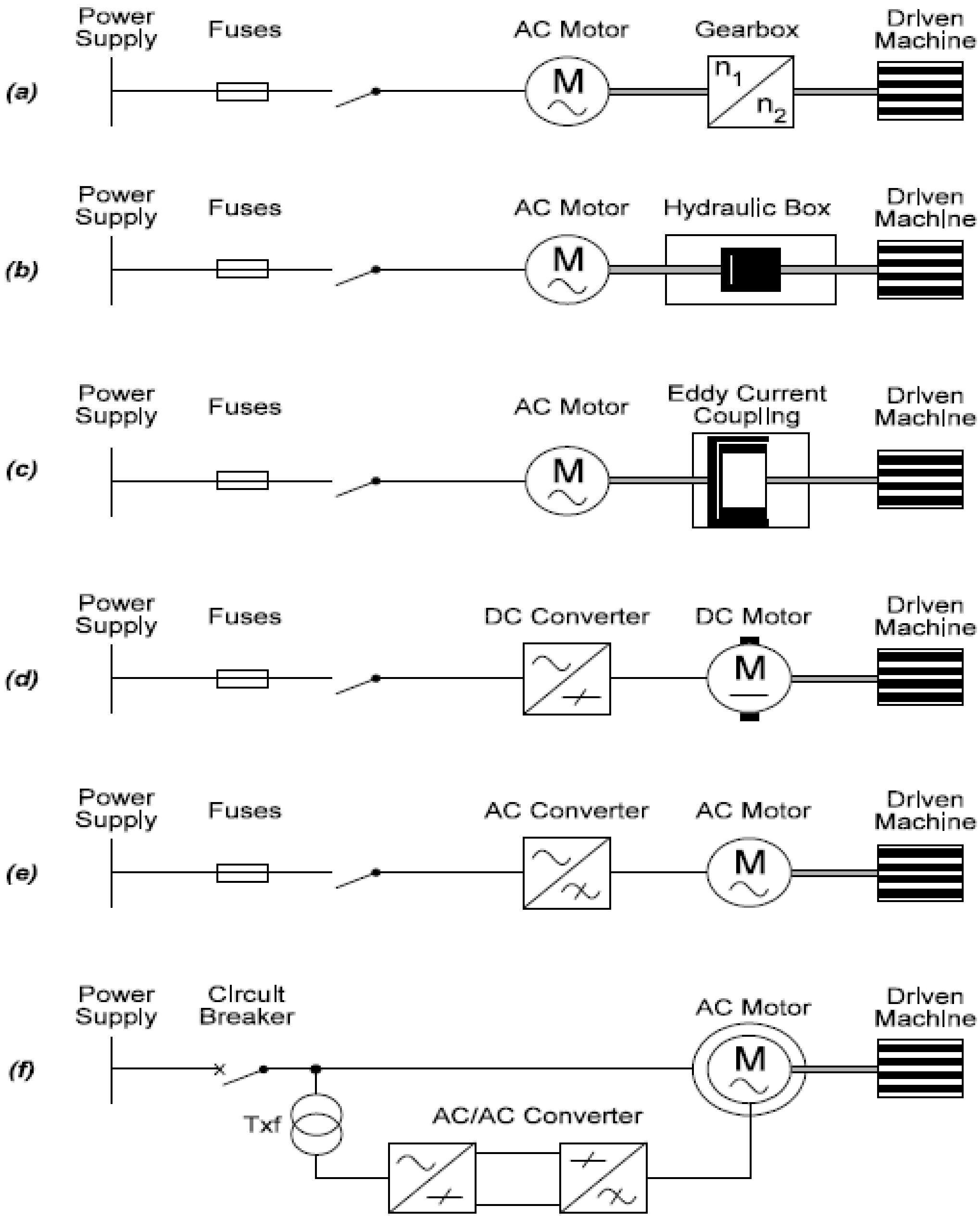

Fig. 2.4 Variable speed drive types [8]

In Fig. 2.5, main electric drive types that are used in the industry are presented.

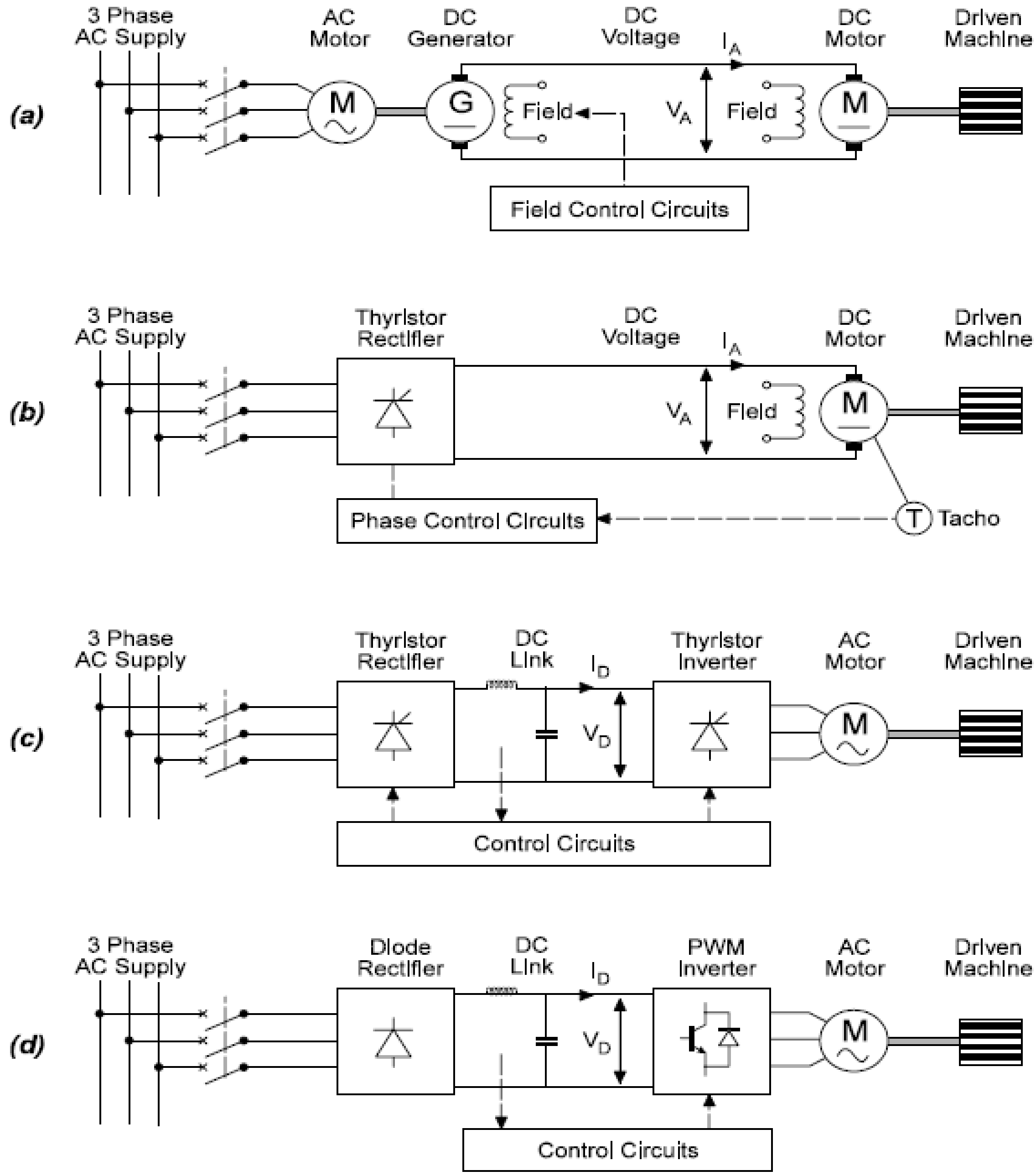

Fig. 2.5 Electric drive types [8]

The Ward-Leonard system consists of 3-phase AC induction motor, which is connected to the DC generator with separate excitation. The DC generator is connected to the DC motor and therefore the Ward-Leonard system is assumed to be a DC drive. Output voltage of the DC-generator is used in speed control of the DC motor. This system allows having excellent torque and speed characteristics. The Ward-Leonard system used to be rather popular in the industry for a lot of decades. It allowed having accurate speed control and in general the system is simple and reliable. However, it has a huge drawback due to rather low efficiency. It is related with the fact that the system consists of three different electrical machines, where an AC machine power is almost double power of DC machines and in the end it leads to high energy consumption. In addition, maintenance cost and repair time are both rather high mostly due to commutators and brushes exchange.

Since 1970s, the controlled AC/DC converters became more popular in industry. In comparison with the Ward-Leonard system they provide lower maintenance cost. In addition, initial price of the system itself is much lower. There are different configurations of the rectifier consisting of a 12-pulse bridge, a 6-pulse bridge or a half-way 3-pulse bridge. [8]

The most common type of rectifiers used in the industry is the thyristor rectifier. It consists of the thyristor rectifier, electronic phase control circuits and a DC motor. The thyristor rectifier is usually a 6-pulse bridge full wave type, where output voltage has low distortion. For large applications, a 12-pulse bridge rectifier with even more reduced harmonic components is more commonly used.

The efficiency of the system with an AC/DC converter is relatively high around 90% depending on the DC motor size. Converters have high durability and simple structure and at the same time can provide power up to several megawatts. The main disadvantage of the system though is a usage of the DC motor whose reliability and efficiency are being lower in comparison with the AC motor. [11]

However, the main advantage of the DC motor is flexibility in terms of speed and torque control. There are still some applications with high demand speed precision and fast dynamic reaction that use the DC electric drive. For example, the speed of the DC motor is controlled either by the field flux or armature voltage. In most cases though there is a requirement to provide constant flux in order to increase the motor speed by increasing armature voltage. The DC motor can also use field weakening approach if voltage reached maximum value. The motor is able to provide nominal torque even in stand-still. Still the output power is zero at zero speed; while torque is constant, the output power is defined in proportion to speed. At the same time, in the field weakening zone, the motor torque decreases proportionally to the speed. Still, the output power remains to be constant (Fig. 2.6)

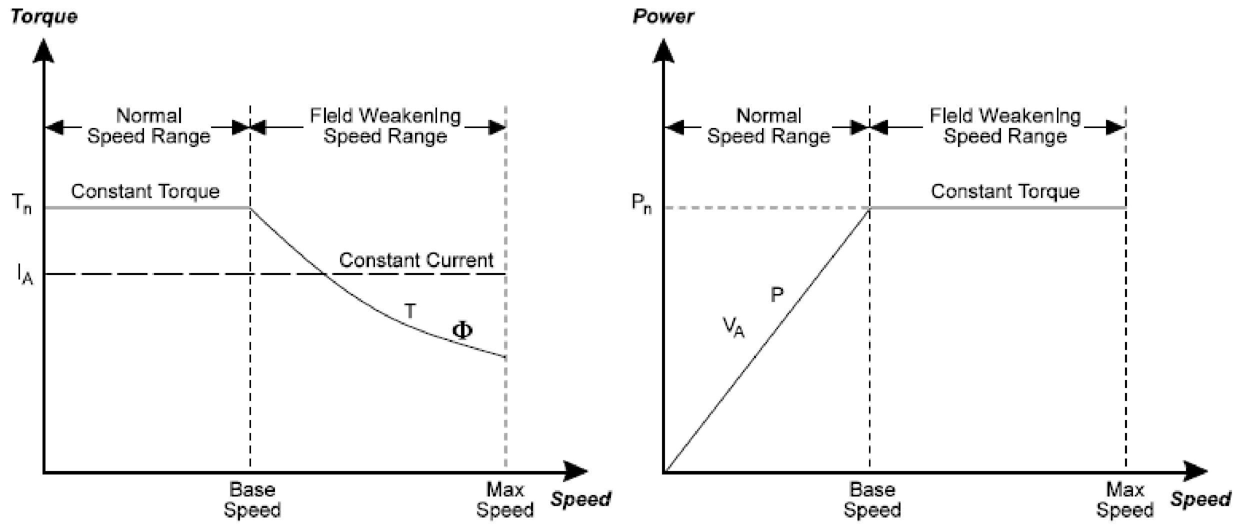

Fig. 2.6 Field weakening zone [8]

The variable speed drive with the AC motor uses an inverter as a main commutation system. The main problem still is a requirement of sophisticated control algorithms. For example, frequency control has been known as an approach of controlling speed of AC motors for decades. However, it only recently became a viable option from economical point of view to control speed of the motors.

Advancement in the development of power electronic devices and digital control systems allowed increasing efficiency of the inverters and significantly decreased the total cost of the variable speed drives. The main feature of the AC drives was always higher reliability and lower cost (especially cost of the motor) in comparison with the DC drives.

In circuits below 100 kW transistors have almost completely replaced thyristors. They provide huge advantage of being able to be completely turned on or turned off by controlling base current without implementing additional circuits with auxiliary elements. It significantly lowers cost of the components and raises the stability. Bipolar junction transistors are the most popular type inside the transistor family. It is related mostly to their low cost. However, additional circuitry is required in order to implement base control of the transistor. Negative base voltage is required for the transistor to stay in the off state and it leads to requirement of current extraction from the base.

IGBT and MOSFET transistors do not suffer from the destructive loss from the base current. Moreover, both devices have possibility to be turned on and off by the minimal gate current. It means that in long terms IGBTs or MOSFETs with higher initial price are even more cost efficient than BJT transistors.

Gate turn-off thyristors alongside with MOS controlled thyristors both provide main benefits of thyristors and transistors. They can be controlled by current pulses forwarded to gate. They mostly used to control motor with power of few hundred kW.

2.3 Vector and frequency controls

Vector control as a term is one of the most used term in the control of industrial processes. The technique was well-known since 1960s but became available only with the development of microprocessors and power electronics in the mid 1980s.

Interaction of magnetic fields in the fixed part (stator) and rotating part (rotor) produces torque as the output of an electric motor. Strength of both magnetic fields defines the strength of the torque. Magnetic fields are the results of the currents flowing in the windings. As a result, the torque is proportional to the product of the currents, which produce the magnetic fields.

Vector control has been promoted as technology that makes AC drive equal to DC drive. In DC drive output torque is a product of current vectors. The armature current produces the torque while the field current produces flux. In normal conditions the field producing current stays constant and consequently the output torque is directly proportional to the armature current. In this case the armature current is often used as a feedback in the closed-loop cascade DC drive controllers considering the fact that both armature and flux currents can be measured by current sensors.

Unlike in the DC machine, the flux producing current and torque producing current inside the induction motor cannot be measured separately for AC machine. The angle between both current vectors changes in time. However, stator current is being as a sum of both the torque producing current vector and the flux producing current vector can be estimated.

The main purpose of an AC vector control is to detach both of the producing current vectors, to estimate them and to provide possibility to control them separately under all speed and load requirements. The main aim of DC drive control is to maintain constant flux current.

The estimation of the current vectors requires the measurement of different variables (motor speed, stator current, frequency, stator current, phase angles, etc) and setting them to the motor model. The motor model consequently includes engine parameters (constants) such as the stator inductance and resistance, rotor resistance and inductance, the number of pole pairs, etc.) Various models consist of different amount of parameters depending on the aim, from simple calculations to rather comprehensive and accurate imitation, which is close to real motors. The higher model accuracy, the higher requirement for the processing power.

Therefore, the most important of the vector control is the motor model, which continuously calculates and simulates different processes inside the motor:

1) Torque-producing current is being calculated through the storing initial motor parameters inside the memory, measuring voltage and current for each phase and through the calculating or measuring motor speed.
2) Flux producing current is also being calculated.
3) Speed control is implemented by comparing speed data with the initial speed setpoint. Speed error output then is provided to the torque control loop.
4) Torque control loop is implemented through the torque, measured or calculated through the current and speed feedback data. Torque error output then is sent to the controller to adjust power electronic devices control.
5) Maintains control of the output parameters and adjusts them depending on the task or process.

Nowadays, AC variable speed drives uses vector control to some extent depending on the application. There are three main types of AC drives available on the market: fixed voltage/frequency drive, sensorless voltage/frequency vector drive and closed loop field oriented vector control drive.

The fixed voltage/frequency drive is suitable for controlling applications with low dynamic requirements (pumps, fans, compressors, etc). It is an open-loop system, structure of which is shown in Fig. 2.7.

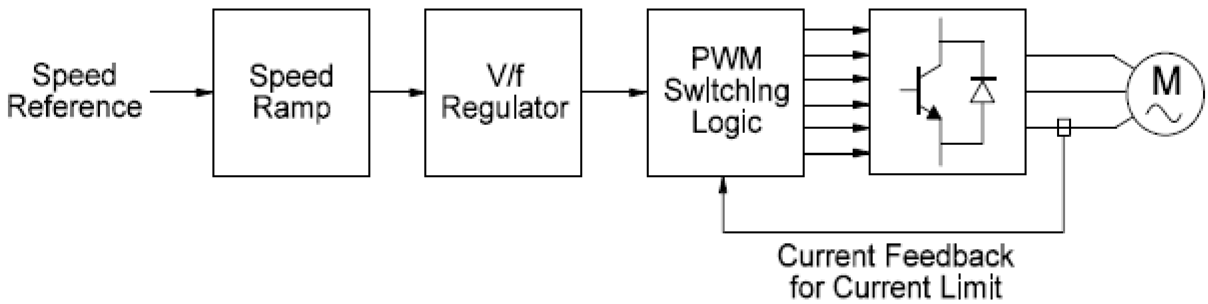

Fig. 2.7 Block diagram for fixed voltage/frequency drive [8]

Speed reference signal is taken from the external source. Then it is converted from the step signal to suitable for the system slowly changing signal. Then signal reaches a block where calculation of magnitudes for both voltage and frequency. The main system requirement is to set voltage/frequency to be constant at any moment of time. The rate of change of voltage and frequency magnitudes defines the motor acceleration. Then signal reaches the module that realizes PWM switching logic based on the various PWM-algorithms. Voltage pattern then is provided to the power electronic switches. In most cases there is no speed feedback in the systems because of the lack of any speed sensors. The main target of the current feedback is an indication and protection against failures and therefore it does not participate in the control strategy directly.

Current feedback participates in the measurement of heat, sets current limit and provides protection for the power electronic components. The current feedback is also used for the slip compensation. The strategy allows having increased speed regulation for an induction motor without installing additional speed sensor.

The main drawback of the system covers in the fact that it is well-suited only for steady state systems with low dynamic requirements. Another drawback is the poor efficiency on the low frequency due to the high voltage drop. To avoid it there is a technique that implements start with additional voltage/frequency boost for the improved flux characteristics and consequently same applies for the starting torque.

The main target of voltage/frequency sensorless flux-vector drives is to reduce the voltage drop for the low speeds. The core of the drive is still fixed voltage/frequency drive. The main feature is the possibility to distinguish torque-producing current and flux-producing current through the mathematical model, which uses measured stator current. It allows using calculated torque-producing current to implement the current limit instead of the measured current. The flux regulator provides possibility to set different setpoints for voltage/frequency ratio and adjust it according to the task. Improved slip estimator is a core component of the sensorless voltage/frequency drive unlike in fixed voltage/frequency drive where it is an optional component. However, the main drawback is still lack of torque control implementation.

