Safety Factors – Conservatism Principle in the Design and Construction of Process, Energy, and HVAC Plants and Equipment

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It can generally be stated that successfully designed process, energy, and HVAC plants plants are characterized by both quality and quantity of the final product. Basic principle for every plant design state that every single piece of equipment must be sized as to perform its function in safe and efficient manner. To this end, due to many uncertainties, engineering design and construction often use the safety factor. Although this term is in everyday use in the engineering practice, the safety factor principle is often oversimplified and misunderstood.
Paper presents a complete overview of application of safety factors in every aspect of plant and equipment design. Exhaustive literature review, coupled with authors' own extensive experience in plant and equipment design, serve not only to present the readers a clear background of basic principles and recommendations for safety factors, but also the help them understand all the complexities and aspects used in its formation.
Examples provided in the dedicated section are meant to substantiate all the outlined concepts, while authors' own recommendations can guide the readers towards correct safety factor selection.

Keywords: *safety factor, design, material properties, equipment specification, capital and operating cost*

1. INTRODUCTION

Main aim of this paper is to present the reader the com–plexities and background behind safety factor definition and its effects on the plant and equipment design. First part of the paper gives an overview of recommendations related to the safety factors for various equipment used in the engineering practice (plants overall, static equipment – piping and pressure vessels, rotating machines, heat exchangers, apparatuses etc.), as well as the consequences of its application, such as impact on CAPEX, reliability, and future plant expansion.

Next, paper presents background of safety factor constituents, including safety factors related to the material properties, including literature data, and the impact of various external factors on the safety factor itself. This information should help readers to address impact of various personnel included in plant life cycle (from feasibility study up to the end of usefull plant life). Recommended safety factor values are alos pre–sented based on extensive literature research and authors' own experience.

Finally, several examples given in the Appendix are presented from everyday practice and authors' own experience which serve to substantiate the previous statements. Also, at the very end of the paper the aut–hors present their own recommendations based on extensive academic and industrial experience accumu–lated over the years.

1.1 Definition and importance of safety factor in engineering design and construction

Engineering activities are always closely connected to plant design and equipment sizing, which are in term based on design parameters typically included in a dedicated design specification. This is true not only for engineering activities related strictly to design, but also for those related to construction, testing, erection, supervision, maintenance etc. Engineering designs typically undergo internal or external verification and, in this day and age, with the development of personal computers and dedicated calculation software, they should also be of high degree of accuracy. However, process, energy, and HVAC plants practically never operate under the defined design conditions (flow rates, temperatures, pressures, phase compositions etc.). Reasons for deviations between operating and design parameters are many, such as:

- use of raw materials (feedstock) different from the design specification;
- lower production capacities or impossibility of maintaining the required quality of final product during plant operation (due to presence of the so–called "bottlenecks" in production process) which may force plant operation with higher quantity of utilities or lower flow rates of process fluids;
- over–sizing of parts of the plant (or equipment) makes it possible to operate the plant with higher quantities of process or lower quantity of energy

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bearing fluids while maintaining the final product quality;

- chemical or physical properties of fluids adopted during the design phase may be different from the realistic values;
- existence of issues related to equipment corrosion or fouling which reduce plant capacity and which cannot be foreseen (in part or in whole) during the design phase;
- installed instrumentation and regulation equipment cannot satisfy the requirements of precise process management;
- modification of the final product and by–product requirements with respect to the design specification (e.g. presence of impurities in final products or feedstock);
- waste materials (outputs – gases, liquids or solids) may require additional treatments to prevent atmospheric pollution (e.g. due to changes in environmental protection standards).

An engineer is always expected to satisfy the three basic criteria during the planning, design and const–ruction of plant or equipment [1]. These are:

- functionality;
- safety;
- economical optimization.

These criteria are interconnected and often in direct contrast with each other. While functionality and optimization criteria should be respected as much as possible, safety criteria are almost always mandatory and dictated by dedicated Laws and Technical regulations.

In order to provide safeguard against the errors resulting in quantitative and/or qualitative deterioration of productivity (either of the entire plant or single subsystems), or even complete and catastrophic plant failure, various engineering calculations include factors whose main purpose is to ensure a certain safety margin. In other words, even if engineers have extensive experience with same or similar plants or equipment, they often resort to oversizing. This engineering practice is a direct result of compensation for various inaccuracies which emerge during the engineering work and is called the *conservatism principle* or *overdesign*. Calculation factor used in it is typically called *safety factor* (although there are several synonyms in use such as design factor, factor of safety, overdesign factor, safety margin etc.).

Safety factor (*SF*) is defined as the ratio between the maximal (*MC*) and design characteristic (*DC*) of plant or equipment

$$SF = \frac{MC}{DC} \tag{1}$$

while margin of safety (*SM*) is defined as

$$SM = \frac{MC - DC}{DC} = SF - 1 \tag{2}$$

Having in mind the drastic definition of [2] which states that “the factor of safety is a factor of ignorance”, it is the Authors’ opinion that it is necessary to review various aspects of safety factor application in planning, design, construction, erection, purchase or sale of plants and equipment.

Generally speaking, there are two types of safety factors in mechanical engineering:

- safety factor related to the material properties;
- safety factor related to the technical performances of plant or equipment.

2. SAFETY FACTOR RELATED TO THE MATERIAL PROPERTIES

Whether they concern system/equipment safety or catastrophic failure prevention, the calculations done during the design phase which aim to ensure correct sizing and correct design are called strength calculations, mechanical calculations or stress analyses. These calculations compare the maximal (or ultimate) stresses which the material of construction can withstand (*MS*, N/m^2) and the actual design stress in the analyzed element (*DS*, N/m^2) in the following form

$$SF = \frac{MS}{DS} \tag{3}$$

in which *SF* denotes safety factor and the design stress *DS* denotes the critical operating stress which may occur within the element in any moment. If the maximal design stress (*DS*) of a critical equipment element is lower than the material ultimate stress (*MS*), this element will be able to withstand the operating condi–tions without damage. Simply put, if *SF* is greater than 1 the construction is safe to operate. This fact is comp–letely self–explanatory, but warrants additional analysis from an engineering point of view.

By applying a safety factor many systems or equip–ment are intentionally build much stronger than required under normal operating conditions. This is done in order to ensure safe operation under unforeseen and occa–sional events which tend to occur during the plant lifetime, such as: unexpected loads due to change of operating conditions (pressure, temperature, concent–rations etc.), cyclic loads (i.e. fatigue), misuse of equipment etc.

Safety factor is often confused with the design safety factor (*DSF*) which is defined as

$$DSF = \frac{MS}{AS} \tag{4}$$

in which *AS* (N/m^2) denotes the allowable stress in the analyzed element.

The term “design safety factor” or “design factor” is defined in a number of engineering standards, practices and recommendations and so engineers are relatively well aware of its meaning. It is common opinion that the design safety factors are well studied and that they ensure design consistency under different operating conditions and loads. For example, ASME standards for pressure vessels and piping typically define the design safety factor of DSF = 3÷4 for calculation of allowable stress based on ultimate tensile strength and DSF = 1.5 for calculation of allowable stress based on yield stress of material.