Vector controlled AC drives are aimed to provide the dynamic higher than DC drives. The block diagram of the vector control is shown in Fig. 2.8

Closed-loop vector control provides possibility to have a speed regulation around 0.01% and high dynamic response. In comparison with fixed voltage/frequency drives the dynamic response of vector control drives is 10 times higher. The block diagram of the vector control drive is a closed loop cascaded system with torque and speed control loops:

1) There are two control loops, which are separated for current feedback and speed feedback. Basically the idea is the same as for the DC drives;
2) Speed reference signal goes to comparator to define the speed error of the system, which is fed to the speed regulator;
3) The setpoint for the current (torque) controller is set to be the speed error signal. The signal is compared with the signal from the system current sensor. The current error then defines whether current should be increased or reduced;
4) Flux control loop is implemented;
5) The generated signal passes to the PWM switching logic block, where the control signal for IGBTs is created according to the desired task. The main target of IGBTs is to generate frequency and voltage and sustain it on the needed level and according to required algorithm.

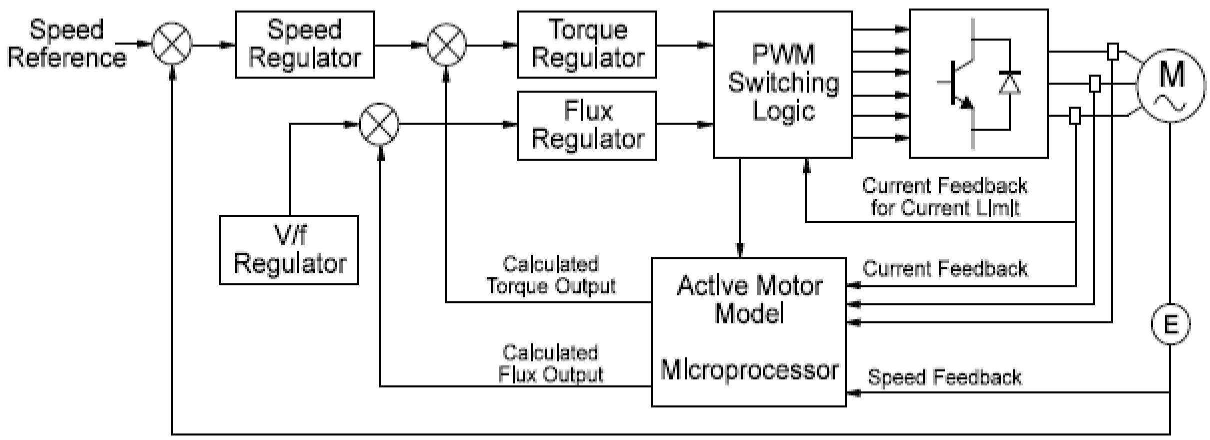

Fig. 2.8 Block diagram for vector control [8]

There is requirement to the shaft mounted speed sensor. In some cases, the speed signal can be simulated by the motor model, which is replacing the speed sensor signal. The drawback of the active motor model is that the calculated speed value can be inaccurate due to difficulties of measuring speed especially on the low speeds.

There are some disadvantages of the vector control in comparison with DC-drives:

1) The vector controller is far more expensive than the cascade DC controller;

2) Encoder speed feedback in most cases is required in order to get accurate feedback response from the system. Sensorless drives provide much lower dynamic response time if these are being compared with drives with sensors;
3) The startup of AC squirrel cage motor may lead to additional stability requirement to the converter (additional protection device, etc.);
4) Regenerative breaking is more complex for implementation. Resistive dynamic breaking systems are often got used with AC drives.

The basic principle of field oriented control is based on connection with complex coordinate vectors. The main target is to define the flux producing current and torque producing current by decoupling field components.

The main parameters can be described by the space phasors in the three-phase coordinate system:

$$\overline{u} = \frac{2}{3}(u_{\mathrm{a}} + \overline{a}u_{\mathrm{b}} + \overline{a}^2 u_{\mathrm{c}}) \tag{2}$$

Instantaneous values can be calculated through the voltage and flux vectors of the motor:

$$\overline{u}_{\mathrm{s}} = \overline{i}_{\mathrm{s}} R_{\mathrm{s}} + \frac{d\overline{\psi}_{\mathrm{s}}}{dt} \tag{3}$$

$$\overline{u}_{\mathrm{r}} = \overline{i}_{\mathrm{r}} R_{\mathrm{R}} + \frac{d\overline{\psi}_{\mathrm{r}}}{dt} \tag{4}$$

$$\overline{\psi}_{\mathrm{s}} = \overline{i}_{\mathrm{s}} L_{\mathrm{s}} + \overline{i}_{\mathrm{r}} e^{j\theta} L_{\mathrm{m}} \tag{5}$$

$$\overline{\psi}_{\mathrm{r}} = \overline{i}_{\mathrm{r}} L_{\mathrm{r}} + \overline{i}_{\mathrm{s}} e^{-j\theta} L_{\mathrm{m}} \tag{6}$$

where $\overline{u}_{\mathrm{s}}$ and $\overline{u}_{\mathrm{r}}$ are the stator and the rotor voltage, respectively.$\overline{i}_{\mathrm{s}}$ and $\overline{i}_{\mathrm{r}}$ are the stator and rotor currents, respectively. R_{s} and R_{R} are the stator and rotor resistances respectively. $\overline{\psi}_{\mathrm{s}}$ and $\overline{\psi}_{\mathrm{r}}$ are the stator and rotor flux linkages, respectively. L_{s}, L_{r} and L_{m} are the stator coil, rotor coil, and magnetizing inductances, respectively.

The electromagnetic torque can be defined through the equation

$$\overline{T}_{\mathrm{e}} = \frac{3}{2}\overline{\psi}_{\mathrm{r}}\overline{i}_{r}\sin\varphi = -\frac{3}{2}\overline{\psi}_{\mathrm{r}} \times \overline{i}_{\mathrm{r}} \tag{7}$$

where $\overline{T}_{\mathrm{e}}$ is the electromagnetic torque.

The main target is the transformation into the coordinate system consisted of two phases. After transformation it is possible to estimate control signals, for example of each is shown in Fig. 2.9:

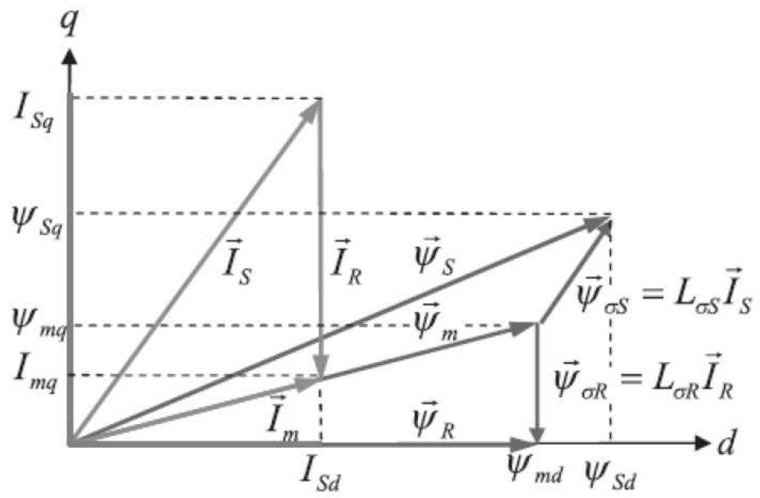

Fig. 2.9 Vector diagram of induction motor with ψ_R=const

To perform a system conversion, the transformation matrix should be used. The transformation results in the imaginary and real part of space-vectors. After the transformation into a two-phase system and should be implemented in α-β coordinate system:

$$\begin{bmatrix} u_\alpha \\ u_\beta \end{bmatrix} = \frac{2}{3}\begin{bmatrix} 1 & -\frac{1}{2} & -\frac{1}{2} \\ 0 & \frac{\sqrt{3}}{2} & -\frac{\sqrt{3}}{2} \end{bmatrix}\begin{bmatrix} u_a \\ u_b \\ u_c \end{bmatrix} \tag{8}$$

It is advisable to perform transformation from two-phase system into a common coordinate system which is rotating *d-q* coordinate system. In the end, the sinusoidal quantities turn to be non-rotating.

$$\begin{bmatrix} u_d \\ u_q \end{bmatrix} = \begin{bmatrix} \cos\rho & \sin\rho \\ -\sin\rho & \cos\rho \end{bmatrix}\begin{bmatrix} u_\alpha \\ u_\beta \end{bmatrix} \tag{9}$$

Rotor flux should be oriented to the *d*-axis of *d-q* rotating system. In this case the rotor flux can be controlled by *d*-component of the current. At the same time *q*-component of the current can control the torque. The torque equation is presented below:

$$\overline{T} = \frac{3}{2}p\psi_{rd}i_{sq} \tag{10}$$

3 Thermodynamics and capacity control

3.1 Basics of screw compressor thermodynamics

The rotary screw compressor is a positive displacement machine like the other reciprocating compressors. Because of that it is possible to apply calculation for the gas compressing from the atmospheric pressure to the required pressure for both reciprocating and rotary screw compressors.

Construction of any compression system requires using the key parameters needed for calculation the compressor performance. [3]

The energy balance in the case of compressors can be written in differential form:

$$dy + dq = dh + d\frac{c^2}{2} + gdz \tag{11}$$

where y is the compressor work input for the specific mass, q the heat flow into the compressor from the outside environment, h the gas enthalpy, c the gas velocity, g the gravitational acceleration, and z the elevation.

The sum of the work input connected to the mass and the heat flow inside the compressor is equal to the change of various parameters such as enthalpy, kinetic energy and head difference. [6] In most cases, kinetic energy change alongside with the static head contribution is considered to be negligible and the compressor is assumed to be isolated from the environment. This assumption leads to the equation

$$dy = dh \tag{12}$$

The change of the enthalpy is

$$dh = \frac{dp}{\rho} \tag{13}$$

where ρ is the gas density and p the gas pressure.

The actual work can be calculated by dividing the referenced work with the efficiency

$$W = \frac{y}{\eta} \tag{14}$$

where W is the actual compressor work, y the work input to the compressor related with the mass, and η the efficiency.

Three different processes are usually used for performance calculation: isentropic compression (entropy is constant), isothermal compression (temperature is constant), polytrophic compression (efficiency is constant).

In the case, when the heat is continuously being removed from the gas, the isothermal compression can be considered. For isothermal compression is valid the following equations

$$pv = \text{const} \tag{15}$$

$$\rho = \rho_1 \left(\frac{p}{p_1}\right) \tag{16}$$

$$y_t = \frac{Z_1 R T_1}{M} \ln\left(\frac{p}{p_1}\right) \tag{17}$$

where y_{t} is the isothermal head.

In most cases, compressors are neither isentropic nor isothermal that leading to the polytropic compression, which can be described by the following equations

$$pv^{n_v} = const \tag{18}$$

$$\rho = \rho_1 \left(\frac{p}{p_1}\right)^{1/n_{\mathrm{v}}} \tag{19}$$

$$y_p = \frac{Z_1 R T_1}{M} \frac{n_{\mathrm{v}}}{n_{\mathrm{v}} - 1} \left[\left(\frac{p}{p_1}\right)^{\frac{n_{\mathrm{v}}}{n_{\mathrm{v}} - 1}} - 1\right] \tag{20}$$

where n_{v} and y_{p} are the volume exponent for the polytropic process and the polytropic head respectively.

The most important parameters for the air system are isentropic efficiency, isentropic work of the compression process and volumetric efficiency. [5] For the isentropic process following equations are valid

$$pv^k = const \tag{21}$$

$$\rho = \rho_1 \left(\frac{p}{p_1}\right)^{1/k} \tag{22}$$

The isentropic work can be estimated through the gas enthalpy or the temperature change

$$w_s = h_{2s} - h_1 = c_p(T_{2s} - T_1) \tag{23}$$

where h_1, T_1, h_{2s} T_{2s}, c_p, are the initial gas enthalpy, the initial temperature, the final gas enthalpy, the final temperature and the heat of the gas during the constant pressure respectively.

Consequently it is possible to derive the isentropic work of the compression process

$$w_s = c_p T_1 \left[\left(\frac{p}{p_1} \right)^{\frac{k-1}{k}} - 1 \right] = RT_1 \frac{k}{k-1} \left[\left(\frac{p}{p_1} \right)^{\frac{k-1}{k}} - 1 \right] \tag{24}$$

where k is the ratio, which is equal to ${}^{C_p}/_{C_v}$.

The theoretical work of the isentropic process can be calculated by dividing the isentropic work and the isentropic efficiency

$$w_s = \frac{c_p T_1 \left[\left(\frac{p}{p_1} \right)^{\frac{k-1}{k}} - 1 \right]}{\eta_{is}} \tag{25}$$

where η_{is} is the isentropic efficiency.

Estimation of the theoretical input power of the compressor can be derived through the mass flow rate and the theoretical work

$$W_0 = \frac{\dot{m} c_p T_1 \left[\left(\frac{p}{p_1} \right)^{\frac{k-1}{k}} - 1 \right]}{\eta_{is}} \tag{26}$$

where W_0 is the theoretical power input.

The isentropic head can be found as follows

$$y_i = \frac{Z_1 R_g T_a}{M} \frac{k}{k-1} \left[\left(\frac{p}{p_1} \right)^{\frac{k-1}{k}} - 1 \right] \tag{27}$$

where v is the molar volume, Z_1 the compressibility, T_a the absolute temperature, M the gas molar weight, and y_i the isentropic head.

Isentropic efficiency is the actual ratio of power delivered to the end-user over the full power used by the compressor. The isentropic efficiency can be delivered as follows

$$\eta_{\mathrm{is}} = \frac{\dot{m}c_{\mathrm{p}}T_1\left[\left(\frac{p}{p_1}\right)^{\frac{k-1}{k}} - 1\right]}{W_0} \tag{28}$$

However, defining the mass flow rate in the case of screw compressors can be a hard task due to sophisticated rotor structure. The calculation of the mass flow rate also differs from various rotor types. In addition, the mass flow rate is affected by screw compressor leakages.

If it is possible to estimate the volume of the gas, trapped inside the chamber, then it is become clear that the total volume of the gas can derived though the equation

$$Q = nq_{\mathrm{v}} \tag{29}$$

where Q is the total volume flow, n the amount of the revolutions per minute, q the amount of the gas (volume) transferred through one revolution.

There are different kinds of screw compressor leakages that were already described before. Therefore, the real volume rate is decreased by the volume flow created by the leakages

$$Q_r = Q - Q_{\mathrm{l}} \tag{30}$$

where Q_{r} is the real volume rate, Q_{l} the volume rate formed by leakages though slippages.

Consequently, volumetric efficiency can be calculated

$$\eta_{\mathrm{v}} = \frac{Q_{\mathrm{r}}}{Q} = 1 - \frac{Q_{\mathrm{r}}}{Q_{\mathrm{l}}} \tag{31}$$

The volumetric efficiency depends not only on the compressor characteristics (compression ratio, compressor's speed and construction, etc), but also on the gas characteristics. High compression rate creates high pressure deferential between compressor parts, and thereby significantly increases the leakage flow. In addition, low compression speed increases time for gas to leak through clearances.

The oil-flooded compressor has less leakage rate, since the oil can be determined as a sealing component for clearances. Volumetric efficiency does not increase directly the power used by the compressor, but at the same time it affects the isentropic efficiency by decreasing it through the mass flow reduction. Consequently, both the volumetric and isentropic efficiencies affect the performance of the compressor.

Mechanical efficiency also takes into account any gearboxes, additional pumps for the oil-feed system, bearings power, etc. It is might also be important value in addition to isentropic and volumetric efficiencies.

3.2 Introduction to screw compressor model

The complexity of the rotary screw compressor and lack of the literature provides also a challenge in developing of a reliable compressor model. In addition, like it was described in chapter 1, the compressor system involves not only the actual compressor, but aftercoolers, lubricant separators, filters, distribution system, etc. It led to the requirement to make major assumptions and simplifications. In addition, the model is focused mostly on the data, which can be received from the datasheets and from the test operations.

From the control point of view, it is also recommended to have as small amount of components in the system as possible. A lot of components do not only make calculations rather overwhelming, but also add disturbances, delay in the calculations, etc.

It is assumed, that two key components of the system are the compressor itself and the load (gas tank). Piping might also be taken into account as a component that creates disturbances and leakages.

Mass of the compressed air can be estimated as the function of the air density and volume

$$m_{\mathrm{air}} = \rho_{\mathrm{air}} V_{\mathrm{air}} = \frac{p_{\mathrm{air}} V_{\mathrm{air}}}{R_{\mathrm{air}} T_{\mathrm{air}}} \tag{32}$$

The compressor wastes energy to compress the air from the atmospheric pressure to the required system pressure. The mass flow affects also the compressor performance through the influence on the pressure behavior and the work of the compression.

It is assumed that the air temperature alongside with the gas constant both stay constant. Therefore, the system mass change becomes the function of the volume and pressure.

The compression work (W) consists of two main parts: the compressor work to increase the gas pressure (W_{o}) and mechanical work (W_{m}) which is assume to be 10% of total compression work to overcome the friction and to rotate the both rotors

$$W(t) = W_{\mathrm{O}}(t) + W_{\mathrm{m}} \tag{33}$$

Isentropic efficiency can be described as a division of reversible and irreversible work. Polytrophic work is assumed to be reversible, while the isentropic work is taken into account by being irreversible.

Mechanical work is being hard to estimate, and therefore in the model according to the different references [1] [6] it stays constant during the whole process and equal to 10% of non-mechanical work. In addition, it depends on the rotors speed and mechanical friction, which are hard to estimate during the real process. Mechanical work relies also on the construction of the compressor itself. There are additional energy that will be consumed by work panels and other electronics that should also be considered in the work calculation. However, it is much easier to calculate the mechanical work and electricity consumed by electronic devices through the field tests.

According to various references ([27], [28]) the power of any positive displacement compressor can be calculated through the adiabatic power equation:

$$P = \frac{k}{k-1} p_1 N Q \left[\left(\frac{p_1}{p_2} \right)^{\frac{k}{k-1}} - 1 \right] \tag{34}$$

where N – number of stages, which is equal to one for screw compressors. [46]

The main equations for the describing the compressor were presented in chapter 3. All the parameters are not calculated in the model, but they are the key components in understanding of the thermodynamics based on the calculation of the system efficiency.

3.3 Capacity control

In air compression systems, where load varies due to application requirements or other factors, such as atmospheric pressure, lightning, air moisture content, control failures, etc. The capacity control is targeted to provide the required system energy performance. [20]

Even though, there are many different ways to implement capacity control for compression systems, it will reduce power requirement, decrease compressor cycling and starting load, manage optimum oil flow of the system and dehumidification if it is required.

In theory, the most optimal way in capacity control is the on-off type, when the electric drive is completely turned off and the required system pressure inside the gas tank is reached. [19] Therefore, there are two main pressure values for the control system based on the information from the pressure sensor: when the pressure reaches the high critical value or low critical value establishing the compressor operating range. Therefore, compressors work until the critical pressure value is reached. After that, it is turned off completely until the critically low-pressure value is reached and it starts filling the gas tank with atmospheric gas. After that the cycle will be repeated.

However, in terms of real compressor systems the approach cannot be considered as optimal. The main disadvantage is the limited operating range when precise output parameters are required. Additionally, the volume of the gas tank should be high enough to provide required duration of pressure output with possibility for electric drive to be disabled for long duration.[20] Moreover, the electric drive starts can create spikes inside the electric grid. Amount of electric drive starts is also quite limited since the excessive motor switching can lead to the motor overheating. Bearings can also be damaged since when start-up speed is low the bearing lubricant has non-optimal viscosity. It can reduce the life-span both of the compressors parts and electrical engine. And last but not the least, the oil return during discontinuous operations cannot be sufficient for effective compressor operation.

All these drawback impacts can be significantly reduced in case with large compression systems where several compressors can be used. However, in terms of laboratory test equipment the system scale is highly limited.

Another capacity control method is quite popular in compression systems. Load-unload mode ideally has lower efficiency in comparison with on-off method, but it can be used for small-scale compressors. [20]

When the pressure reaches its desired value, electrics is not turned off completely, but instead the discharge valve is opened to release the gas into the atmosphere. [20] In this case, the compressor is working only against the atmospheric pressure and work to compress the gas itself is not required. There are only mechanical losses left in the system. The method requires fewer starts and stops significantly reducing the drawbacks of the on-off model. However, during the blow down time, the discharge valve should be opened slowly in order to prevent the excessive stress to the compressor parts.

The laboratory equipment also uses this method mostly to prevent the critical pressure inside the gas tank to be reached.

The Simulink model implements capacity control through the limitation of the highest pressure value. Also during the cycle when the pressure reaches its critical value, the electrical motor works only against the mechanical losses. Blow down time though is being set by the manufacturer meaning that discharge valve is opened within some initially adjusted delay range. In addition, it means, that power consumption is also decreased with the some delay leading to the fact that mechanical losses will be applied alone only after the blow down time will be surpassed.

Suction throttle control is also a wide spread control method. The idea behind the method is the mass flow control through the inlet. The mass control device is usually the butterfly valve that limits the mass flow if the pressure inside the gas tank reaches the required value. When the valve is closed, the mass flow is decreased thereby decreasing the power. The compression work is also decreased. However, suction throttling method does not allow the gas to completely stop reaching the gas tank. It is related to the fact that valve cannot be not fully closed. [19]

Therefore, most industrial facilities combine load-unload and suction throttling methods. When the high enough pressure is reached inside the gas tank, the valve is closing, working only in a pure modulation method. If the conditions when it is impossible to close the valve anymore are fulfilled, the compressor control system start using the load-unload mode in order to decrease the pressure.

4 Modeling of rotary screw compressor

4.1 Real system preliminary analysis

In the thesis, the screw compressor is analyzed with the parameters, which are presented in Table 1 [22].

Table 1

The screw compressor parameters

Model	Rated motor power, kW	Nominal speed, RPM	Mass flow, m^3/min	Working pressure, bar
Kaeser SM9	5,5	2910	0,90	7,5

The instrumentation diagram is presented in Fig. 4.1.

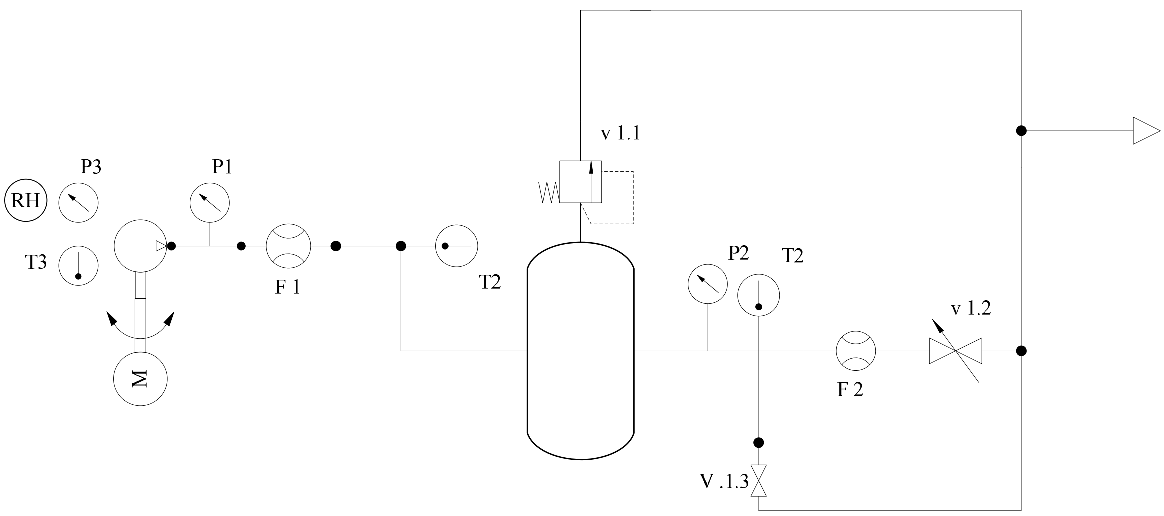

Fig. 4.1 Instrumentation diagram and pipes of laboratory equipment

The system consists of a motor, a frequency converter, a gas reservoir, pressure and temperature sensors, inlet and outlet valves, and flow sensors. The system working capabilities were estimated during the tests, which were done for different motor speeds. The data was collected using pressure, current and power sensors. The tests were done for estimation of time required to fill the compressor. The inlet valve during the tests was completely open meaning that mass flow was not limited. The connection between the motor and compressor is implemented via the belt drive.

The pressure tank behaves like an integrator since it storages the air. However, its volume is limited and the limitation is implemented through the capacity control valve. The pressure should stay constant after the point when it reaches the required value inside the gas storage.

The system does not utilize the speed sensor and control algorithm relies on the data collected from the pressure sensors. Inside the motor, the current sensor is installed not only for safety measures, but also it participates in the calculation of the power.

The comparison of the pressure distribution over the time is shown in Fig. 4.2.

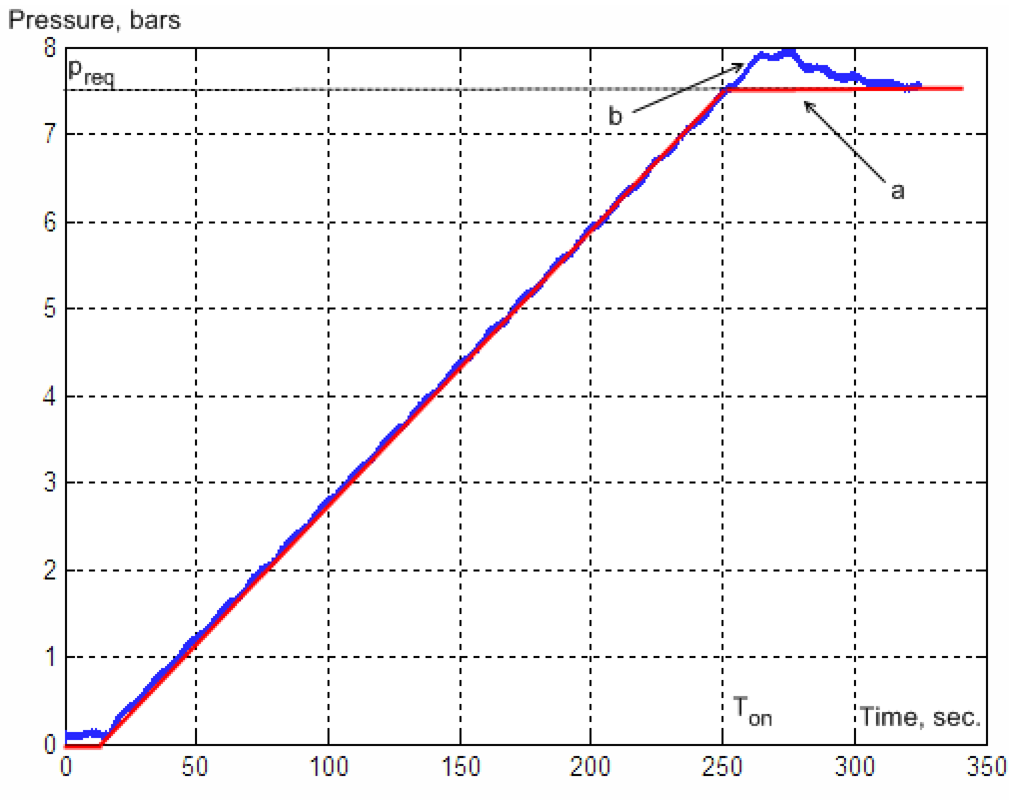

Fig. 4.2 Pressure distribution over the time: a – ideal pressure distribution, b – pressure distribution in laboratory tests for motor speed of 2044 RPM

In both cases, the pressure reaches the required value p_{req} in the time T_{on}. Analysis of the pressure graphs shows that the transient overshoot is around 4% ÷ 4.3%. It is connected the fact, that when the critical pressure of 8 bars inside the gas tank is reached, the outlet valve is opened in order to decrease the pressure to nominal values.

The time required to fill the gas tank is almost proportional to the motor speed. Another difference is that the real pressure has fluctuations with almost sinusoidal distribution. The amplitude of fluctuations is around 0.1 bars. The frequency of fluctuations stays constant for different speeds even though it should vary. The fluctuations most likely are caused by pressure sensor measurement error, since they occur even when machine is completely stopped.

Data of the power distribution over the time collected from the laboratory test has shown that from pressure value of 4.4 bars there is a power consumption rise with value of 1.1 kW (Fig. 4.3).

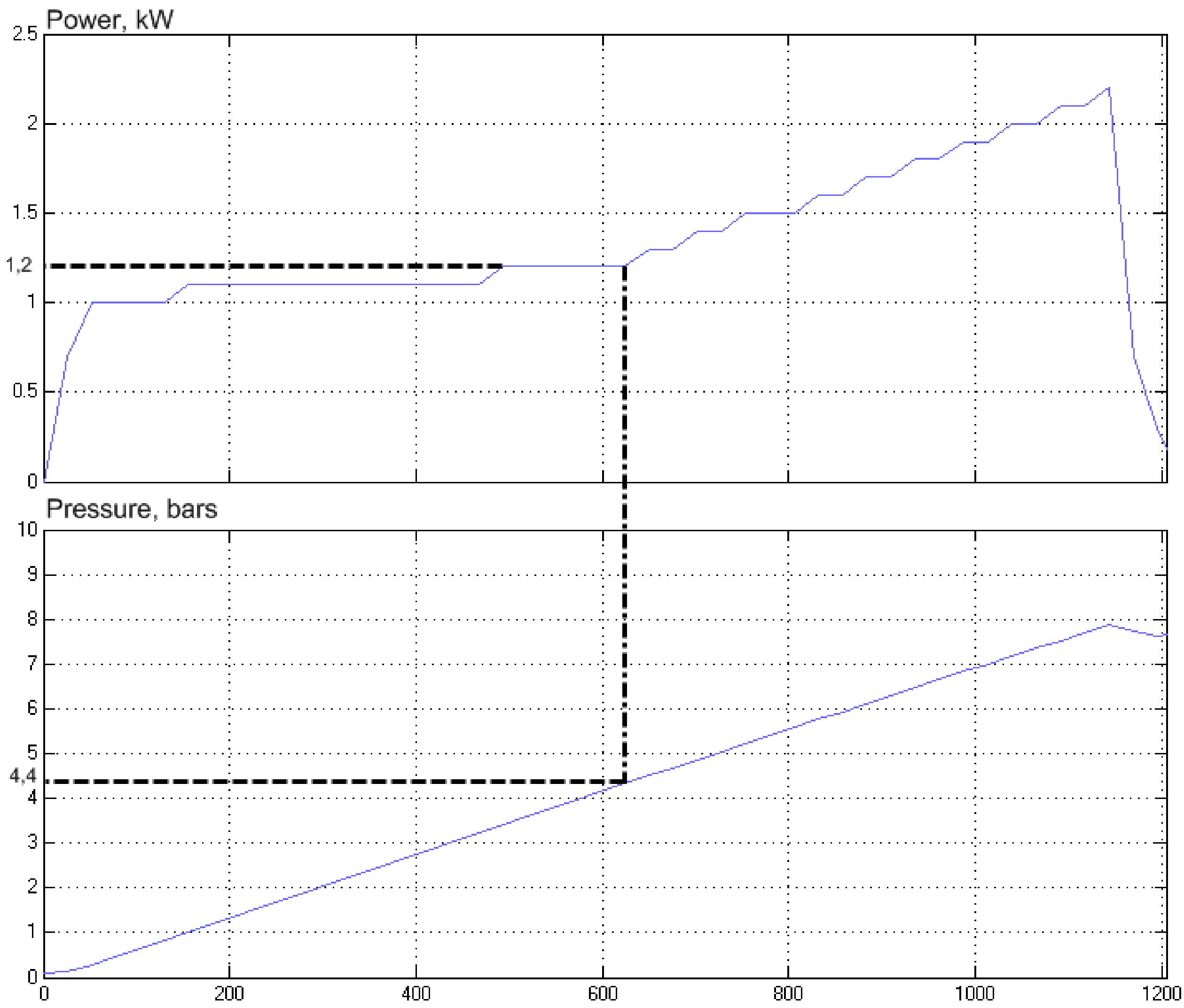

Fig. 4.3 power and pressure distributions over the time for speed 584 RPM

Data connected with different speeds has provided the same results: after pressure of 4.4 bars, the compressor is increasing its consumption on the 1.1 kW. The rise is caused by the oil-pump where the mass flow of the lubricant is enhancing proportionally to the pressure increase.

When the air tank is filled with air at the required pressure, the capacity control opens the valve and from this point the compressor operates only against the atmospheric pressure. The power decreases to the 30%–40% of nominal values.

4.2 Control system of laboratory equipment

According to the Sigma 2 control manual [21], there are different states and control modes implemented for the compressor.

In the «stop» state, the machine is still connected to the power supply, but it is switched off. In the «ready» state, the motor is stopped and the inlet valve is closed; the oil separator is disconnected from the distribution network and machine starts operating as soon as the minimum pressure value is reached. In the «load» state, the machine is connected to the grid, the inlet valve is opened, and the machine delivers the required air to the gas storage. In the «idle» state, the compressor is still working, but with the low load and therefore low power consumption; the inlet valve is closed, the oil separator is disconnected from the distribution grid. Small amount of air still circulates through the hole in the inlet valve to the discharge port and then again back to the inlet port via the venting line.

Dual operating mode has two working states, which are «idle» and «load». The compressor working preset has minimum and maximum operating pressure values. When the maximum pressure is reached, the machine is set to be in the idle state. When the preset idle state time is passed, the machine moves to the «ready» state. Idle time is based on the compressor starting frequency. The short idle time allows the motor to be stopped and started more frequently.

In the «quadro» mode, the machine is switched to from the «load» to the «ready» state after the consumption of low compressed air. In the periods with high compressed air consumption, the machine is turned from «load» to «ready» through the «idle» state. To operate under these conditions, the controller requires operating and standstill (idle) times.

The «vario» mode is the type of the «dual» mode. The main difference lays in the automatic control of the idling time for higher or lower machine frequency compensation. The continuous mode utilizes the same approach as the «dual» mode. However, the control method does not use «ready» state.

Dynamic mode is based on the motor temperature estimation. It switches the motor from the «load» to the «ready» state at the low drive motor temperature and from the «load» to the «ready» via rhe «idle» state at high motor temperatures. Modulating control implements additional mechanical regulation through the changing delivery rate within the machine control range. The inlet valve closes proportionally to the required air consumption.

The frequency-controlled drive compares actual pressure to the required pressure and according to the operating mode changes the motor states. For example, if the air consumption increases, the converter increases the motor speed, and thereby increases the delivery rate. If the target pressure is reached, the converters changes speed to the minimum and then proceeds to the state according to the control method.

4.3 Electrical motor model for direct on-line connection

There are different control types at which the AC motor can operate. For the processes that do not require any sophisticated algorithms is still very popular direct on-line startup for the AC motor. Therefore, for processes, such as fans, compressors, etc, especially if they use in long term conditions, direct on-line is still very common. [14]

Basically, to describe the direct on-line mode, the equivalent scheme of the AC motor should be reviewed. Simplified equivalent scheme of the induction motor is presented in Fig. 4.4. It consists of 3-phase rotor winding and 3-phase stator winding. Both of them connected to symmetrical 3-phase voltage sources. Simplification means that there are no eddy current losses, uneven air gap distribution or high order magnetic field harmonics. [24] [10]

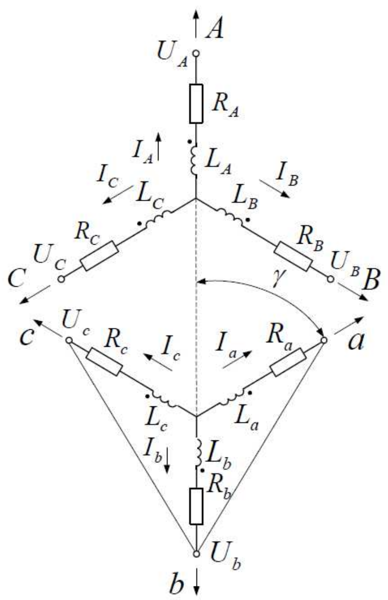

Fig. 4.4 Simplified equivalent circuit of induction motor [24]

Flux linkage equations show that the flux linkage of each winding depends on the current in each phase. All these relations connected through the mutual inductance.

According to the Newton law, the produced torque on the shaft of the machine can be defined according to

$$J\frac{d\overline{\omega}}{dt} = \overline{T} - \overline{T}_{\mathrm{L}} \tag{35}$$

where J, $\overline{\omega}$, $\overline{T}_{\mathrm{L}}$ are the moment of the inertia on the shaft of the motor, the speed of the motor, the load torque respectively.