Key difference between SF and DSF is in their definition, since the allowable stress (i.e. *AS*) is typi–cally defined by technical regulations. To ensure safety the ratio

$$\frac{AS}{DS}=\frac{SF}{DSF} \tag{5}$$

has to be greater than, or at least equal to 1, which me–ans that $AS \geq DS$ or $SF \geq DSF$.

2.1 Design safety factor – literature data

In various literature sources (e.g. [3-7]) safety factor is defined based on type of loading, type of equipment and other parameters. Some of the examples are given in Tables 1 and 2 which show typical *DSF* values related to the material ultimate tensile strength and based on the type of loading.

Table 1 Typical design safety factors (*DSF*) based on type of loading

Material	Type of loading			
	Constant	Slightly variable	Broadly variable	Impact
Cast iron	4	6	10	15
Malleable cast iron	4	6	8	12
Carbon steel	4	6	8	12
Stainless steel	5	6	10	15
Other metals (Al, Cu, Pb etc.) and alloys	5	6	8	12
Wood	6	10	14	20
Brick	15	20	25	30
Rock	15	20	25	30

Table 2 Typical design safety factors (*DSF*) based on type of equipment or element

Component	*DSF*	Component		*DSF*
Boilers and pressure vessels	5	Gears	Static loads	5
Flywheel shafts etc.	8		Cyclic loads	8
Shafts	12	Steel ropes	General service	6
Steel constructions	5		Mine shafts	7
Welds (no cyclic loads)	4		Elevators	10
Turbine rotors and pedals	4	Springs	Less important	4
Bolts and nuts	8		More important	6

2.2 Formation of safety factor

According to its definition, the safety factor needs to ensure that the critical load of any element is not reached under any circumstances. In other words, the minimal value of safety factor equals to the prescribed value of the design safety factor. Numerical value of actual safety factor depends on multiple parameters. Principal parameters are:

- element loadings (i.e. forces and moments);
- material of construction;
- geometric characteristics at the critical point of the element.

Besides the above listed parameters, safety factor depends also on the element manufacturing method and load variations during the element lifetime. It is also important to consider that due to the environmental factors (e.g. corrosion, erosion etc.) during its lifetime the load bearing mass of any element will be reduced resulting in rise of element stresses. Finally, occasional loads due to external factors (e.g. seismic events or wind gusts) can result in short–term loads and high material stresses. Obviously, to calculate all of the above loads and stresses, an engineer needs to apply correct mathematical model and failure theory.

Material properties (e.g. yield stress, modulus of elasticity etc.) are determined experimentally, and even the best tests performed on identical samples made of identical materials may yield a dissipation of the results of ~5% [2]. This dissipation is a result of both material inhomogeneity and the test procedure itself, including accuracy of measuring equipment. It is important to note that if the same material properties reported in different literature sources are compared, this dissipation is significantly greater and can be well above 15% [2].

Procedure for forming the safety factor based on the above listed parameters is given in [2] as:

$$\mathrm{SF} = \mathrm{DSF} \cdot \mathrm{F_m} \cdot \mathrm{F_{ls}} \cdot \mathrm{F_g} \cdot \mathrm{F_{ft}} \cdot \mathrm{F_r} \tag{6}$$

In which:

- F^m partial safety factor for the material;
- F^{ls} partial safety factor for the load stress;
- F^g partial safety factor for the geometry;
- F^{ft} partial safety factor for the failure theory;
- F^r partial safety factor for the reliability.

Summary of each individual partial safety factor listed above is given in Table 3 based on analyses and recommendations explicitly given in [2], and implicitly given in [3-7].

2.3 Examples of constituents of pressure vessel safety factor

When discussing either pressure vessels or piping systems, there are several interesting constituents which form the elements of the total safety factor. These are discussed in detail in further text.

2.3.1 Corrosion allowance

The thickness of pressure vessel or piping system ele–ments are calculated according to standardized proce–dures from major design codes (e.g. ASME or EN codes). In these calculations, the possible influence of the fluid on the material is introduced through the cor–rosion allowance. Since corrosion is a complex pheno–menon, it is not possible to define specific recommen–dations for assessing the corrosion allowance. There–fore, the corrosion allowance is based on experience with the similar (or the same) material under operating conditions similar to those for the proposed design (e.g. by using predicted corrosion rates published in reliable sources). Typically, design codes and standards do not indicate the minimal required value of corrosion allowance. However, as a general guideline, for carbon and low alloyed steels where no severe corrosion is expected, a minimal allowance of 1÷2 mm should be used, while for more severe conditions this value should be increased up to 4 mm. On the other hand, if experience shows that the corrosion is only superficial or does not occur for the combination of the design parameters and construction material, a zero value can be used in design.

Finally, when a pressure vessel or piping system go into a corrosive service with no reliable service experi–ence or other data, it can be suggested that inspections be made at regular intervals in order to establish relati–vely accurate corrosion rate under service conditions.

Table 3 Typical partial safety factors

Factor		Conditions
Partial safety factor for the material		
F_m	1.0	Material properties are well known; have been experimentally obtained from tests on a specimen identical to the component being designed and from tests representing the loading to be applied
	1.1	Material properties are known from a handbook or based on manufacturer values
	1.2 ÷ 1.4	Material properties are not well known
Partial safety factor for the load stress		
F_{ls}	1.0 ÷ 1.1	Load is well defined as static or fluctuating; there are no anticipated overloads or shock loads; an accurate stress analysis method has been used
	1.2 ÷ 1.3	Load is defined in an average manner, with overloads of 20÷50%; stress analysis method may result in errors less than 50%
	1.4 ÷ 1.7	Load is not well known or the stress analysis method is of doubtful accuracy
Partial safety factor for the geometry		
F_g	1.0	Manufacturing tolerances are tight and held well
	1.0 ÷ 1.1	Manufacturing tolerances are average
	1.1 ÷ 1.2	Manufacturing tolerances are large
Partial safety factor for the failure theory		
F_{ft}	1.0 ÷ 1.1	Failure theory to be used is derived for the state of stress
	1.2	Failure theory to be used is a simple extension of the preceding theories
	1.3 ÷ 1.5	Failure theory is not well developed
Partial safety factor for the reliability		
F_r	1.1	Required reliability for the part is not high (e.g. less than 90%)
	1.2 ÷ 1.3	Required reliability for the part is average (i.e. 92÷98%)
	1.4 ÷ 1.6	Required reliability for the part is high (e.g. greater than 99%)

2.3.2 Adopting the design parameters (pressure and temperature) for pressure vessels

Both design pressure and temperature should be diffe–rent (i.e. more severe) from normal operating conditions of vessel. By adopting the design values of pressure and temperature, the safety factor is inherently introduced into the mechanical design of both pressure vessels and piping systems.

Design pressure

Pressure vessels and piping systems have to be designed to withstand the maximal pressure which can occur under any possible condition in the process system. If a pressure relief device is installed in the system, the design pressure is typically taken as the pressure at which the relief device is set (i.e. set pressure). Set pressure will normally be up to 10% higher than the normal working pressure, in order to avoid the possibility of incorrect operation in case of minor process disturbances.

For example, the API 520 recommends a 10% mar–gin between the normal operating pressure and the set pressure of relief device [11]. When the design pressure is determined, the hydrostatic pressure at the base of the vessel should be added to the operating pressure, if significant [8].

System design pressure can also be determined by the installed equipment. Another example of design pressure definition which is often met in the engineering practice is the shut–off head of a centrifugal pump.

Design pressure is not to be confused with the so called maximal allowable working pressure or MAWP. By definition (ASME Section I), the MAWP is the "pressure determined by employing the allowable stress values, design rules and dimensions". According to this definition, MAWP is the highest level of pressure that a piece of equipment *could be* exposed to, while design pressure is the highest level of pressure it *should be* exposed to in normal operating conditions. In other words, the design pressure should always be lower than, or at maximum equal to, the MAWP.

Systems subject to external pressure should be designed to resist the maximal differential pressure that is likely to occur in service. Systems which are likely to be subjected to vacuum should be designed for a full negative pressure of 1 bar, unless fitted with an effective and reliable, vacuum breaker [8].

Based on more than a few sources (engineering handbooks and design guidelines issued by various companies), the design gauge pressure of a pressure vessel can be adopted by using the following rule

$$p_{des} = \begin{cases} 3.5\ \text{barG} & \text{for}\ p_{ope} = -1 \div 2.5\ \text{barG} \\ \max(0.9 + 1.05 \cdot p_{ope}; p_{set}) & \text{for}\ p_{ope} > 2.5\ \text{barG} \end{cases} \tag{7}$$

in which:

- p^{des}, barG, design gauge pressure;
- p^{ope}, barG, maximal operating gauge pressure;
- p^{set}, barG, set pressure of the relief device.

In [9] the following rule is set: pressure retaining parts shall be designed for a pressure at least 8% or minimaly 3.5 bar above maximal operating pressure.

Design temperature

The allowable stress and other properties of vessel and piping construction materials depend on the tempe–rature. This suggests that the design temperature should be considered to be at least equal to the maximal wor–king temperature which can occur under any circum–stance, with reasonable allowance for uncertainties. Typically, for materials that are not included in standardization, the maximal temperature at which the material can be used is limited to the 2/3 of its melting temperature [10].

When the lowest operating temperature expected in service is lower than ambient (e.g. due to the process requirements, operational upsets, ambient conditions etc.), steels can become prone to brittle fracture. This minimal value of the temperature used in the design is typically denoted as minimum design metal temperature or MDMT. In such cases, it may prove that the impact testing is required in order to ensure that no brittle fracture occurs during the system operation.

Standard [9] defines that higher design temperature shall be 10°C above maximal operating temperature, and lower design temperature shall be 5°C below minimal operating temperature. Based on several other sources, the design temperature can be adopted according to

$$t_{des} = \begin{cases} -6 - 0.95 \cdot \left| t_{ope} \right|^{1.025} & \text{for } t_{ope} \le 0°\text{C} \\ 10 + 0.9 \cdot t_{ope}^{1.025} & \text{for } t_{ope} \ge 0°\text{C} \end{cases} \quad (8)$$

in which:
- t_{des}, °C, design temperature;
- t_{ope}, °C, maximal operating temperature.

Vessels which will operate at temperatures lower than –60°C or higher than 400°C should be designed for the minimal or maximal anticipated operating tempe–rature – in such cases design temperature is equal to operating temperature $t_{des} = t_{ope}$.

2.3.3 Simultaneously acting loads on pressure vessels

Pressure vessel has to be able withstand stresses that cause elastic or even plastic deformations under all pos–sible loading conditions and include the effect of field hydrostatic tests, wind, explosion blast pressure, acce–leration, connected piping, transportation and instal–lation.

The loads to which a vessel is exposed can be clas–sified as either (see Table 4):
- major loads, which should always be considered in design, and
- subsidiary loads.

Table 4 Major and subsidiary loads acting on pressure vessels [8]

Major loads	Subsidiary loads
Design pressure: including any significant static head of liquid	Local stresses caused by supports, internal structures and connecting piping
Maximal weight of the vessel (including insulation, cladding or refractory) and its contents, under operating and hydrostatic test conditions	Bending moments caused by eccentricity of the center of the working pressure relative to the neutral axis of the vessel
Wind loads	Shock loads caused by water hammer or by surging of the vessel contents
Earthquake (seismic) loads	Stresses due to temperature differences
Loads supported by, or reacting on, the vessel	Loads caused by fluctuations in temperature and pressure

It should be noted that a vessel is typically not exposed to all of the numbered loads at the same time, and it is the task of the designer to determine the worst combination of simultaneously acting loads.

For example, according to [9] lifting lugs have to be designed with a safety factor of 2.

It is often seen in engineering practice that two distinct design teams work independently on design of pressure vessels and connecting piping systems. Since the timeline is a very important factor in design process, there is often no time for convergence between the multiple design teams. The easiest way to avoid potential operational and safety issues is the agreement between both teams to respect the same recommendations for nozzle loads. Consequence of this approach is that an additional safety factor is introduced in the design. Besides the recommendations given by certain companies for strengthening of nozzles on pressuriyed equipment, there are also several standards that define the loads on the vessels, such as the API 560, API 661, TEMA and few others.

2.3.4 Piping system components stress intensification and flexibility

Much as pressure vessels, piping systems must with–stand the stresses which cause both elastic and plastic deformations over their lifetime, without causing catas–trophic failure (e.g. burst or fatigue). However, effects of either sustained (e.g. weight and pressure) and self–limiting loads (e.g. thermal expansion) on the generated stresses and strains in the piping system components are generally not straightforward. This is especially true for discontinuity locations such as bends or tees. These locations, due to their inherent construction, result in stress concentration and increase with respect to the straight pipe, while they are also often more flexible than the straight pipe of equivalent length. These effects are taken via the inclusion of the so–called stress intensification factors, stress indices and flexibility factors, which can be directly adopted by using formulae from design codes (e.g. ASME B31.1, ASME B31J, EN 13480–3 etc.).

However, although the inclusion of these factors should inherently contribute to the inclusion of safety factors, it was shown that they do not always result in conservative stress estimates [12]. Therefore, the desig–ner should be very cautious when selecting the approp–riate stress intensification and flexibility factors.

3. SAFETY FACTOR FOR TECHNOLOGICAL AND MECHANICAL DESIGN OF PROCESS, ENERGY, AND HVACPLANTS AND EQUIPMENT

Safety factor is used during both process and mecha–nical design in order to cover the possible deviations in plant or equipment operation from the conditions pre–sumed in the design specification. Application of safety factor results in satisfactory system productivity even under variable operating conditions (within the expected margins). Let us consider an example of distillation column and possible energy bearing fluid variations. In distillation system the pressure of steam (and its related temperature) used in the reboiler directly influences column operation and the operation of the complete distillation system. Same can be said for the temperature and flowrate of cooling water in column condenser. These deviations from the design point must be covered by an appropriate safety factor.