Electromagnetic torque is equal to

$$T_{\mathrm{e}} = \frac{3}{2}p\overline{\psi_{\mathrm{s}}} \times \overline{i_{\mathrm{s}}} \tag{36}$$

However, in order to utilize these equations in the modeling task, they should be simplified according to the space vector theory. Equations that used for describing the induction motor after utilization:

$$\overline{u_{\mathrm{s}}} = R_{\mathrm{s}}\overline{i_{\mathrm{s}}} + \frac{d\overline{\psi_{\mathrm{s}}}}{dt} + j\omega_k\overline{\psi_{\mathrm{s}}} \tag{37}$$

$$\overline{u_{\mathrm{r}}} = R_{\mathrm{r}}\overline{i_{\mathrm{r}}} + \frac{d\overline{\psi}_{\mathrm{r}}}{dt} + j(\omega_{\mathrm{k}} - p\omega_{\mathrm{m}})\overline{\psi_{\mathrm{r}}} \tag{38}$$

$$\overline{\psi_{\mathrm{s}}} = L_{\mathrm{s}}\overline{i_{\mathrm{s}}} + L_{\mathrm{m}}\overline{i_{\mathrm{r}}} \tag{39}$$

$$\overline{\psi}_{\mathrm{r}} = L_{\mathrm{m}}\bar{i}_{\mathrm{s}} + L_{\mathrm{r}}\overline{i_{\mathrm{r}}} \tag{40}$$

$$T_{\mathrm{e}} = \frac{3}{2}p\overline{\psi_{\mathrm{s}}} \times \bar{i}_{\mathrm{s}} \tag{41}$$

$$J\frac{d\omega}{dt} = T - T_{\mathrm{L}} \tag{42}$$

Considering a squirrel-cage induction motor, it has short circuited rotor and therefore

$$\overline{u_{\mathrm{r}}} = R_{\mathrm{r}}\bar{i}_{\mathrm{r}} + \frac{d\overline{\psi}_{\mathrm{r}}}{dt} + j(\omega_{\mathrm{k}} - p\omega_{\mathrm{m}})\overline{\psi}_r = 0 \tag{43}$$

Then from equations (37) – (42) it is required to exclude both i_r and ψ_s:

$$0 = -k_{\mathrm{R}}R_{\mathrm{R}}i_{\mathrm{s}} + \frac{1}{T_{\mathrm{R}}}\overline{\psi}_{\mathrm{r}} + \frac{d\overline{\psi_{\mathrm{r}}}}{dt} + j(\omega_{\mathrm{k}} - p\omega_{\mathrm{m}})\,\overline{\psi}_{\mathrm{r}} \tag{44}$$

$$T_{\mathrm{e}} = \frac{3}{2}p\overline{\psi_{\mathrm{r}}} \times \bar{i}_{\mathrm{s}} \tag{45}$$

$$J\frac{d\omega}{dt} = T - T_{\mathrm{L}} \tag{46}$$

$$r = (R_{\mathrm{s}} + k_{\mathrm{r}}^{2}R_{\mathrm{r}}) \tag{47}$$

$$L_{\mathrm{s}}^{'} = (L_{\mathrm{s}} - \frac{L_{\mathrm{m}}^{2}}{L_{R}}) \tag{48}$$

$$k_{\mathrm{R}} = \frac{L_{\mathrm{m}}}{L_{\mathrm{R}}} \tag{49}$$

$$T_{\mathrm{R}} = \frac{L_{\mathrm{r}}}{R_{\mathrm{r}}} \tag{50}$$

Next step is to analyze the direct start of the machine at no-load. For simplicity we build a model using a static coordinate system since there is almost no result difference in comparison with the rotating coordinate system model. In this case, α is the real axis and β the imagination axis. Then it is possible to obtain the following equations

$$u_{\mathrm{s\alpha}} = ri_{\mathrm{s\alpha}} + L_{\mathrm{s}}^{'}\frac{di_{\mathrm{s\alpha}}}{dt} - \frac{k_{\mathrm{r}}}{T_{\mathrm{r}}}\psi_{\mathrm{r\alpha}} - k_{\mathrm{R}}p\omega_{\mathrm{m}}\psi_{\mathrm{r\beta}} \tag{51}$$

$$u_{s\mathrm{\beta}} = ri_{\mathrm{s\beta}} + L_{\mathrm{s}}^{'}\frac{di_{\mathrm{s\beta}}}{dt} - \frac{k_{\mathrm{r}}}{T_{\mathrm{r}}}\psi_{\mathrm{r\beta}} + k_{\mathrm{R}}p\omega_{\mathrm{m}}\psi_{\mathrm{r\alpha}} \tag{52}$$

$$0 = -k_{\mathrm{R}}R_{\mathrm{R}}i_{s\alpha} + \frac{1}{T_{R}}\psi_{\mathrm{r\alpha}} + \frac{d\psi_{\mathrm{r\alpha}}}{dt} + p\omega_{\mathrm{m}}\,\psi_{\mathrm{r\beta}} \tag{53}$$

$$0 = -k_{\mathrm{R}}R_{\mathrm{R}}i_{\mathrm{s\beta}} + \frac{1}{T_{\mathrm{R}}}\psi_{\mathrm{r\beta}} + \frac{d\psi_{\mathrm{r\beta}}}{dt} - p\omega_{\mathrm{m}}\,\psi_{\mathrm{r\alpha}} \tag{54}$$

$$T_{\mathrm{e}} = \frac{3}{2}pk_{\mathrm{R}}(\psi_{\mathrm{r\alpha}}i_{\mathrm{s\beta}} - \psi_{\mathrm{r\beta}}i_{\mathrm{s\alpha}}) \tag{55}$$

$$J\frac{d\omega}{dt} = T - T_{\mathrm{L}} \tag{56}$$

Using Laplace transform:

$$u_{s\alpha} = r(1 + T_s^{'}s)i_{s\alpha} - \frac{k_r}{T_r}\psi_{r\alpha} - k_R p\omega_m \psi_{r\beta} \tag{57}$$

$$u_{s\beta} = r(1 + T_s^{'}s)i_{s\beta} - \frac{k_r}{T_r}\psi_{r\beta} + k_R p\omega_m \psi_{r\alpha} \tag{58}$$

$$0 = -k_R R_R i_{s\alpha} + \frac{1}{T_R}(1 + T_s^{'}s)\psi_{r\alpha} + p\omega_m\, \psi_{r\beta} \tag{59}$$

$$0 = -k_R R_R i_{s\beta} + \frac{1}{T_R}(1 + T_s^{'}s)\psi_{r\beta} - p\omega_m\, \psi_{r\alpha} \tag{60}$$

$$T_e = \frac{3}{2}pk_R(\psi_{r\alpha}i_{s\beta} - \psi_{r\beta}i_{s\alpha}) \tag{61}$$

$$Js\omega = T - T_L \tag{62}$$

$$T_s^{'} = \frac{L_s^{'}}{r} \tag{63}$$

According to equations (57) - (63), it can be built a common view of the induction motor model in *α-β* coordinate system. The motor part of the rotary screw compressor is shown in Fig. 4.5.

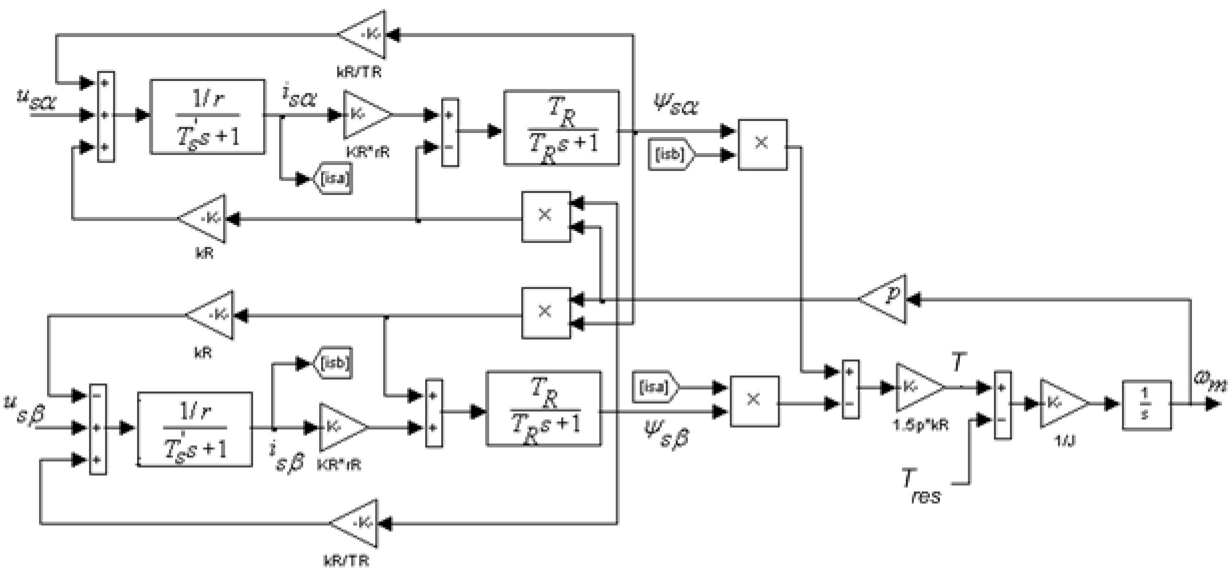

Fig. 4.5 Direct on-line model for electrical machine [10]

Unfortunately, the compressor manufacturer did not provide the exact motor parameters. Therefore the equivalent motor parameters are calculated based on the parameters from Kravchik's catalog of the induction motors [18]. The calculation of parameters is carried out according to the approach based only on the utilization of information got from the nameplate data [16].

4.4 Compressor and gas storage model

The rotary screw compressor model is presented in Fig. 4.6.

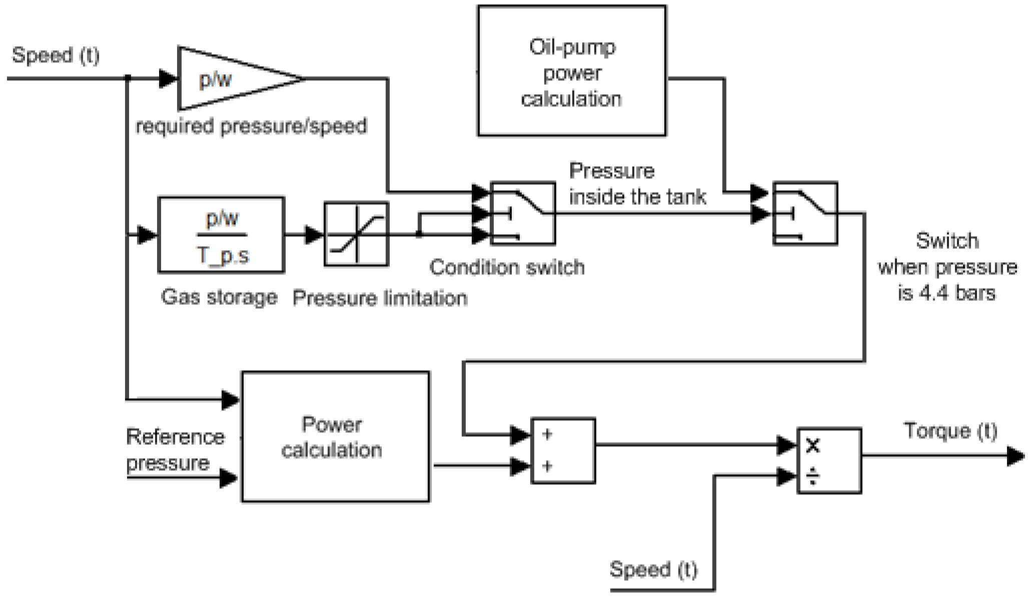

Fig. 4.6 Compressor model

The main task of the model is to calculate the load, which the compressor is producing. Model consists of two main parts, which are the compressor itself and the gas storage. The compressor is presented in the model as a device that converts the speed from the motor to the required pressure. It will be presented as a gain divided by a motor speed meaning that the maximum output pressure is reached, when the motor is fully accelerated to the referenced speed.

The gas tank is implemented through the integrator with the pressure limitation. The compressor's time constant defines the time, in which the gas reservoir is completely filled with the pressured air. The condition switch realizes the pressure checking. When the required pressure is reached, it switches the model to the constant pressure production.

The compressor creates the load, which is applied to the motor. The load can be estimated through the adiabatic power equation (34), which is calculated in the «Power calculation» block. Then power is divided by the motor speed to get the required resistive torque.

Through the various laboratory tests, the mechanical power losses are estimated that are being around 10% of the compressor load.

To build a model, there are required parameters which are presented in Table 1. The first parameter is the reference pressure. It is assumed to be a limitation of the system pressure output and also required for torque production estimation. The second parameter is mass flow. Mass flow varies according to the motor rotational speed: the higher the motor speed the higher total mass flow of the system. With the maximum rotational speed, the mass flow value is according to the nameplate data.

Pressure fluctuations are estimated according to the laboratory tests. The model follows ideas of the isentropic compression. It means that the model does not receive external heating or cooling. Consequently, the temperature does not participate in the compressor model.

4.5 Direct on-line start up simulation

First, there is implemented the direct on-line start up without any control. It means that the motor is connected to the electrical grid directly. There are no frequency converters nor flow control devices. Curves for torque, speed, current and power are presented in Fig. 4.7.

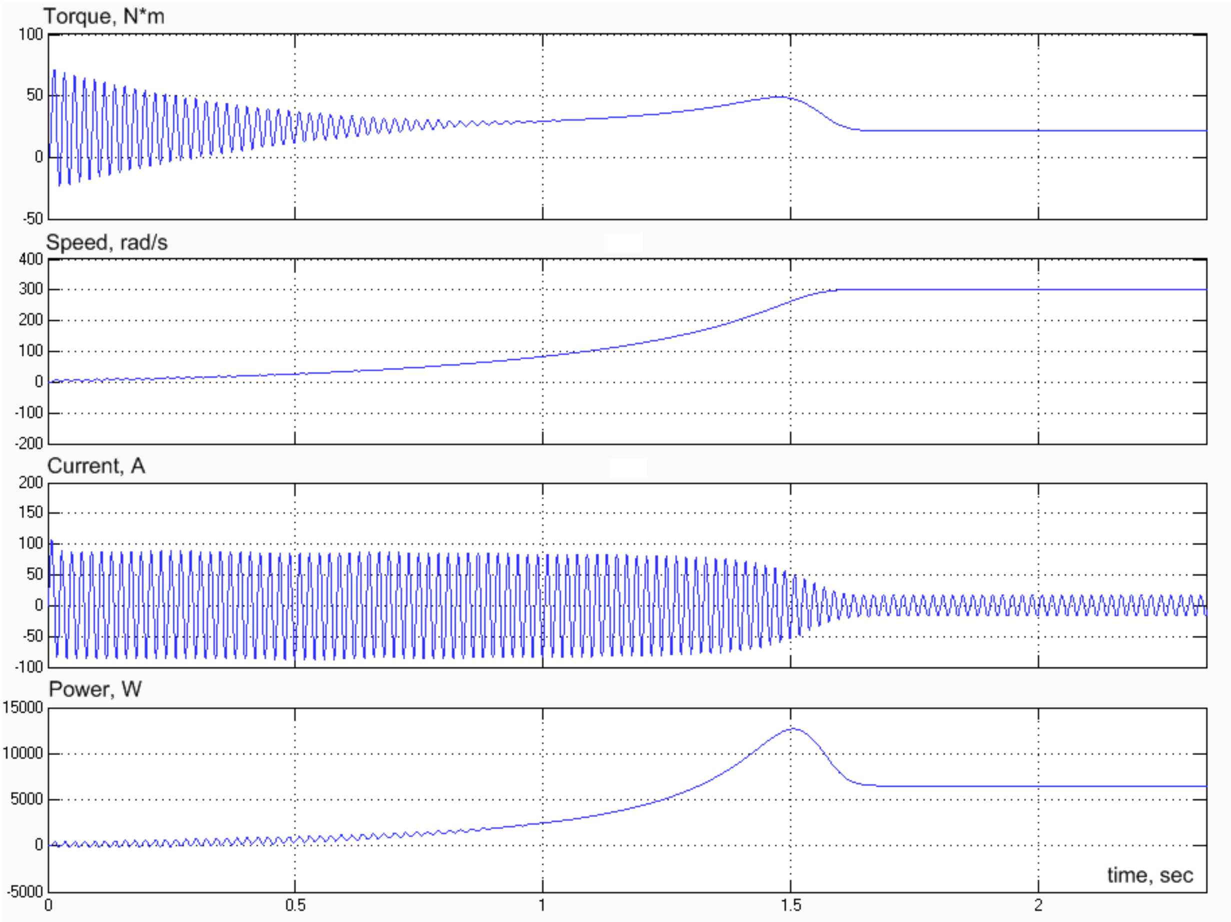

Fig. 4.7 Torque, speed, current and power during the start-up (transients)

Time required for the motor for complete acceleration is around 1.65 seconds. During that time, the total machine power is three times higher than the nominal value. It is clear that if it is needed to stop and start machine frequently, the motor life-span will be significantly reduced due to overheating.

In the established mode (Fig. 4.8), all the motor curves are close to nominal. Basically, if there is no requirement for the compressor to start and to stop often, and if there is air consumers with constant air consumption, the direct on-line start-up without any control can be considered as satisfactory.

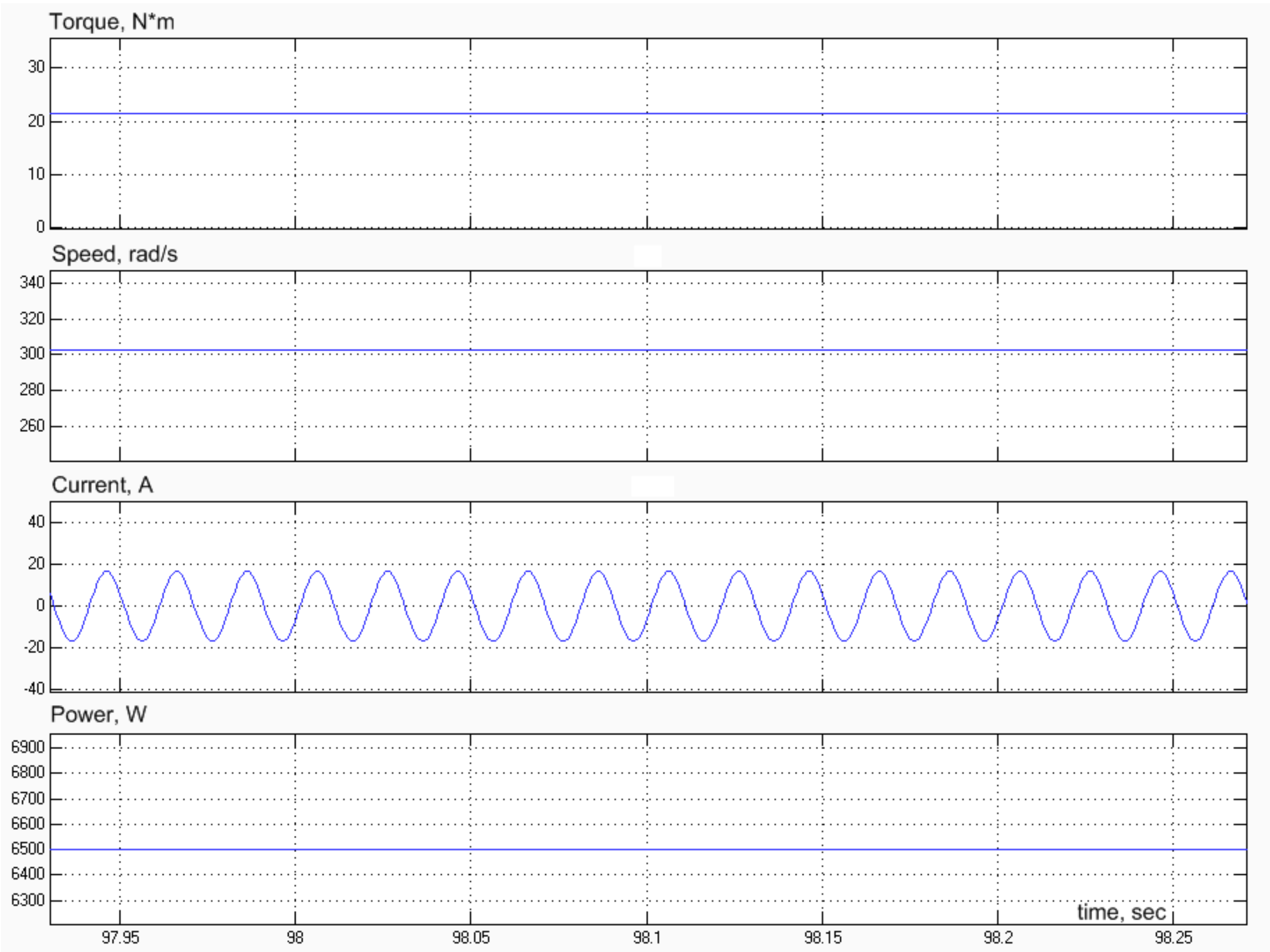

Fig. 4.8 Torque, speed, current and power in the established mode

In the Simulink model with direct on-line start, it is assumed that there is constant pressure consumption, meaning that power after the transients is constant and with values close to nominal (based on the nameplate data).

When the pressure reaches the desired value, the motor in ideal case should work in no-load mode, since the inlet valve is partly or completely closed. Some air still circulates through the pipes. The laboratory equipment though works at 36% of the nominal load.

The pressure comparison for both the simulation and real cases is shown in Fig. 4.9. It is clear that both distributions are similar. The main difference is the transient overshoot for the graph of the laboratory equipment.