Safety factor is typically given a lot less attention than the design safety factor in the professional literature. For this reason, the further text shall focus on the use of safety factor in the development of the technical documentation, or more precisely in plant and equipment design.

It is common practice for many major companies to define the safety factor (either directly or indirectly) in project contract or in the design specification. If the safety factor is defined in this way, the design engineer's job is made easier because he or she does not have to suggest or adopt its value. For example, one of common requests with this respect is to design the entire plant by considering the margin of 10% on the flowrates, while keeping all the other process conditions (pressures, temperatures, concentrations etc.) unvaried. In this way all the technological systems and sub–systems are oversized uniformly throughout the plant.

Safety factor can also be defined for single machine, apparatus, tank, pressure vessel, pipe etc. This safety factor ensures normal system operation even with fluctuations of both quantity and quality of the raw material (composition, temperature, pressure, flowrate) as it stabilizes process flow quality within the first couple pieces of equipment within the technological line. In this way, the stabilized flow is sent towards the rest of the process after this first "stabilizing" stage. For example, the aim of fermentation of starchy raw materials (corn, wheat etc.) is production of liquid ethanol with the by–product of gaseous carbon–dioxide containing fumes of water, ethanol and other alcohols, aldehydes, ketones etc. The amount of carbon–dioxide, as well as composition of this gas mixture, depend greatly on the fermentation parameters (type of yeast, fermentation temperature etc.). The first apparatus that treats this gaseous mixture is water absorber which strips most of the alcohol fumes into water, which in term yields significantly better gas composition at its outlet (less impurities in carbon–dioxide). If the absorber is adequately oversized, the carbon–dioxide capture process downstream of it will operate under relatively stable conditions, regardless of all the disturbances resulting from fermentation process itself.

Introduction and application of the safety factor in plant lifetime starts with the clients request and is subsequently applied by engineers and designers, equipment manufacturers, erection company and other subjects involved in plant design, erection, commissioning and finally operation.

3.1 Clients design specification

Client should define certain capacity margin during the design or revamp planning in order to be able to increase plant productivity in the future. This is neither quick nor simple task knowing that it requires market analysis and forecast for the next 5 to 10 years [8]. For the engineering team working on this task special issue represents cooperation with economy and marketing specialists who usually do not have adequate technological background and so form requests which are hard to be defined technically. This, in fact, is one of the main reasons for introduction of the safety factor: the client lacks a clear picture of what he wants both in term of final product quantity and quality.

In order to provide a safeguard from the possible errors in the contract, or practically from himself, the client in these situations expects the vendor or consultant to suggest plant parameters such as: plant capacity, final product quality, plant safety, capital expenditures, operating expenses etc. After reviewing the vast amount of (often contrasting) information from various vendors, the client will usually not be able to define exact guidelines for the further work. Wanting to avoid any responsibility in this phase of plant lifetime, the client will introduce safety factors, often following "the more the better" mentality, suggesting that the plant or equipment will operate better if oversized.

A bad example of this exact mindset is presented in [13] which states that the normal request is that "the plant should be designed to operate adequately under conditions of reduced throughput (say by 50%), and for increased production (say 25%)". Setting a request as broad as this (minimal to maximal plant capacity ratio of 2.5) is a very challenging requirement for the plant design engineer. If this request is justified by market requirements (current and future), the client should consider investing into two separate plants having a capacity of 60±6% of nominal capacity each. In this way the client will have a more reliable solution and achieve major savings in operating expenses.

Example 1 shows how the previously stated requirement negatively affects the operation of a tray distillation column.

3.2 Designers and fabricators

Next link in plant design are engineers (designers) and fabricators. They too are often guided by the principle "the more the better" since the undersized equipment shall not operate as required which could negatively influence designer or fabricator reputation. This is especially true if the design engineer is aware of the fact that the client is unsure of the supplied input data. Experienced engineers are sometimes able to recognize the critical points troubling the client and proceed with the technical design with relative ease. However, in most cases, and especially if client requests elevated financial guarantees and penalties for the job, an engineer will opt for oversizing both single pieces of equipment and the plant in whole.

Another important factor that must be considered during the design development is that the efficiency of equipment and plant both reduce during their lifetime.

3.3 Equipment manufacturers

Equipment manufacturers, especially those producing equipment which requires specific approvals, often declare somewhat inferior equipment characteristics, thus leaving margin for potential issues related to poor equipment installation and/or operation. For example, consider pumps or fans. Their selection is based on specifically defined conditions, such as water at 20°C or air at ambient pressure and 20°C. If the designer selects a pump or fan for conditions other than these the responsibility for potentially bad selection falls on the designer alone. However, only the inclusion of manufacturer's name in the plant malfunctioning and non–conformity reports is unpleasant to say the least.

Internal safety factors applied by equipment manu–facturers can also be introduced due to uncertainty in the quality of the supplied sub–assemblies or single parts (for example, supply of bolts from a new vendor).

Due to above, it is a common practice of many equipment manufacturers and vendors to provide gua–ranteed equipment capacity or other important parame–ters with a certain margin (typically up to 5%).

3.4 Stock equipment

As a general rule, during plant design and erection the equipment selection shall be based as much as possible on standardized equipment [1]. This means pumps, compressors, valves, mills etc. already sized and desig–ned by the manufacturer which the designer must incor–porate into his own project.

Manufacturers tend to stockpile a certain amount of the predefined size ranges of equipment. In these cases, in order to minimize lead time, equipment may be selected having the first larger capacity than the required (calculated) value thus achieving additional margin. Also, manufacturers tend to give lower price and better guarantees in case of stock equipment than for specific equipment (designed from scratch for the exact application) made in limited quantity [14].

3.5 Erection company

Erection company tends to introduce additional safety factors related to the local site conditions: weather, lack of resources (e.g. adequate lighting) or again due to uncertainties related to the fabrication materials (such as welding electrodes, flanges, gaskets etc.).

3.6 Plant operation and maintenance

Plant operation and maintenance personnel typically have their own view on equipment and process mana–gement. In their view, the least problems may be expec–ted from mechanically oversized equipment since it will not result in production stops and unplanned shut–downs. This way of thinking is particularly often applied to relatively simple and inexpensive equipment whose failure may lead to plant shut–down. Typical example are pipe elbows and valves used in hydraulic and pneumatic solids transport, since these are subject to a significantly higher levels of abrasion than the straight pipe segments.

3.7 Modification of process conditions

Certain deviations, either foreseen or unforeseen, bet–ween the actual and design conditions can always be expected in a process, energy, and HVACplants. Pro–cess conditions for each plant system and the entire plant may vary on a daily basis based on business politics which dictates operating process parameters. Also, there may be frequent start–ups and shut–downs affecting significantly process parameters.

Plant start–up is especially critical since, due to ignorance or lack of attention, there can be occurrences of significant flowrate peaks resulting in excessive fluid flow velocities. Problems such as abrasion, erosion, column flooding and vibrations (of pipes, tube bundles etc.) can be extremely dangerous [15]. During plant shut–down the reduced flowrates and resulting flow velocities may, for example, result in intensive fouling (of heat exchange surfaces, column trays etc.) due to particle sedimentation.

Variations in process conditions may be also caused by varying atmospheric conditions. For example, ambi–ent air temperature always varies based on weather conditions, which influences the heat duty of air-cooled condensers and water cooling towers [16].

3.8 Critical plant sections

Some plant sections may be fundamental for correct plant operation, and so are called critical sections (or even critical equipment). In other words, the term "critical equipment" includes all equipment which dictate plant performances and safety.

These pieces of equipment are intentionally over–sized regardless of varying operating conditions due to their obvious influence on the correct plant operation.

Part of this "critical equipment" is doubled during the early design, thus creating parallel connection bet–ween them to provide back–up in case of malfunction. In this way, even if one equipment fails, the other will take over, thus ensuring smooth plant operation. Basic example with this respect is rotating machinery: in order to provide continuous flowrate, pumps are typically installed as operating (one or more) and back–up (at least one), thus ensuring continuous flowrate of fluid. In this way the costs of pump procurement are multiplied, but is achieved plant continuity even in case of single pump failure. Other possibility, for example, may be to install 3 parallel pumps (2 operating and 1 stand–by) instead of 2 (one operating and one stand–by) having capacity 50% of design capacity each. In this way the capital expenditure remains more–or–less unvaried with increased operating flexibility and overall reliability.

3.9 Calculation inaccuracy

In all natural sciences, such as physics and chemistry, as well as in engineering disciplines including mechanical, civil, electrical, and chemical engineering, models are routinely used to analyze and interpret the behavior of real–world systems and phenomena. In general terms, a model may be defined as a simplified or idealized representation of a system or phenomenon that captures the essential characteristics of physical reality while neglecting less significant details in order to enable analysis and prediction [17]. According to Eykhoff [18], a model is described as "a representation of the essential aspects of an existing system (or a system to be constructed) which presents knowledge of that system in usable form". Because real systems are inherently complex and involve numerous interacting variables and external influences, it is practically impossible to account for all factors governing their behavior. Conse–quently, any model represents only an approximation of reality rather than its complete description. In practice, a model is considered acceptable if it adequately repro–duces the essential behavior of the system under consi–deration and provides a reasonable balance between practical applicability and fidelity to the physical reality it is intended to represent.

In the same time, this means that the engineering calculation and design are not completely exact tasks. This is true even today, in spite of the fact that various analytical and numerical procedures have been deve–loped for calculation and sizing of equipment, machi–nes, piping systems, steel structures etc. In other words, the data is often used in the engineering practice (equ–ations, recommendations, approximations etc.) which does not have sufficiently sound background to become a general rule. For example, a 10% margin may be retained due to uncertainties in evaluation of physical properties [19].

One of the classical examples with this respect are the criterial equations. These are equations based on similarity theory expressed in form of dimensionless numbers which are obtained by statistically analyzing experimental results. Experimental work by definition includes a small number of physical models (often only one), and is very often performed for a single operating fluid under limited conditions set by the experimental set–up itself. Even in these limited experimental condi–tions statistical analysis yields equations with significant deviations. For example, [20] states that the deviations up to 25% are not uncommon in experimental analysis.

First example to demonstrate this is the research of heat transfer during the water flow in the shell side of heat exchangers with helical tube coils described in [21]. Statistical analysis of the experimental data (based on 3 different heat exchangers, 44 working regimes, water as operating fluid and the temperature range of 21÷73°C) yielded the following criterial equation

$$\mathrm{Nu} = 0.50 \cdot \mathrm{Re}^{0.55} \cdot \mathrm{Pr}^{1/3} \cdot (\eta/\eta_w)^{0.14} \tag{9}$$

for the following range of the important parameters:

- Reynolds number Re = 1000÷9000;
- Prandtl number Pr = 2.6÷6;
- hydraulic diameter dh = 9.1÷18.3 mm.

Paper authors claim that the correlation ratio of the above equation is CR = 94.7% and that the mean square deviation is SD = 8.25%. Based on these statistical parameters it can be concluded that the proposed equa–tion provides a very good fit to the experimental data and that it can safely be used in the engineering practice.

On the other hand, the proposed equation, as well as many other correlations, is obtained by analyzing a single operating fluid, and so its application to the other fluids (for example for oils having significantly higher viscosity than water) remains questionable. Also, the question remains what will happen when the operating parameters are outside of the indicated values (for example Reynolds' numbers outside of the range 1000÷9000, exchangers having different hydraulic diameters etc.). The following data are cited from [15] and shows calculation inaccuracy related to the heat transfer coefficient for single phase fluid flow in heat exchangers:

- shell and tube – tube side – inaccuracy ± 10%;
- shell and tube – shell side – inaccuracy ± 20 ÷ 50%;
- plate – inaccuracy ± 10 ÷ 30%.

Unlike equation (3.1) in the open literature there are a lot of equations having less than compelling statistical parameters. For example, the often cited and wide–spread book [22] gives the following equation for calculation of the clear liquid height on a sieve tray of a column

$$h_L = (1-\varepsilon)\cdot\left\{H_w + 0.731\cdot\left[\frac{V_L}{B_w\cdot c_d\cdot(1-\varepsilon)}\right]^{2/3}\right\} \tag{10}$$

Above equation is given without stating its limita–tions nor its statistical parameters. This very equation was put to a test on a column having diameter 314 mm in [23] and it has been shown by comparing its values with the actual measurements that its statistical parame–ters are very poor (CR = 0 and SD = 57.2%). In other words, the values obtained by using (3.2) have proved to be very different from the actual experimental data.

There are many examples as the one shown above, which points to the fact that during the design due attention should be given to the problem at hand. This also implies that the calculations should include certain safety factors in order to obtain a reliable technical solution.

As already demonstrated, equations presented in the open literature can have less than satisfactory statistical parameters, and so it is evident that their use will yield certain deviation from the realistic behavior. Same is valid for the engineering recommendations – for exam–ple, recommendations for fluid flow velocity in pipes are often outdated and based on the techno–economic analysis not applicable in today's industry [24–25].

Examples 2 and 3 illustrate the influence of corre–lation inaccuracies on the accuracy of the calculations.

3.10 Recommended values

Sometimes during the engineering (calculation) phase recommended or approximate values based on literature or experience are used. Basic characteristic of all recommendations is that they are all approximate and based on experience, which may not even be practice–related but based on partial numerical calculations. In order to be sure of the validity of calculations based on recommended values, a safety margin of up to 50% is used. Degree of the safety margin is in close connection with the quality of the applied recommendation.