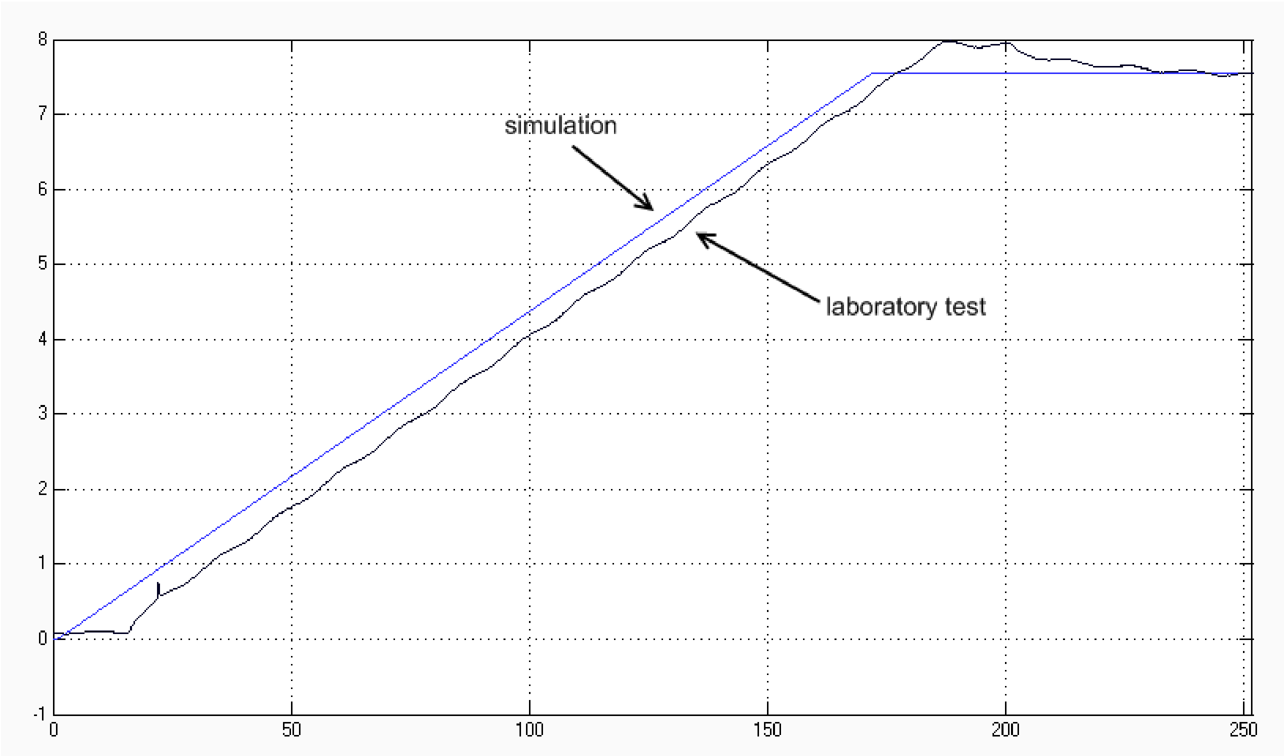

Fig. 4.9 Pressure distribution over the time

The Simulink model of the compressor is shown in Fig. 4.10. In the case of direct on-line connection, there is no flow control meaning that additional resistive torque is equal to 0 and the flow rate is equal to 100%.

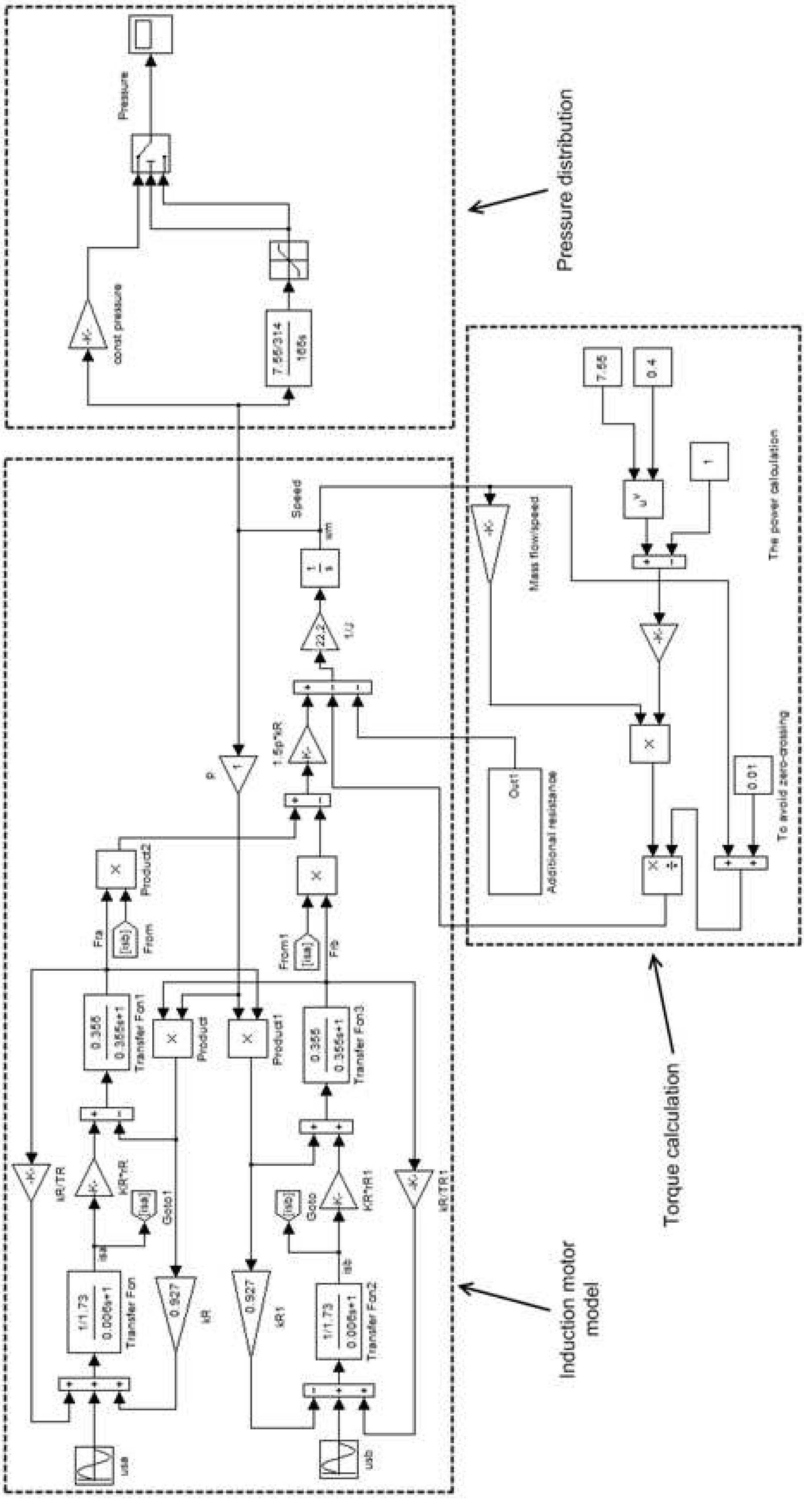

Fig. 4.10 Model for direct on-line induction motor simulation in Matlab Simulink

4.6 Throttling simulation

Capacity control was described in chapter 3. In the model, the capacity control is simulated through the throttling. Throttling is generally implemented via controlling open angle of the inlet valve, which allows changing the mass flow rate of the air. Ideally, the characteristics should be depended proportionally to the open angle of the valve. However, it is nonlinear due to the pressure drop caused by the change of the temperature of the air when it reaches the valve. The temperature change alongside with friction creates the pressure drop in the valve, which is affecting the total power.

The estimation of the pressure drop should include a lot of parameters, such as a diameter of the pipes, the type of the valve, and the air temperature exactly after the valve. All these parameters in most cases cannot be checked, and therefore in reality and in the simulation is used another method. Studies of the throttling applied the following graph of power consumption distribution over the mass flow (Fig. 4.11)

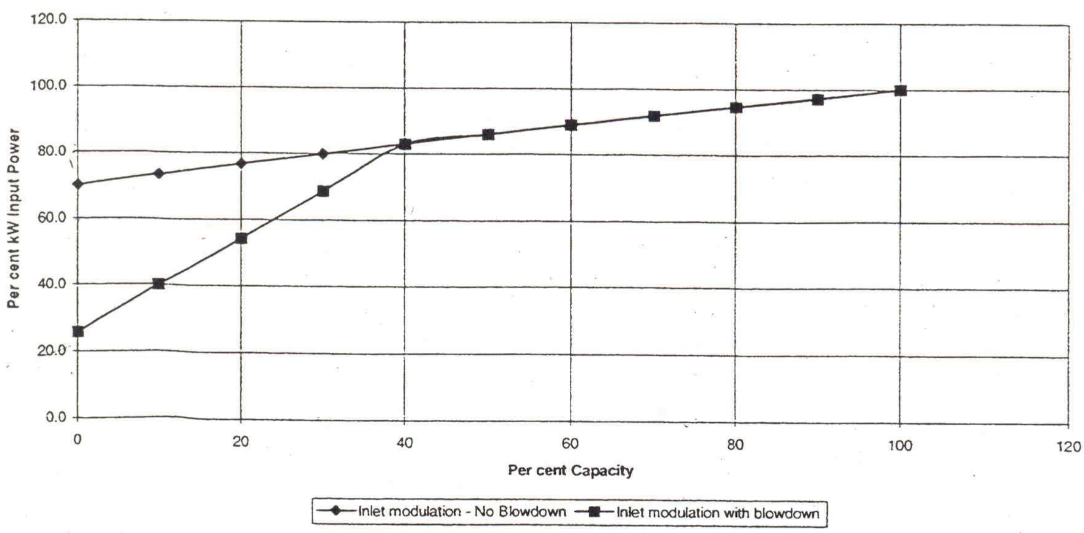

Fig. 4.11 Power distribution over the mass flow capacity for inlet modulation (throttling) [25]

In the case of inlet modulation without blowdown the lowest power requirement is approximately 70% even if the mass flow is low. This is caused by the fact that the compressor should overpass the system total pressure. Without cycling the characteristics between 0% and 100% is relatively linear for different conditions and compressor manufacturers.

The following equation defines the power distribution over the capacity in percents [26]:

$$P_{\%} = 68\% + 0{,}32C_{\%} \tag{64}$$

where $P_{\%}$ is the power in percents and $C_{\%}$ the capacity in percents.

According to (65), it is possible to modify the torque calculation part of the model.

In Fig. 4.12, there are presented curves during the transients in case of capacity control, when flow was reduced on 50%. It is clear that during the first 0.8 seconds, the electrical machine experiences heavy overload. So, it is still not recommended doing the motor startup frequently since it can cause damage to the electrical motor or significantly reduce its characteristics.

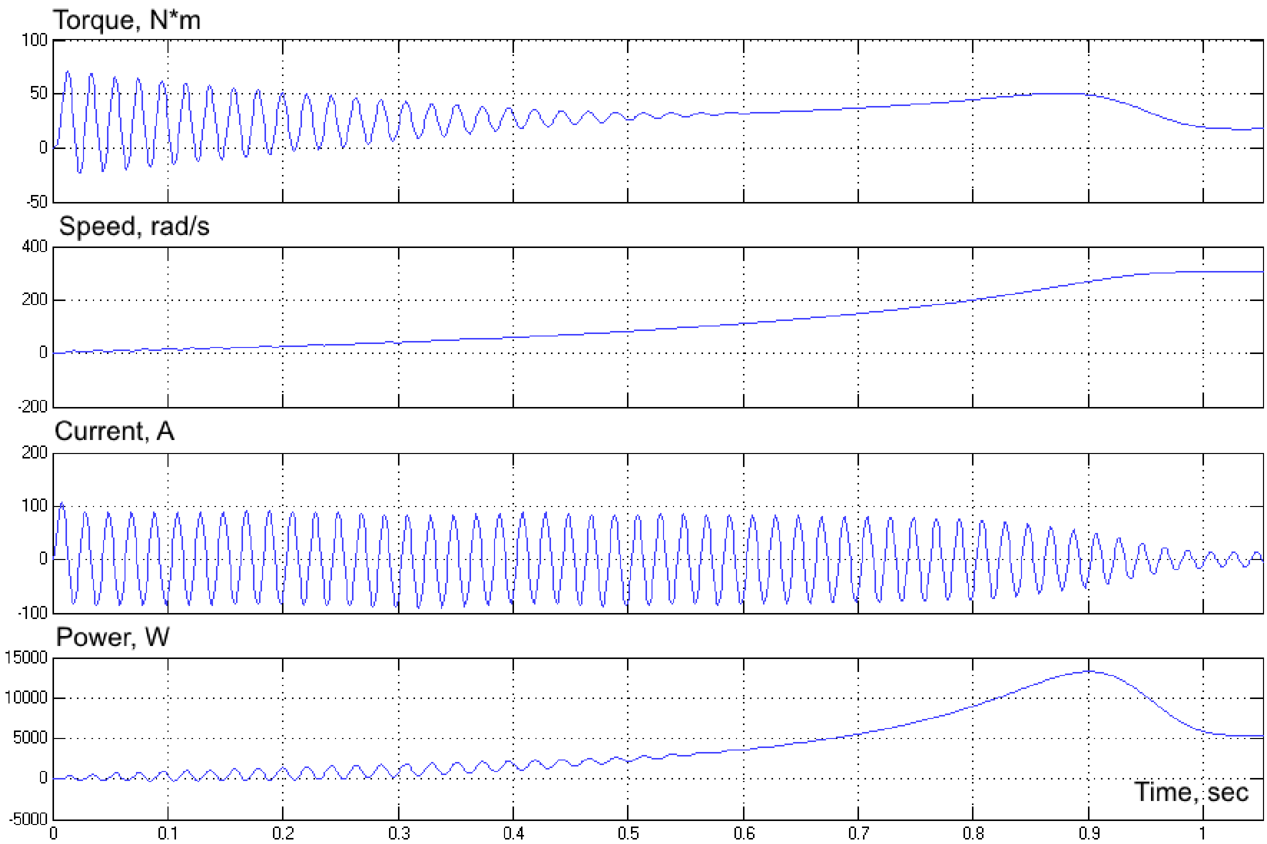

Fig. 4.12 Torque, speed, current and power curves for compressor in case of capacity control during transients

It is still assumed that there is constant air consumption meaning that after the compressor reaches the referenced pressure, the power consumption is the same.

In Fig. 4.13, there are presented curves after the stabilization. Pressure graph is shown in Fig. 4.14.

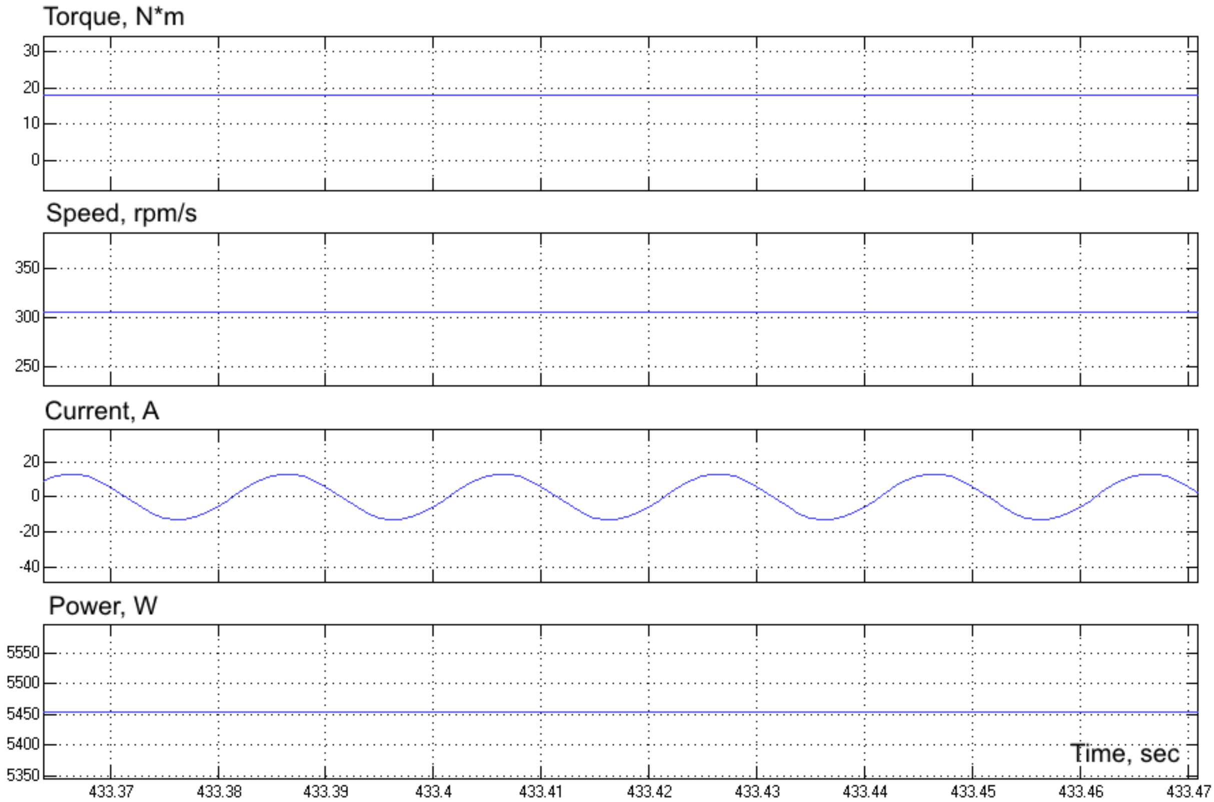

Fig. 4.13 Torque, speed, current and power curves after the stabilization for capacity control method

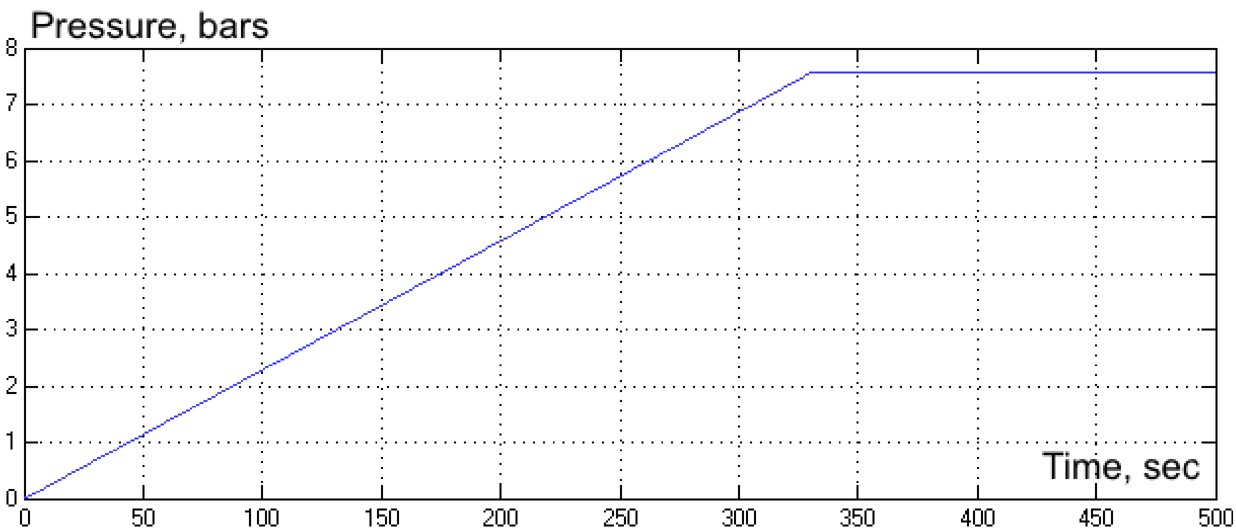

Fig. 4.14 Pressure distribution for capacity control

The power consumption is 84% of the nominal value, while the mass flow is reduced to 50%. The throttling control method has the limited control range. The method is useful when the mass flow capacity stays from 70% to 100%. However, if the mass flow rate becomes lower than 70%, the energy consumption is relatively high.