3.11 Pilot plants and scaling problem

If design engineer is not in possession of reliable (empirical) data, or if the necessary data is not readily available in the open literature, then there is no adequate and sound basis for equipment sizing, engineering calculations and design. In these cases, it may prove necessary to build a pilot (i.e. experimental) plant with the aim to collect the necessary data. Before being used in an actual project, the results obtained on the pilot plant need to be scaled–up to the actual plant capacity. In these cases, process, energy, and HVACengineers need to be familiar with the limitations of scaling met–hods and need to know how to best select the variables necessary for equipment sizing.

Pilot plant data is almost always necessary for sizing biochemical reactors and filters for removal of solids from various fluids, unless accurate empirical data is at the designer's disposal. On the other hand, heat exc-hangers, distillation columns, pumps, as well as many other conventional equipment can typically be calcu–lated with sufficient accuracy even without pilot plants. Table 5 presents an analysis of the important factors for data scaling [14].

Table 5 Scaling factor and safety factor

Type of equipment	Key variables (excluding flowrate)	Key variables describing equipment dimensions and capacity	Scaling factor	Safety factor %
Pilot plant is necessary				
Batch crystallizer with mixer	Solubility	Flowrate Heat exchange surface area	> 100 : 1	20
Batch reactor	Capacity Balance	Volume Holding time	> 100 : 1	20
Continuous reactor	Capacity Balance	Flowrate Holding time	> 100 : 1	20
Rotating filter	Filter cake pressure drop and permeability	Flowrate Filtration surface area	> 100 : 1 25 : 1	20
Pilot plant is not necessary				
Centrifugal pump	Head	Flowrate Power Impeller diameter	> 100 : 1 > 100 : 1 10 : 1	10
Tray or packed column	Balance Flow velocity	Flowrate Diameter Height	> 100 : 1 10 : 1	15
Shell and tube heat exchanger	Temperatures Viscosity Thermal conductivity	Flowrate Heat exchange surface area	> 100 : 1 > 100 : 1	15
Cyclone separator	Particle size	Flowrate Diameter	10 : 1 3 : 1	10

4. OVERVIEW OF SAFETY FACTOR RECOMMENDATIONS

This section presents an overview of various literature sources related to the safety factor (some of the recom–mendations have already been mentioned in the pre–vious sections). In general, recommendations presented in the literature can be divided into two major cate–gories:

- recommendations for the entire plant design;
- recommendations for single pieces of equipment.

4.1 Plants

Recommendations for the overall plant design are typically related to the increase of flowrates throughout the entire process section. In [8] the authors present general recommendations for selection of safety factors for the overall plant design based on the level of deve–lopment of the technological process. These recom–mendations are as follows:

- new and insufficiently established process SM = 15%;
- relatively new process SM = 10%;
- redesigned process SM = 7%;
- licensed process SM = 5%;
- established process SM = 3%.

In [26] it is stated that the LPG plants, considering the today's level of knowledge of the process, should typically be designed considering the safety margin of SM = 8÷12%, which is significantly less than the safety margin used for the plants built approximately 30 years ago which was SM = 35÷40%.

4.2 Critical equipment

For some critical heat transfer equipment (e.g. boilers and heat exchangers), if leakage occurs due to failure of a single tube of a tube bundle, it can be plugged. In this way, if the equipment is oversized, the intervention will not influence the overall plant productivity [27].

Below are given two examples showing some of the possible criticalities related to heat exchangers:

- if a heat exchanger (cooler) is unable to maintain safe operating conditions for an exothermic che–mical reaction, it may compromise plant safety and, in the worst-case scenario, lead to an uncontrolled reaction, characterized by a continuously increasing reaction rate and temperature, potentially resulting in an explosion;
- evaporators and condensers on distillation columns have direct influence on plant capacity and/or phase compositions at the column outlet.

Of course, there are many other examples all of which have in common that the "critical" equipment is sized considering greater safety margin than the rest of the plant. However, design engineer also needs to be aware that excessive oversizing may result in issues related with decreased flow velocity, such as increased fouling of heat exchange surfaces [15], transition from turbulent to laminar flow regime etc.

4.3 Piping systems

Besides the safety factors regarding the transport capa–city, piping systems need to include an additional factor – ageing or age factor.

4.3.1 Ageing factor

Piping systems are exposed to corrosion, abrasion and sedimentation of dirt and scale. From flow resistance point of view, the state of pipe surface deteriorates over time due to the mentioned phenomena. Designers need to take into consideration this fact by employing a certain reserve typically called ageing or age factor. This factor takes into account the following conse–quences of surface deterioration over time:

- due to sedimentation the flow section of the pipe reduces which results in increased flow velocity and pressure drop (considering that the flowrate remains unchanged);
- roughness of the pipe walls increases (or more precisely of the contact surface between fluid and pipe wall/scale) which once again results in the increased pressure drop.

In [28] there is a general recommendation that the calculated friction factor (ξ) should include a safety margin of SM_{ξ} = 10÷30% (SF_{ξ} = 1.2÷1.3). This additional margin should cover the change in pipe roughness of steel pipes over the period of 5 to 10 years under normal operating conditions. This age factor concerns only friction factor and is employed independently of the safety factor for increased pipe flowrate.

It should be noted that under single–phase turbulent pipe flow the reduction of flow area by 10% yields an increase in pressure drop of ~30% under the constant flowrate. On the other hand, if the pressure drop is constant the same reduction of the flow area results in a decrease of flowrate by ~13%. In Example 4, this is shown for the case of rough pipe flow.

4.3.2 Safety factor for piping systems

Safety factor for piping systems, in terms of pipe flow, refers to the increase of the transport capacity of the pipeline and is employed by increasing the flowrate with respect to the design value. Reference [28] gives a recommendation that the safety margin on the flowrate should be adopted as SM_V = 10÷20%. Same sources recommend that the minor losses be calculated by using conservative tables from [29] as these provide good results for long–term plant operation.

4.4 Pumps, fans, blowers and compressors

Chapter of [29] regarding the flow machines states that the selection of fans, blowers and compressors should be made by considering a flowrate increased by 5% (SM_V = 5%). Identical recommendation is given in [30].

For pumps, the recommended safety margin, according to [31], is SM_V = 10%. However, the designer must has to be certain to select the pump which shall operate satisfactorily under realistic operating conditions (i.e. not too far away from the best efficiency point of the pump and above the minimal safe flowrate for the selected pump).

One additional possibility to provide safety margin for pumps is to select a pump with an impeller sufficiently below the maximal impeller diameter for the selected pump. In this way, in case of under–sizing or change in operating conditions of the plant/piping system, only the pump impeller may be changed, without the need of replacing the entire pump. Naturally, in this case the installed power (i.e. motor) must be able to cover all the possible conditions.

4.5 Heat exchangers

Much like piping systems, heat exchangers often result in confusion when the correct definition of safety factor is concerned. For a correct understanding of the problem, it is necessary to separate the safety margin referring to the fouling of heat exchange surfaces and safety factor regarding the increase of heat duty which provides a margin with reference to the nominal capacity.

Heat transfer coefficient (W/(m^2·K)) based on which the heat exchange surface area is determined, can be expressed in several different ways:

- by considering the clean heat transfer coefficient (k^c) which does not take into account the fouling;
- by considering the service heat transfer coefficient (k^s) which takes into account the fouling;
- by considering the design heat transfer coefficient (k^d) which is obtained once the final exchanger sizing is performed (based on the final heat exchange surface area).

According to [32], the overall increase of heat exchange surface area which takes into account the design and calculation inaccuracies and uncertainties as well as operating issues is 70÷80%. Out of this number, approximately 30÷50% can be assigned to the heat exchanger fouling.

4.5.1 Heat exchanger fouling factor

Heat exchanger fouling is a dedicated topic and is often wrongly interpreted in various literature sources. This fact has in part been transferred to the engineering practice. Within this paper the fouling phenomenon will not be discussed in details as, for example, in [1] or [33]. However, it is necessary to provide the reader with basic explanations in order to obtain a clear insight into the engineer's responsibility regarding this matter.

Overall heat transfer resistance (value reciprocal to k_s) is composed of the resistances existing for the clean heat exchangers (value reciprocal to k_c) and resistances due to the fouling of heat exchange surfaces (so–called fouling factors) which are determined based on the operating fluid. Based on above, the overall heat transfer resistance can be expressed as

$$\frac{1}{k_s} = \frac{1}{k_c} + R_h + R_c \tag{10}$$

in which R_h and R_c (m^2·K/W) denote fouling factors for hot and cold fluid sides, respectively. Fouling factors can range from minor values (order of magnitude 1/10000 m^2·K/W) characteristic for fluids which do not generate sediment or scale (e.g. distilled water) to very important values (order of magnitude 1/500 m^2·K/W) for fluids susceptible to solids sedimentation (e.g. residue from vacuum distillation of oil).

Fouling must be taken into consideration for sizing of any heat exchanger [1].

4.5.2 Safety factor for heat exchangers

Safety factor for heat exchangers is based on heat transfer coefficient, and so it inherently takes into account also the fouling factors

$$SF_k = \frac{k_s}{k_d} \tag{11}$$

This fact is explicitly stated in several literature sources, such as [34–35]. Safety factor, as defined above, takes into account all the inaccuracies described in Section 3.9, but if an engineer is certain of his technical solution, he can also disregard the safety factor (i.e. use $SF_k = 1$). In [36] a heat exchangers safety margin of $SM_k = 15\%$ is recommended, in [37] the range SM_k = 10–30% is recommended, while [35] recom–mends $SM_k = 10\%$. According to [15], the safety margin is recommended as $SM_k \leq 20\%$, and even greater values can be applied in cases of laminar tube flow, mist flow boiling and condensation on the shell side in case of heavy gravitational downpour of condensate.

If a heat exchanger is relatively small, a 10% or even a 20% overdesign may not be objectionable because it is the extra absolute cost and not the percentage that is significant; for larger heat exchangers, a smaller over–design, such as a 5% margin, may be more prudent [38].

Besides the equation (4.4), safety factor can also be defined by increasing the flowrate or required heat duty of the exchanger

$$SF_Q = \frac{Q_s}{Q_d} \tag{12}$$

in which Q_s and Q_d (W) denote heat duties based on k_s and k_d, respectively. In [19], [31] and [38] a safety margin of $SM_Q = 10\%$ is generally recommended.

It is also worthwhile to mention that according to [34], for shell and tube heat exchangers the safety factor needs to be ensured by increasing the length of the tube bundle, and not by increasing the diameter of the shell.

When discussing the sizing of shell and tube heat exchangers it is also important to note that once the sizing is performed, a specification sheet is issued. This document represents the basis on which the rest of the technical documentation is ordered. Based on the standardized diameters and tube lengths, heads and other constructive elements the fabricator may provide even greater safety margin in terms of greater heat exchange surface area (S, m^2) [14]. According to [31], this unaccounted margin is typically $SM_S = 5\div10\%$.

Example 5 discusses the additional safety factor that is sometimes included by the heat exchanger manufac–turer. In contrast, Example 6 illustrates a poor inter–pretation of the effect of increasing the exchanger surface area on its capacity.

Finally, it needs to be mentioned that for the complex process, energy, and HVACsystems containing several interconnected heat exchangers, an application of excessive safety factor may result in substantial difficulties during plant operation – example 7 shows one simple case.

4.6 Columns for gas–liquid systems

These include distillation, absorption and desorption columns, which are typically defined by two distinct safety factors. The first reflects column height and can be expressed based on the number of trays (Nt):

$$SF_{Nt} = \frac{Nt_d}{Nt_c} \tag{13}$$

in which Nt_c denotes the required number of trays obtained by calculation and Nt_d the adopted number of trays. Minimal safety factor of SF_{Nt} = 110% is recom–mended in [39] regardless of the calculation method. It is also recommended that the flowrate of the reflux pumps be increased by 25%. In [16] it is stated that a minimal safety margin of $SM_{Nt} = 5\div10\%$ is desirable. Based on research (experimental measurements and design calculations) on 5 semi–industrial columns, an uncertainty factor of 7÷16% is determined in [40], and so based on this reference, the recommended value of the safety margin is $SM_{Nt} = 7\div16\%$.

Second safety factor related to these columns is defined based on the flowrates as

$$SF_V = \frac{V_d}{V_c} \tag{14}$$

in which V_c and V_d denote calculated and increased phase flowrates in the column, respectively. In [31] it is stated that the usually adopted safety factor is $SF_V = 110\%$.

When applying safety factors SM_{Nt} and SF_V the designer must not forget that either one of these factors has direct influence on the related heat exchangers (eva–porators and condensers) as well as on the reflux pump.

4.7 Regulation valves

According to [31] regulation valves should be selected based on the safety margin on the flowrate of appro–ximately $SM_V = 20\%$.

5. RELIABILITY-BASED DESIGN AND PROBABI–LITIC RISK ASSESSMENT AS A FRAMEWORK FOR INTERPRETING SAFETY AND DESIGN FACTORS

Reliability-Based Design (RBD) ensures that technical systems achieve performance targets despite inherent variability. Rolling-element bearings, for example, are rated for a nominal service life, but a small fraction may fail earlier, justifying the use of a reliability factor - the probability that the system operates for the intended life. Similarly, process, energy, and HVACequipment such as pumps or heat exchangers are rated for nominal capacity, but manufacturing tolerances or operating conditions can reduce their actual output. Incorporating reliability into design allows overall system perfor–mance to be maintained even if some components un–derperform. RBD integrates probabilistic approaches into engineering, linking traditional design or safety factors with quantified system reliability [41,42]. Historically, safety factors were applied as deterministic margins to accommodate uncertainties in materials, loads, and operating conditions [43]. Modern RBD interprets these margins probabilistically, e.g., a 10–20% nominal margin may correspond to a reliability index $\beta \approx 2$–2.5 [44].