4.7 Outlet valve control

The method is quite rare, but still can be used to control the mass flow rate. The approach is based on the idea of increasing the resistive torque applied to the electrical engine by installing the valve to the outlet of the compressor. Consequently, the speed is decreased leading to the reduced mass flow rate. The additional load is added through the block «additional resistance» in Fig. 4.8.

During the simulation, however, it was realized that this approach has significant drawbacks. Like in case of capacity control, the amount of startups should be low. It can be clearly seen in Fig. 4.15.

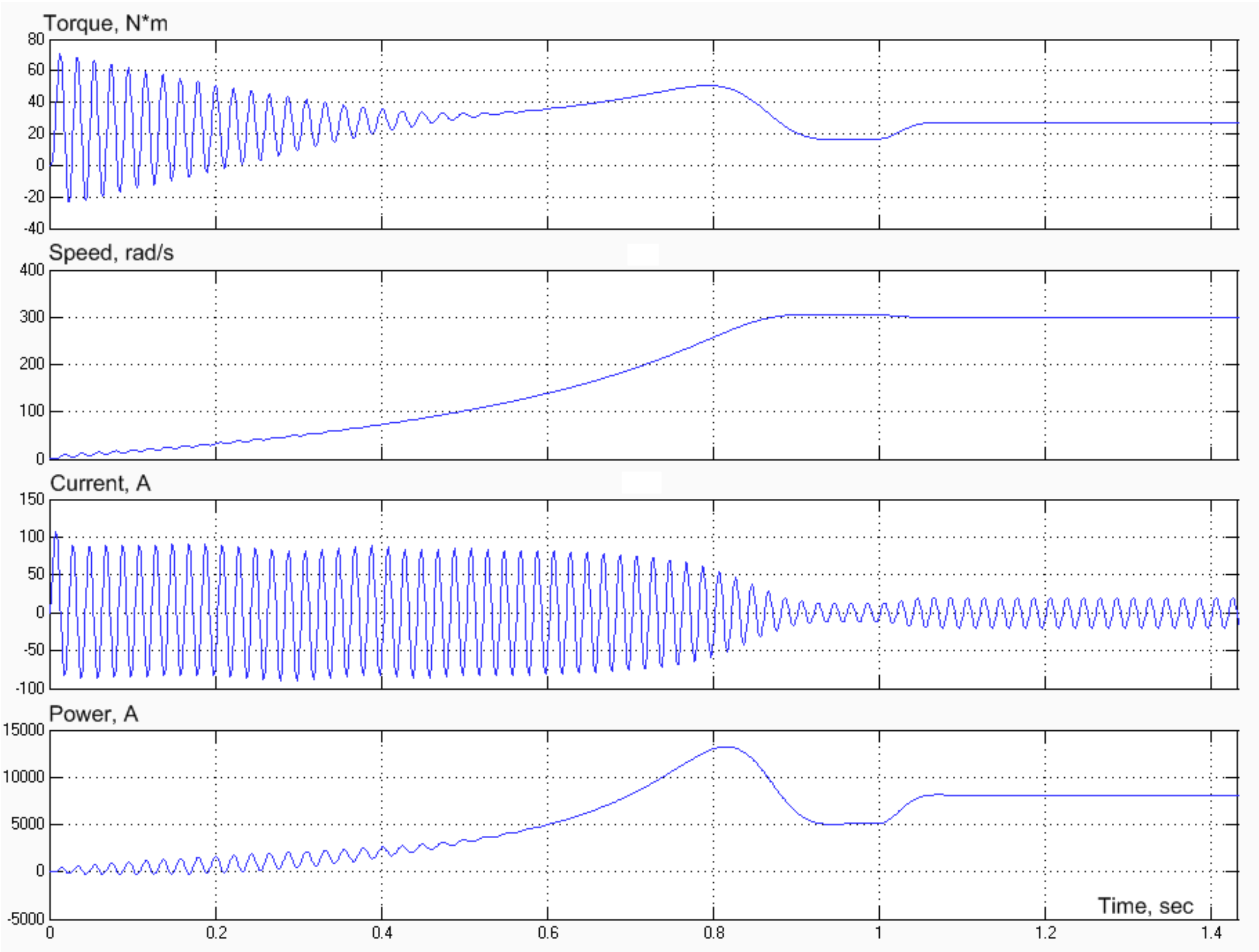

Fig. 4.15 Torque, speed, current and power curves during the transients for outlet valve control method

The additional load should be applied after the motor has completely accelerated. If the additional load is applied instantly, the motor could not overwhelm starting resistive torque. It is related to the nature of the induction motor: the applied resistive torque should not exceed the machine starting torque. During the simulation, the additional resistive torque (10 Nm) is added on the first second.

Another drawback is that the control range is quite limited based on the information that is received during the simulation. For example, the gas tank filling time (Fig. 4.16) is only increased of 4.3 seconds. However, the power consumption is increased of 62.5% (Fig. 4.17) According to the simulation data, this method can be used only in case if the motor rated power is high and the motor works without high load or the compressor is rather small when significant power increase is not strongly affecting the energy losses. Still the energy consumption required for the implementation of the control method is quite high in comparison with other control methods.

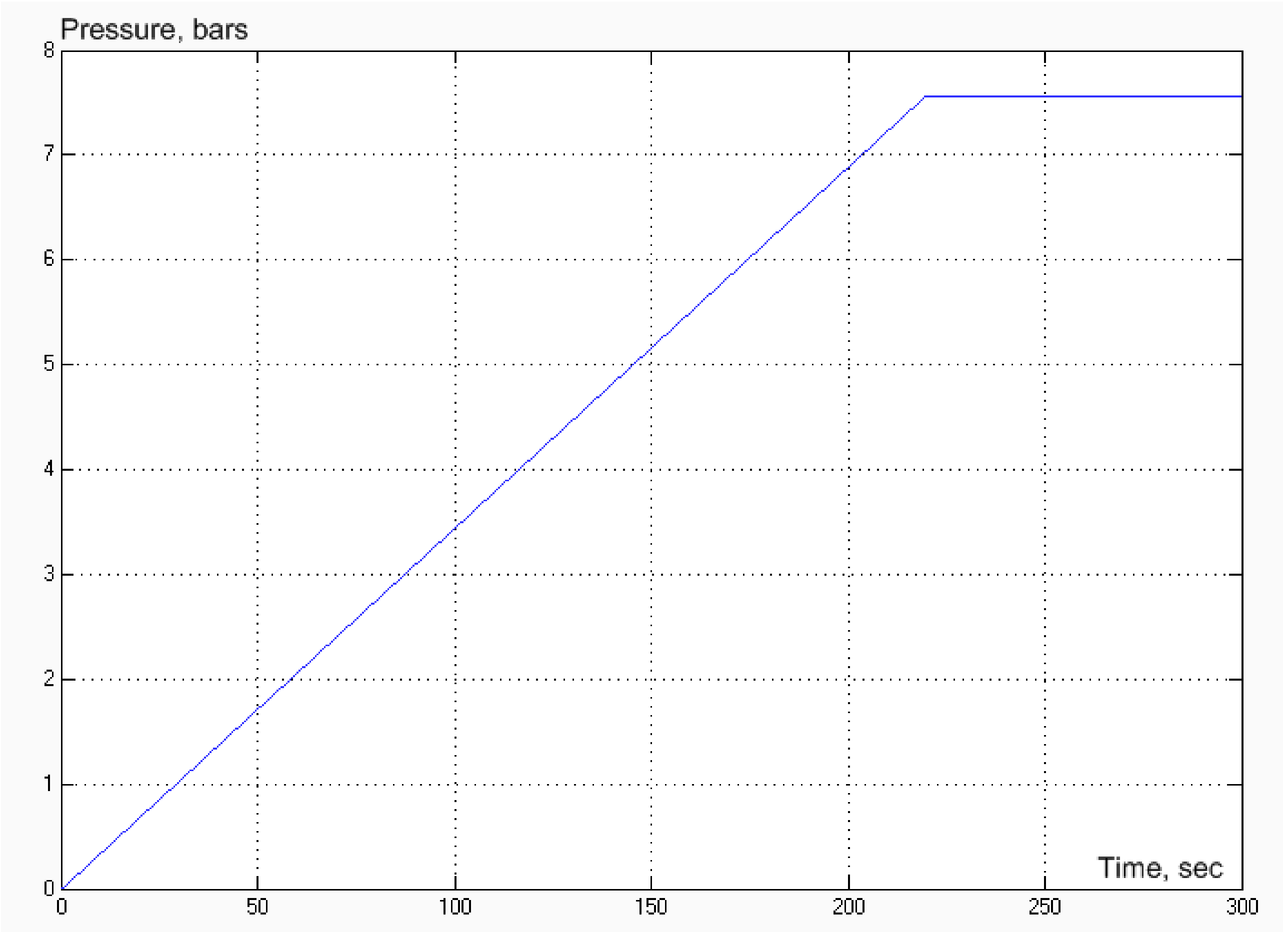

Fig. 4.16 Pressure distribution over the time in case of outlet valve control

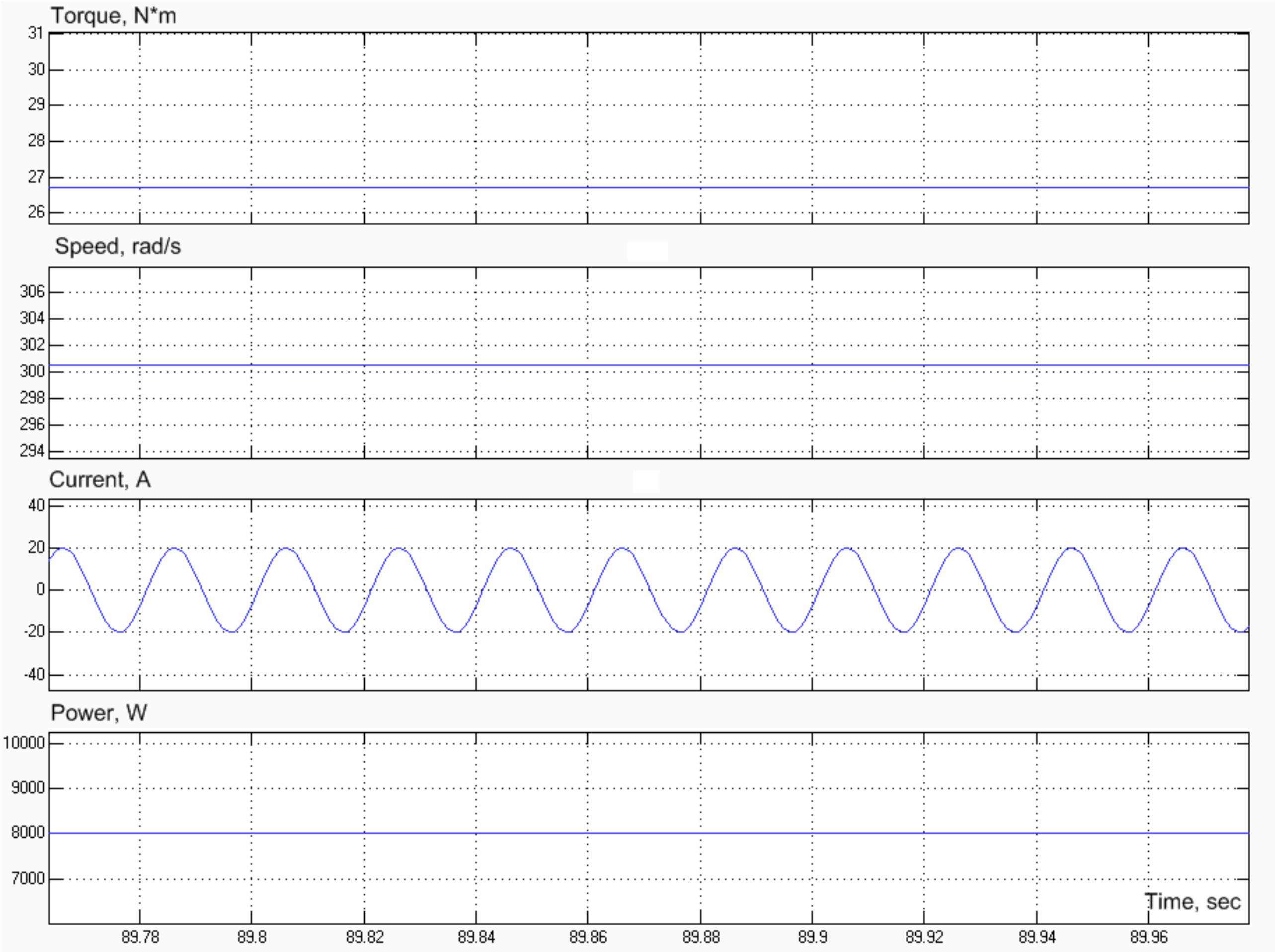

Fig. 4.17 Torque, speed, current and power curves after stabilization for outlet valve control

4.8 Induction motor vector control model

The vector control equivalent scheme is based on the following equations [10] [13]:

$$u_{\mathrm{Sx}} = r(1 + T_{\mathrm{s}}')i_{\mathrm{Sx}} - \omega_{\mathrm{K}} L_{\mathrm{s}}' i_{\mathrm{Sy}} - \frac{k_{\mathrm{r}}}{T_{\mathrm{r}}}\psi_{\mathrm{Rx}} \tag{65}$$

$$u_{\mathrm{Sx}} = r(1 + T_{\mathrm{s}}')i_{\mathrm{Sy}} + \omega_{\mathrm{K}} L_{\mathrm{s}}' i_{\mathrm{Sx}} + k_{\mathrm{r}} p \omega_{\mathrm{m}} \psi_{\mathrm{Rx}} \tag{66}$$

$$0 = -k_{\mathrm{r}} R_{\mathrm{R}} i_{\mathrm{Sx}} + \frac{1}{T_{\mathrm{R}}}\psi_{\mathrm{Rx}} + s\psi_{\mathrm{Rx}} \tag{67}$$

$$0 = -k_{\mathrm{r}} R_{\mathrm{R}} i_{\mathrm{Sy}} + (\omega_{\mathrm{k}} - p\omega_{\mathrm{m}})\psi_{\mathrm{Rx}} \tag{68}$$

$$T_{\mathrm{e}} = \frac{3}{2} p k_{\mathrm{R}} (\psi_{\mathrm{rx}} i_{\mathrm{sy}}) \tag{69}$$

$$Js\omega = T - T_{\mathrm{L}} \tag{70}$$

It is assumed that the model utilizes two control channels, which are connected to each other. *X*-channel defines the rotor flux linkage, while *y*-channel defines the current i_{sy}. If it is possible to organize the constant signal in the *x*-channel, then the flux linkage of the rotor is constant and the signal of the second channel estimates the output torque of the system. In this case, the induction motor model is similar to the DC-machine model.

There is a requirement to implement controllers in the both channels. The first channel requires only a flux linkage controller, while the second channel requires both the current and speed controller.

The scheme of the vector controlled induction motor is shown in Fig. 4.18.

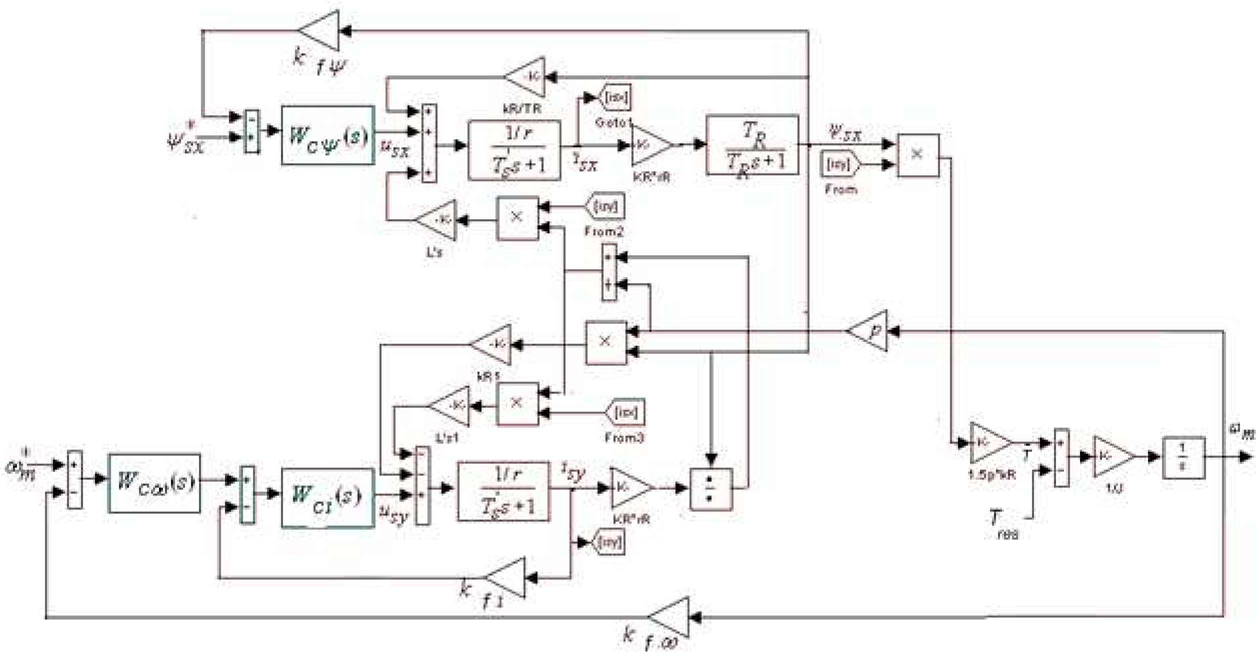

Fig. 4.18 Induction motor model for vector control [10]

Since there is a speed feedback, the both controllers in the second channel utilize PI-controllers with the technical optimum settings. Same applies to the flux linkage controller. The calculation of the motor parameters (resistances, inductions, etc) same like in case of the direct on-line model. The full view of the vector control model for simulation is shown in Fig. 4.19.

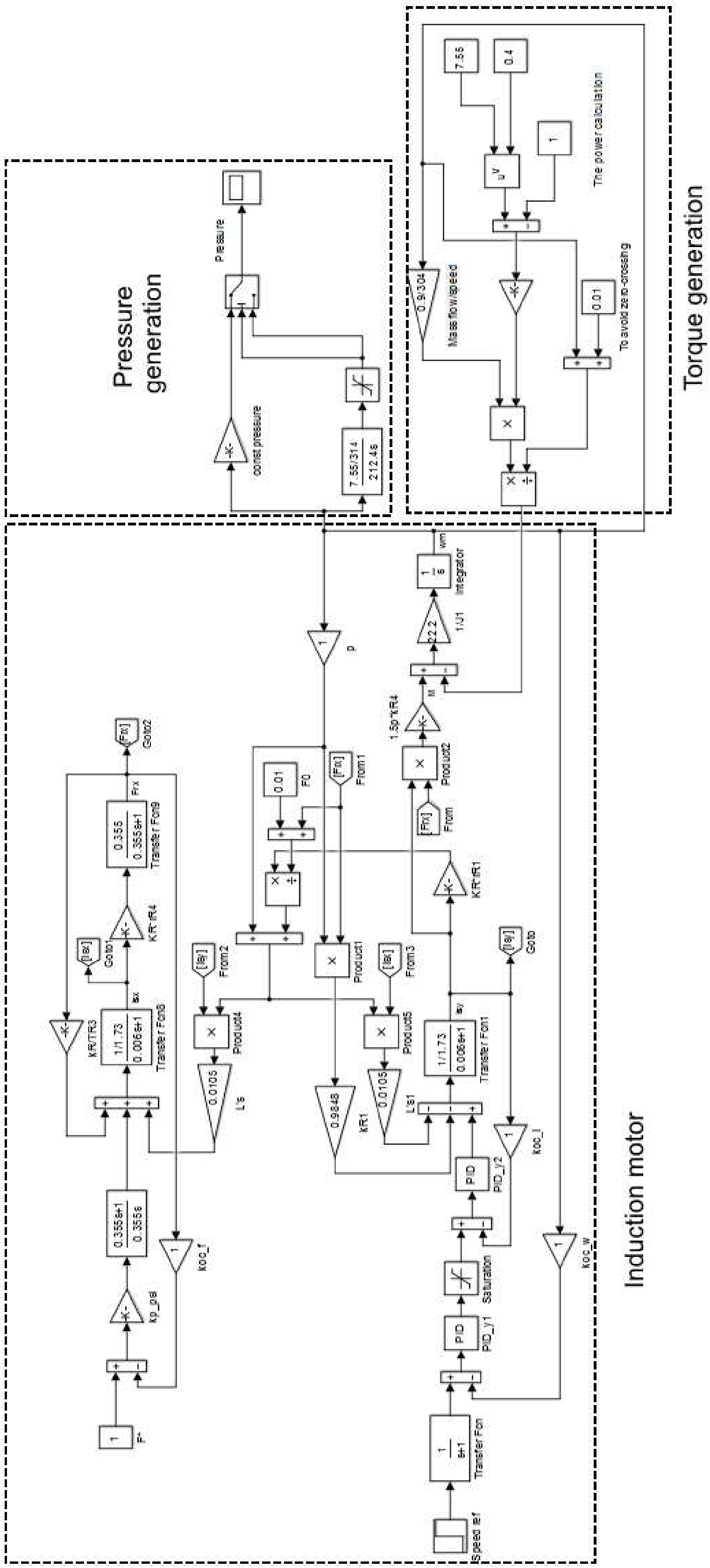

Fig. 4.19 Model for vector control simulation in Matlab Simulink

The model has some simplifications since it is developed to compare it with the throttling control method. It does not implement the power rise caused by the oil-pump and it does not utilize the blowdown technique. The motor speed was decreased twice in the comparison with the nominal speed in order to reduce the mass flow rate to 50%.

In comparison with the direct on-line model, it is possible to realize the «soft-start» approach thereby significantly increasing the starting capabilities of the system. The curves during transients are presented in Fig. 4.20.

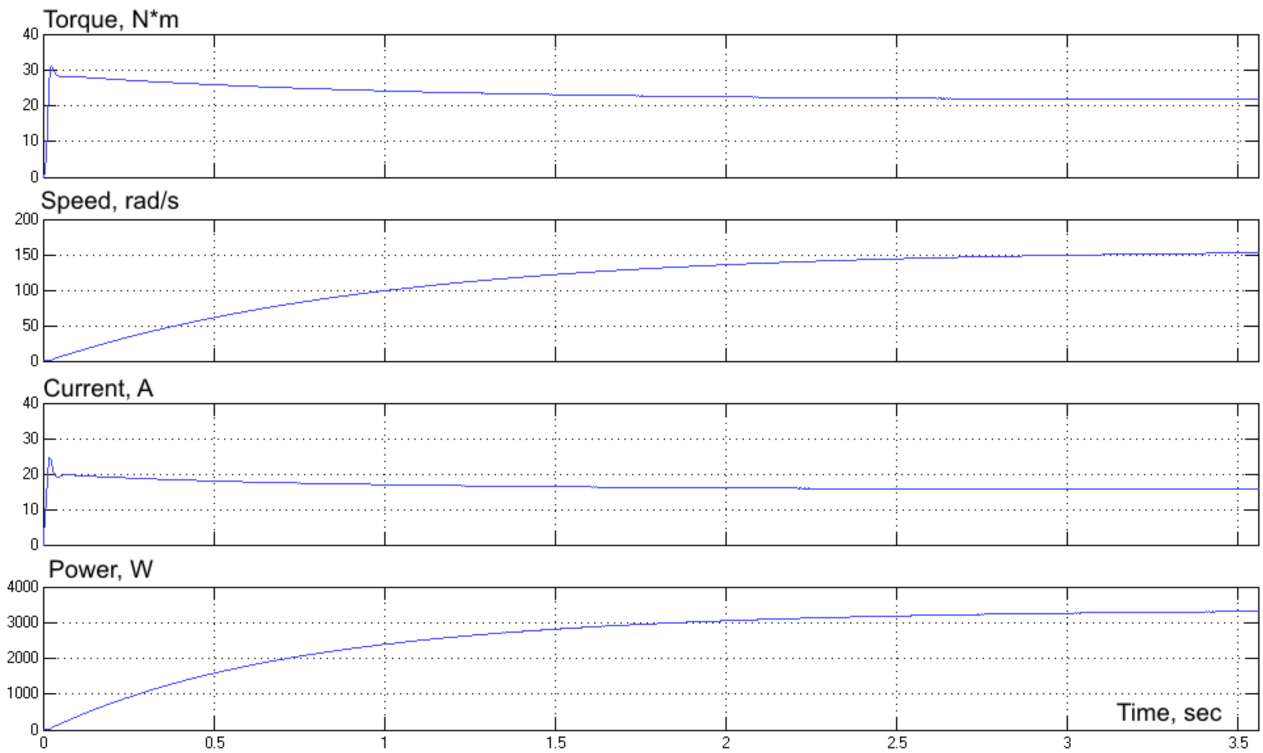

Fig. 4.20 Torque, speed, current and power during startup

The characteristics during the transients are significantly improved meaning that for vector control it is possible to make startups more frequently without harming the motor or the compressor. It is important to note that constant current is a result of approximation to the DC-machine. The graph after the transients is shown in Fig. 4.21. The pressure distribution over the time is shown in Fig. 4.22. There is no any difference in the pressure distribution in comparison with other control types.

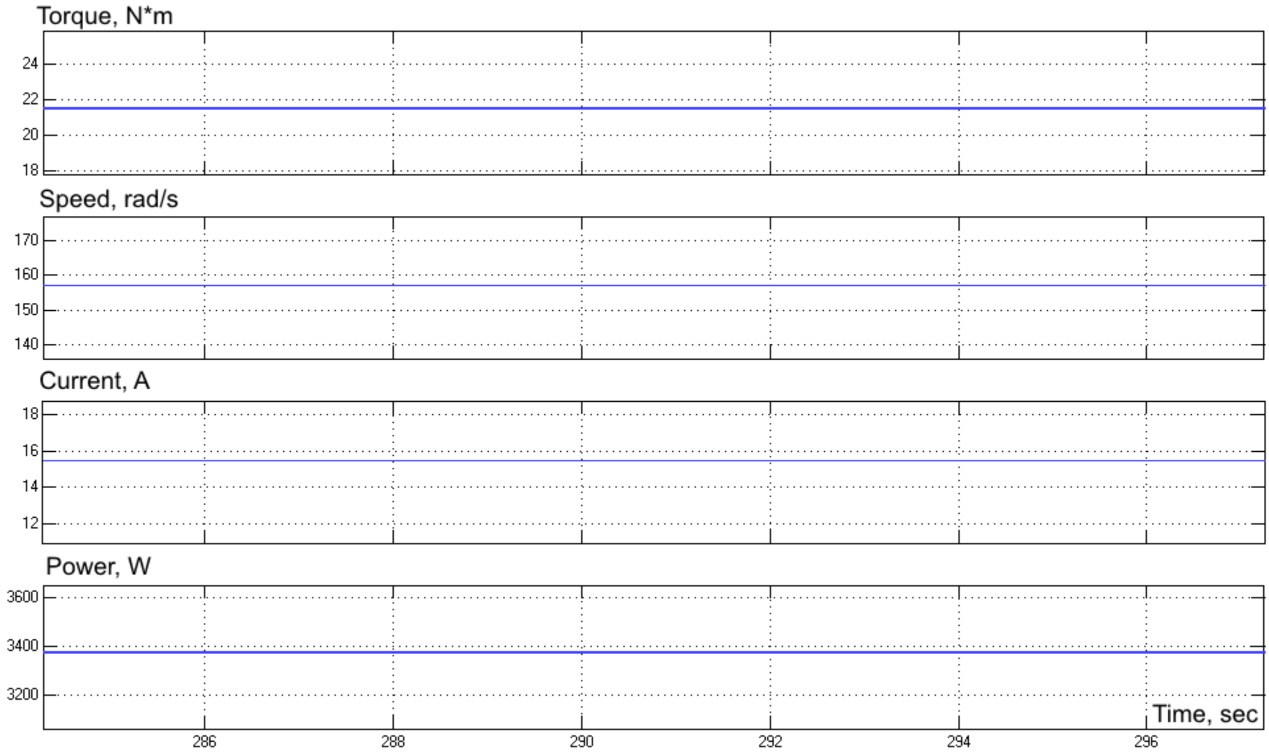

Fig. 4.21 Torque, speed, current and power after stabilization

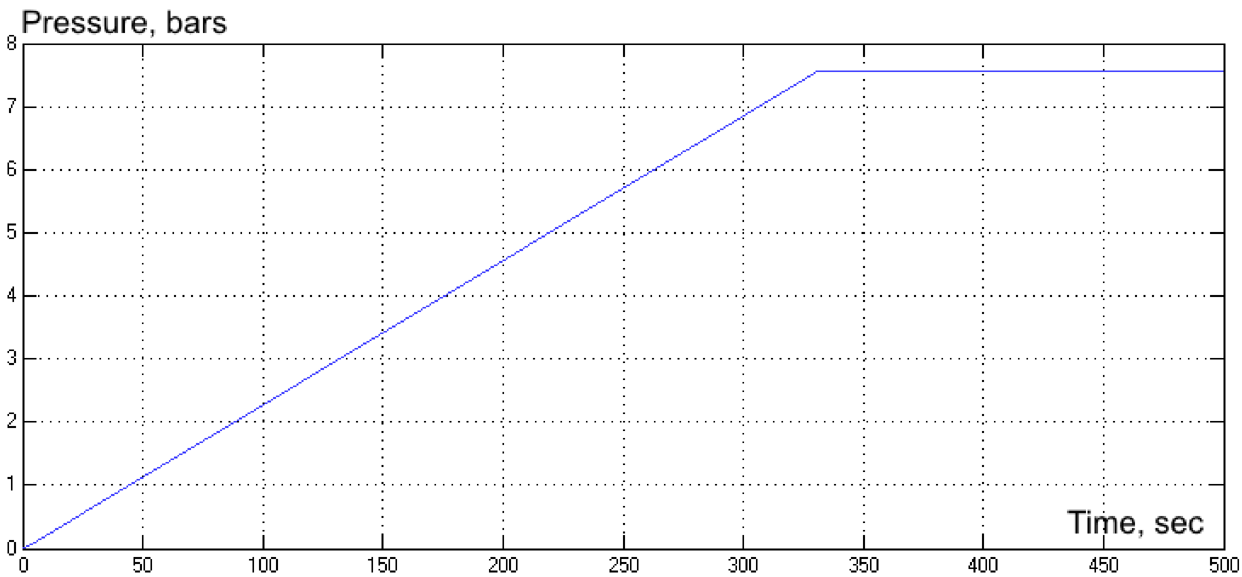

Fig. 4.22 Pressure distribution over the time

4.9 Power consumption comparison

Key task of the thesis is to estimate the power consumption of the different control methods. The estimation is carried out when there is requirement to decrease the mass flow to 50%. The power distribution is provided in Fig. 4.23.

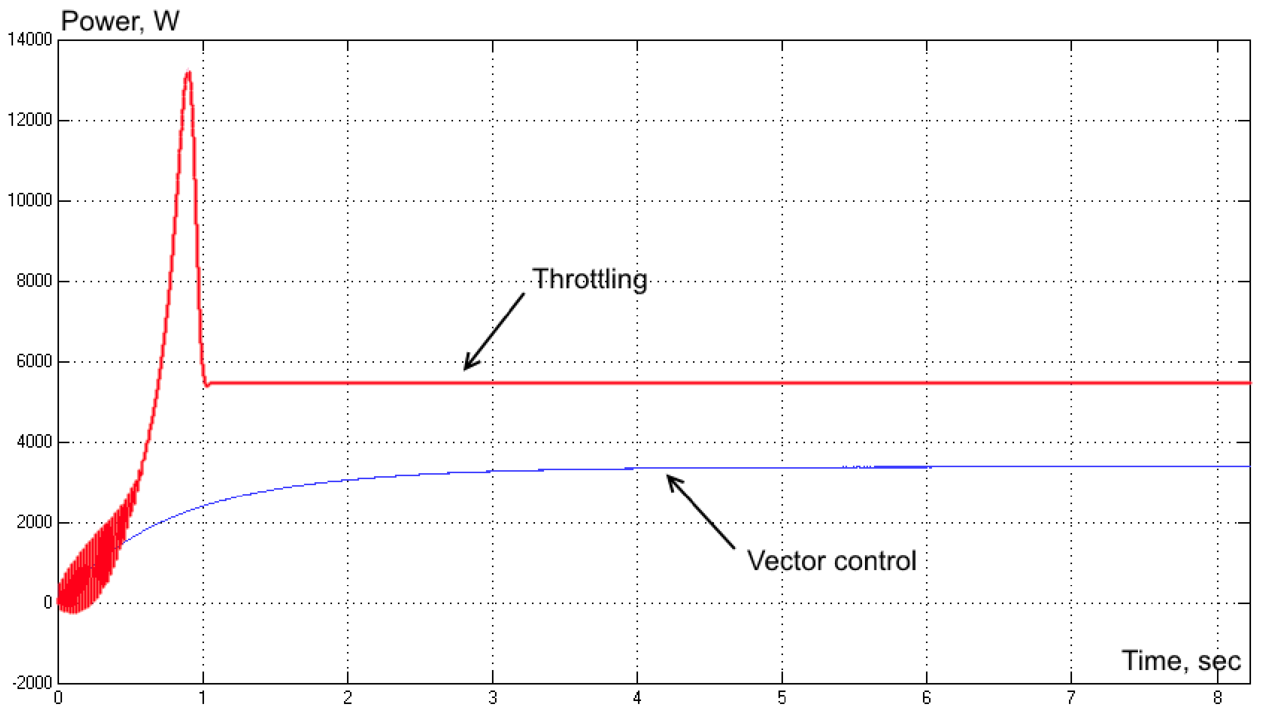

Fig. 4.23 Power distribution for vector control and throttling

During transients, the graph for throttling has a significant spike, while vector control provides smooth power production. After stabilization trend is the same for both curves. The relatively long time for vector control to be settled is caused by time constant in the soft-start (ramp-function generator) and it can be adjusted if it is required.The resulting pressure has the same distribution over the time. However, after the transients power required for throttling is 5454 W, whereas for vector control it is 3380 W resulting to 61% difference.

However, when the mass flow reduced to 81%, the difference in power has significantly decreased to 15%, where for throttling – 6140 W, for vector control – 5340 W (Fig. 4.24).

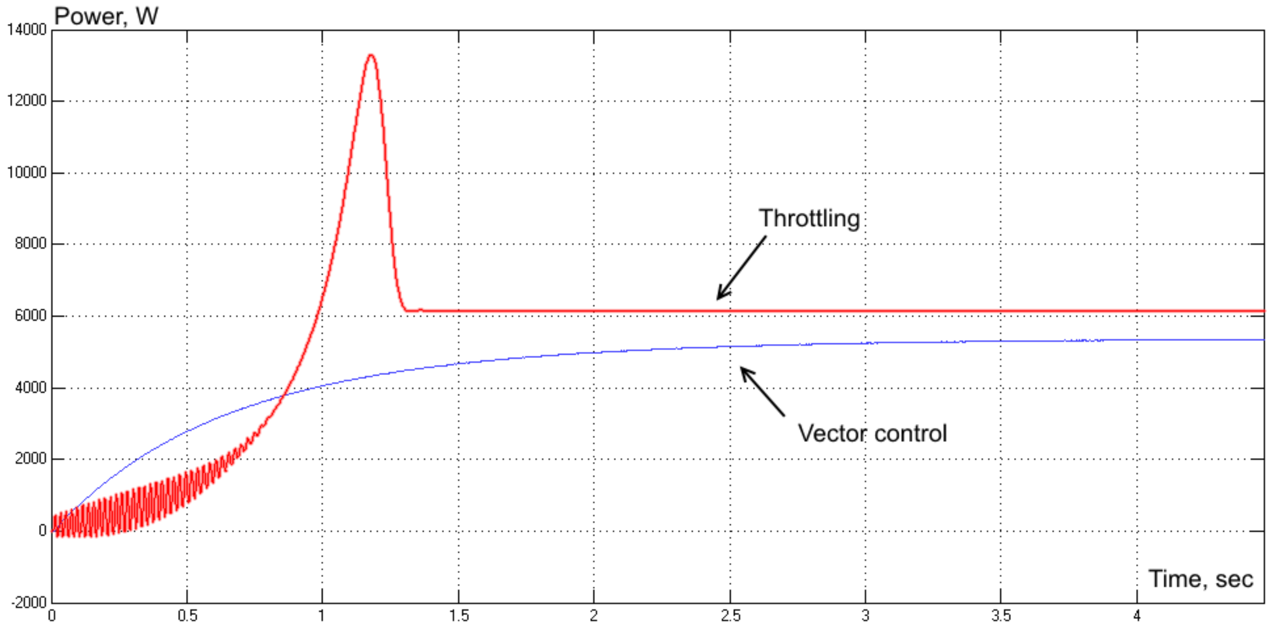

Fig. 4.24 Power distribution for vector control and throttling

For small changes in the mass flow rate, the power consumption difference is not significant. Additionally, throttling does not require an expensive converter for the operation. Also the amount of the sensors in the system may be lowered thereby decreasing the total price of the system.

However, if there is requirement for frequent start and stop or for wide operational range, then vector or frequency control should be considered as more optimal. Additionally, it still suggested calculating the life cycle cost of the system in order to estimate if it is more profitable to purchase the equipment with frequency control. The main reason of that are long duty ratios of the compressors in general. Compressors tend to operate during the whole year with some stops for preventive maintenances. For example, for the case that is shown in Fig. 4.25, the additional energy consumption of the modulation control during one year is 7012 kWh or 252432 MJ. So, the total energy consumption should be precisely analyzed in order to choose the best option if lifetime is taken into account.

4.10 Comparison of experimental and simulation results

The second key task of the thesis is to compare laboratory test results with the simulation. In order to make a precise simulation, the model enables both the oil pump and the capacity control valve.

According to the datasheet, when the pressure reaches the required parameter, the total load of the system is decreased to the 67% thereby implementing capacity control. In the model, capacity control is implemented through the gain and switch. Gain decreases the power to 33%, while the switch realizes the shifting to the required power according to the algorithm. The transient overshoot is also taken into the account, since it can be calculated through the technical optimum settings.

The simulation model does not utilize the belt gearing for simplification. Generally, it leads to the difficulties to calculate exact time, when the test equipment reaches the desired pressure. It can be clearly seen in the pressure distribution graph, where there is a slight difference in angle between simulation and experimental graphs. The curves for different speeds are presented in Fig. 4.25–4.28.

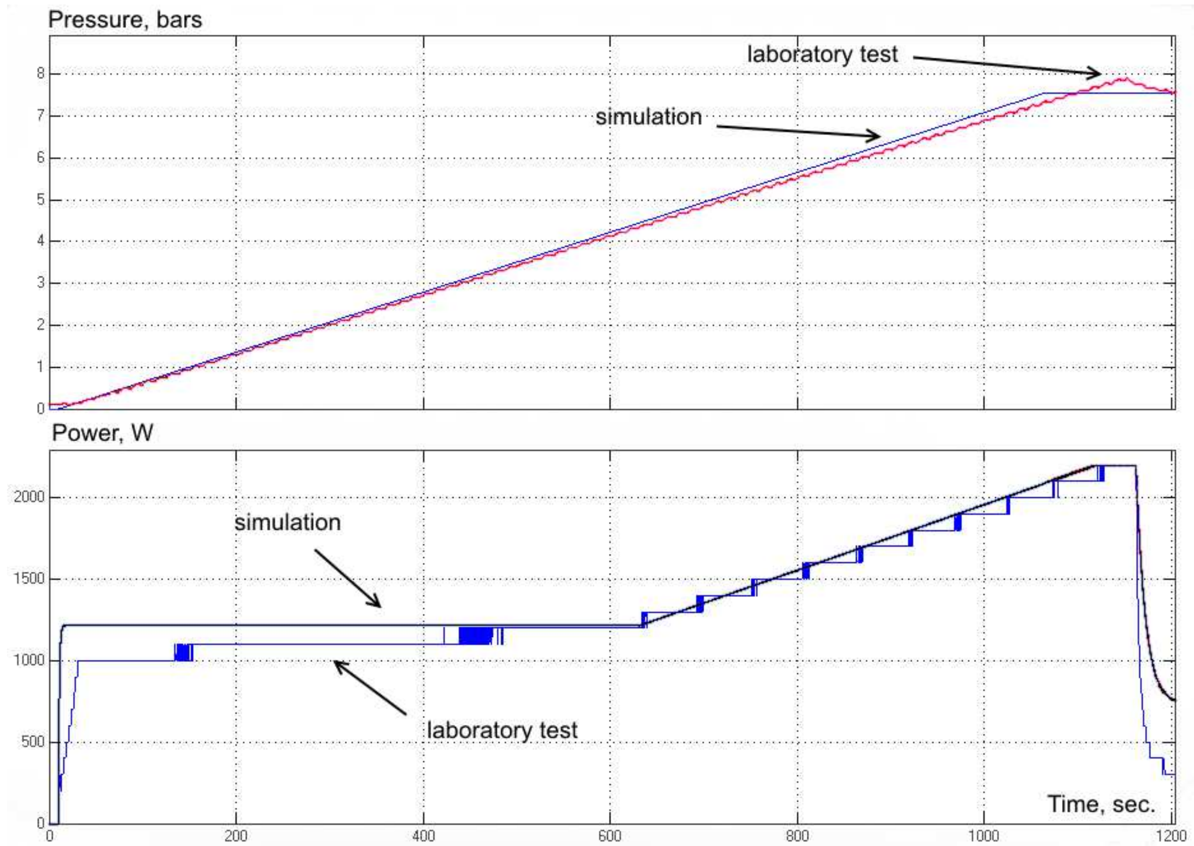

Fig. 4.25 Pressure and power curves for simulation and experiment for 586 RPM

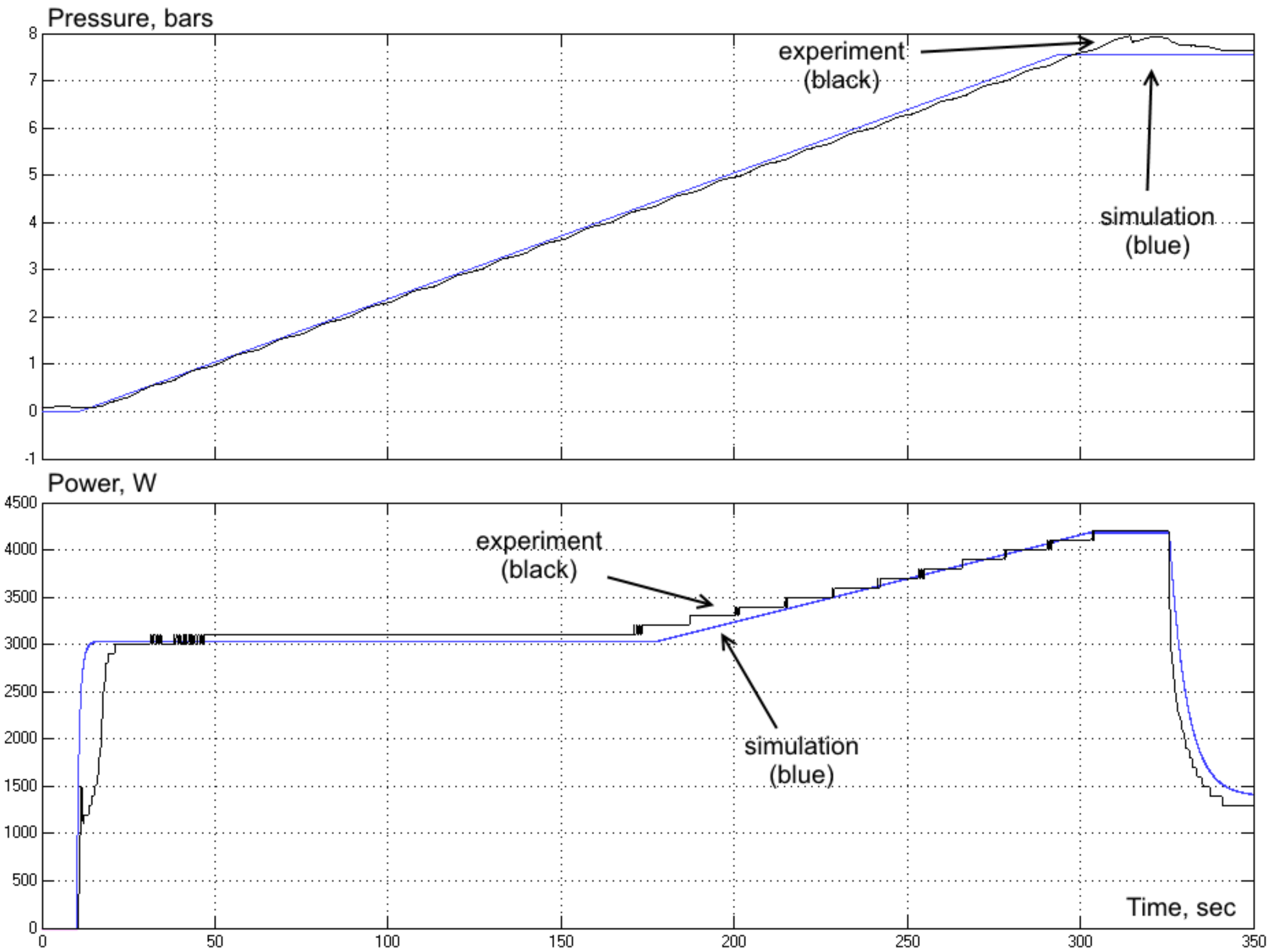

Fig. 4.26 Pressure and power distribution for simulation and experiment for 1752 RPM

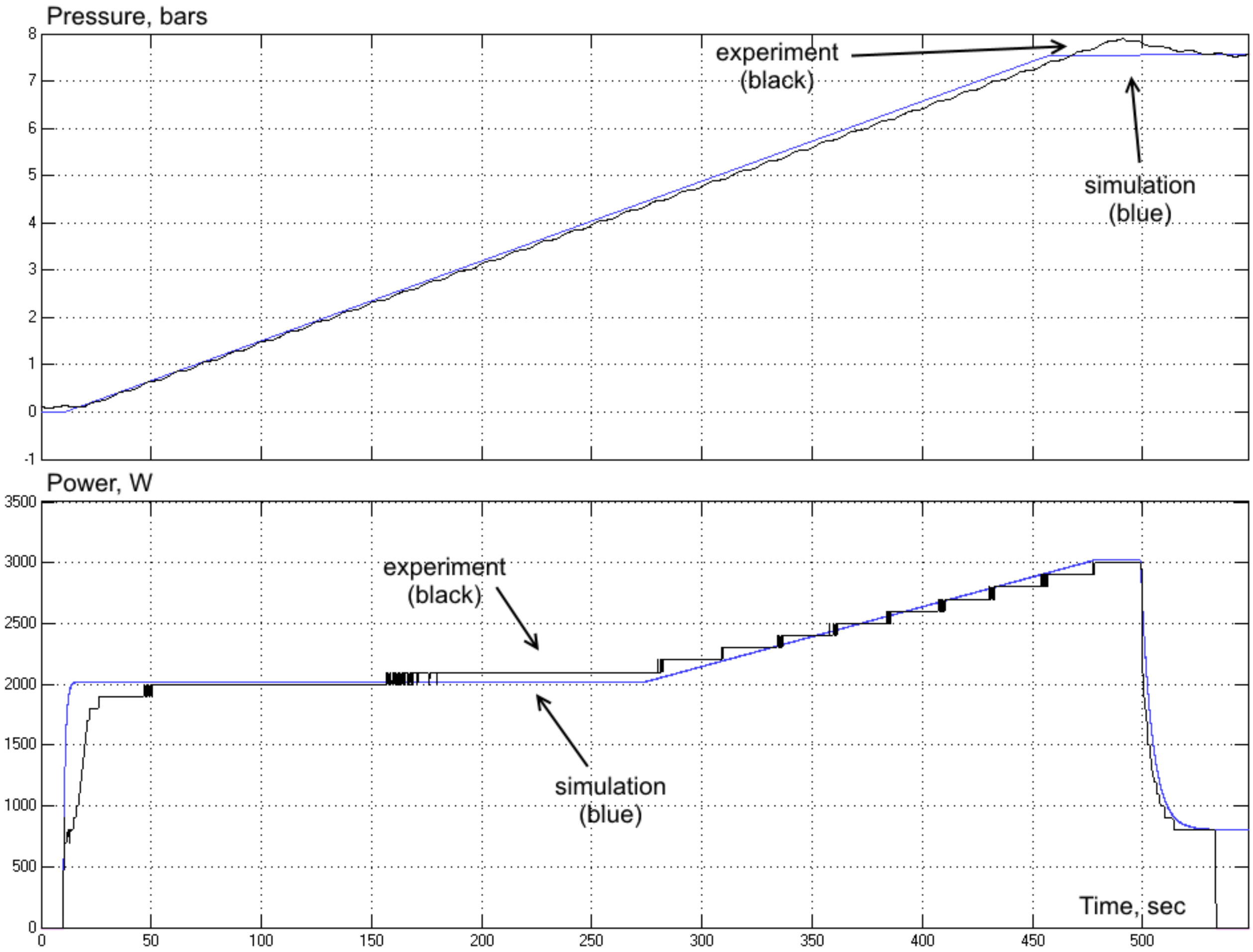

Fig. 4.27 Pressure and power distribution for simulation and experiment for 1168 RPM

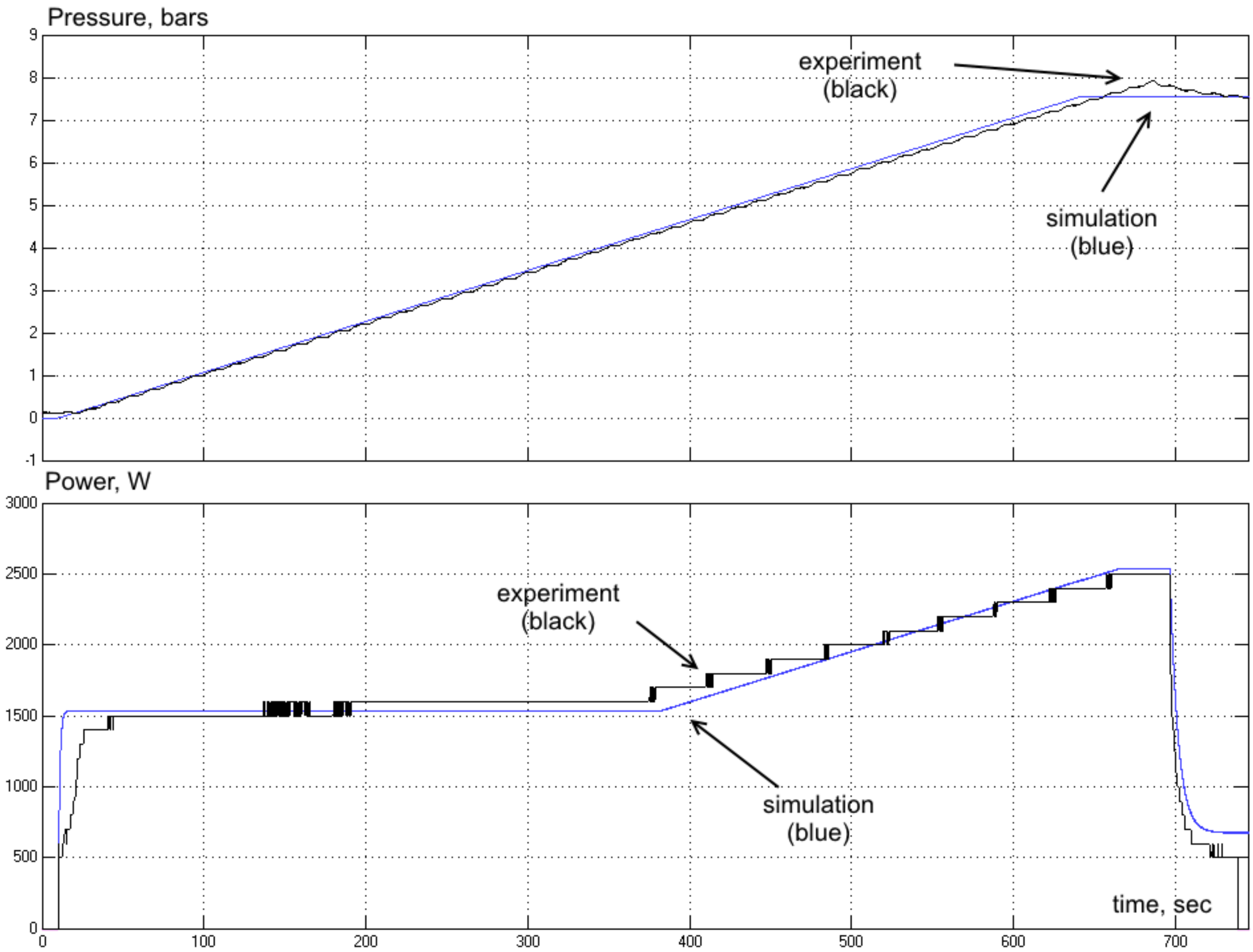

Fig. 4.28 Pressure and power distribution for simulation and experiment for 876 RPM

The curves for different speeds have approved the high precision of the model. The more deep analysis of the curves is provided in the following chapter.

4.11 Recommendations for the system improvement

According to the comparison of the simulation and real experiment, basic recommendations are developed for the compressor improvement. Recommendations in the thesis are applied to the laboratory equipment. However, recommendations can be considered for any compressor.

It is rather problematic to define the rule for the time required to fill the gas storage. It was caused by the belt drive, which has the first-order transfer function with constant coefficient and time constant. The time constant is depended on a lot of parameters, such as speed of the motor, the environment temperature, abrasion, etc. [11] Also the belt drive has its own efficiency leading to the some wasted power. So, it is recommended to implement the direct connection between the engine and the compressor.

The compressor system does not need a low setpoint time. However, it is still recommended decrease it. For example, during the interval of 0.1 seconds some disturbances may occur (for instance, accidental voltage drop) and during this time the control system might not even realize that the disturbance has happened and consequently does not adjust the speed of the motor. Additionally, high setpoint time does not allow building a very precise model in the first place since during the test for different setpoint times, the expected output result is slightly different. Moreover, it is impossible to estimate the fluctuation caused by the leakages, since the measured pressure data already has heavy fluctuations caused by the sensor errors.

The simulation has faster time for the power to be settled in comparison with the experiment. It leads to the assumption that the time constant for soft start in experiments is even higher than in the simulation. The experimental current for the laboratory equipment during the transients is slighter than for the simulation. According to the simulation, the common starts for the motor are not going to be harmful. It leads to possibility to stop the machine completely instead of applying 33% of the nominal load after the reaching the required pressure inside the gas tank. However, most of the compressors have a protection that limits the amount of starts/stops to five per hour. [26] It is important to adjust the compressor working cycle according to the end-user requests.

It is still recommended to install speed sensor for vector control implementation or to use DTC control [15] instead of frequency control. With the speed sensor compressor control principle would be more reliable. To implement the system with current feedback and pressure feedback without speed sensor, it would be required to implement two PID-controllers. One of them will be installed on the pressure cascade and another on the current cascade. Simulation of such systems has proven sometimes inadequate reaction of the derivative part to the disturbances which led to the high spikes in the current cascade at least during the simulations. [23] [10]

Leakages should be estimated and minimized. During the experiments it was impossible to estimate the leakages. Nevertheless, reduction of leakages on 10% may decrease total energy consumption on 10% for the frequency control. [26] The recommendations to reduce leakages are mostly practical, rather than theoretical such as checking the noise from the pipes or applying ultrasonic diagnostic, etc. [26]

Summary

Nowadays, the tendency about reduction of energy consumption is becoming wildly concerned topic. The compressors being one of the most common devices in the industry are the key target of the researches connected to energy efficiency.

There are approaches directed to the enhancing mechanical part of compressors, such as developing new types of bearings, new casings, new more efficient rotor types or more durable pipes. In the thesis, however, different approach is used, which is targeted on the estimation and comparison of different control types.

During this work, the equipment which is part of the screw compressor alongside with the typical control methods is investigated and described. Based on the analysis, simulation models for different control approaches are developed. To verify the simulation model, the data from the real laboratory object is collected. The object utilizes the various ways to control the mass flow inside the system. The tests are carried out for different speeds to estimate the time required to fill the air storage. Output curves for simulation and real experiment are provided. After the model verification, the problems during developing the model are described and recommendations how to increase the overall energy efficiency of the concerned screw compressor.

Additionally, it is described the model development for various control methods and different models are compared with each other. In order to do this the same production output is compared for models. As the result, the throttling method is described as a satisfactory for the limited control range. If there is requirement to decrease the mass flow below 70%, the control method becomes inefficient due to high energy consumption after that point. Also the compressors tend to work with high duty ratios so even in the control range from 80% to 100%, there is a requirement for calculation of the payback periods for different approaches. Outlet valve control is shown quite low energy efficiency, but it is still considerable for small compressors or in case when the compressor works for short periods of time. After the simulation and comparison, the recommendations of possible system improvement are provided.

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