Probabilistic Risk Assessment (PRA) extends this framework to complex systems, quantifying failure likelihoods and incorporating component-level uncerta–inties into system-level models [45,46]. Applications include optimizing maintenance, predicting failure rates, and ensuring safe operation in modern industrial sys–

tems, including collaborative robotics [47]. By combining design margins, reliability indices, and probabilistic modeling, engineers achieve a coherent framework for robust, safe, and cost-effective design.

5.1 Application of Probabilistic Risk Assessment to Safety and Design Factors

Within the PRA framework, risk is defined as the product of the probability of an undesirable event and the magnitude of its consequences

$$R = P \cdot C \tag{15}$$

where P denotes the probability of occurrence of a given scenario and C represents the associated consequences. In reliability analysis, the probability of failure g is commonly formulated using a limit-state function

$$g(X) = R(X) - S(X) \tag{16}$$

where R denotes the resistance or capacity of a system and S represents the applied load or demand. Failure occurs when

$$g(X) \leq 0 \tag{17}$$

In probabilistic reliability analysis both quantities are treated as random variables characterized by their mean values μ_R and μ_S, and standard deviations $\sigma_R$and σ_S. A convenient measure of reliability is the reliability index

$$\beta = \frac{\mu_R - \mu_S}{\sqrt{\sigma_R^2 + \sigma_S^2}} \tag{18}$$

which represents the normalized distance between the mean operating point and the failure boundary. The corresponding probability of failure can be approximated as:

$$P_f = \Phi(-\beta) \tag{19}$$

where Φ denotes the cumulative distribution function of the standard normal distribution.

Within this framework, the conventional deterministic factor of safety

$$n = \frac{R}{S} \tag{20}$$

can be interpreted as a simplified deterministic representation of a probabilistic reliability requirement. Introducing the coefficients of variation

$$V_R = \frac{\sigma_R}{\mu_R} \tag{21}$$

and

$$V_S = \frac{\sigma_S}{\mu_S} \tag{22}$$

an approximate relationship between the safety factor and the reliability index may be written as

$$n = 1 + \beta \cdot \sqrt{V_R^2 + V_S^2} \tag{23}$$

This relation indicates that the magnitude of a deterministic safety factor implicitly reflects the statistical variability of loads and resistances as well as the targeted reliability level. In many contemporary design codes the prescribed safety factors result from calibration procedures based on statistical data for loads, material properties, and acceptable levels of structural risk. Consequently, PRA does not replace deterministic design rules; rather, it provides a probabilistic interpretation of safety factors and explains their calibration with respect to acceptable probabilities of failure [48,49].

As said before, in addition to mandatory safety factors, engineers frequently introduce design margins in the design of process and energy systems. These margins are not primarily intended to prevent structural failure but rather to ensure reliable system performance under realistic operating conditions characterized by uncertainties in process parameters, hydraulic calculations, and operating regimes. Typical examples include additional pump capacity, increased heat-transfer surface in heat exchangers, or oversizing of control valves.

For such situations a functional limit state can be defined as:

$$g(Q) = Q_{\text{available}} - Q_{\text{required}} \tag{24}$$

where $Q_{available}$ represents the available system capacity (e.g., pump flow rate) and $Q_{required}$ denotes the process demand. Functional failure occurs when the available capacity becomes smaller than the required one.

Consider a simple example in which the mean available pump capacity and the required flow rate are $\mu_{Q_a} = 110\ \text{m}^3/\text{h}$ and $\mu_{Q_r} = 100\ \text{m}^3/\text{h}$ with coefficients of variation $V_{Q_a} = 0.05$ and $V_{Q_a} = 0.07$

The corresponding standard deviations are $\sigma_{Q_a} = 5.5$ and $\sigma_{Q_r} = 7$. The resulting functional reliability index becomes

$$\beta_f = \frac{110 - 100}{\sqrt{5.5^2 + 7^2}} = 1.123 \tag{25}$$

and probability of failure to achieve required capacity is $P_f = 0.131$

This value corresponds to a probability of functional failure on the order of 13.1%. Such reliability levels are commonly acceptable in process, energy, and HVAC systems because the consequences of functional failure are typically operational rather than safety-related (e.g., temporary reduction of plant capacity or adjustment of operating conditions).

The example demonstrates that a typical 10% design margin for pump capacity can be interpreted probabilistically as a measure that reduces the likelihood that the system will fail to meet the required process performance due to uncertainties in operating conditions or calculation assumptions.

Therefore, PRA provides a unified conceptual framework for interpreting both categories of design factors encountered in engineering practice. Mandatory safety factors prescribed by standards primarily limit the probability of hazardous structural failure, whereas design margins introduced by engineers ensure stable and reliable system operation under realistic process variability.

From this perspective, the present analysis does not attempt to replace deterministic engineering design pro–

cedures with full probabilistic modeling, but rather to clarify that both standardized safety factors and prac–tical design margins can be consistently interpreted within the broader probabilistic framework of risk and reliability analysis.

Table 6 connects probabilistic reliability indicators (*β, Pf*) with traditional deterministic design/safety fac–tors, making it useful for Reliability-Based Design in mechanical and process systems.

Table 6 Traditional deterministic design factors vs probabilistic reliability indicators

Reliability Index β	Probability of Failure P_f	Design Margin (%)	Notes
0.0	0.50	0	Very low reliability
0.5	0.31	5–6	Low reliability
1.0	0.16	10	Minimally acceptable
1.123	0.131	12	Example case
1.5	0.067	15	Moderately safe
2.0	0.023	20	Typical design margin
2.5	0.006	25	High reliability
3.0	0.0013	30	Extreme safety

5.2 Limitations of RBD in process, energy, and HVACEquipment Design

As discussed in the previous section, Reliability-Based Design (RBD) provides a formal framework to relate design margins to probabilistic measures of failure. Over the past fifty years, RBD has been gradually introduced across various engineering disciplines, inc–luding process, energy, and HVACengineering. Its influence is increasingly evident in equipment safety, plant maintenance, and design, particularly in areas related to accident, fire, and explosion protection.

However, the practical application of RBD for design factors in plant design and equipment selection remains limited due to the lack of reliable statistical data on actual component performance. As a result, proba–bilistic assessments often rely on assumptions or esti–mated parameters, which constrains the ability to fully replace traditional deterministic safety factors.

For pumps, compressors, heat exchangers, and similar industrial units, there are typically no published datasets quantifying the variability of actual capacities, operational loads, or long-term degradation. As a result, calculated reliability indices (β) and probabilities of failure (P_f) are largely based on assumed or estimated parameters, rather than measured distributions. Conse–quently, RBD cannot fully replace classical design factors in these contexts. Nevertheless, it serves as a useful conceptual tool, allowing engineers to rationalize safety margins, perform scenario analyses, and under–stand the relationship between probabilistic reliability and traditional project margins, even when precise probabilistic data are unavailable.

6. CONSEQUENCES OF APPLICATION OF SAFETY FACTOR

Application of safety factor has a direct influence on the investment costs of a plant (i.e. capital expenditures). An oversized plant or even single individual pieces of equipment have a certain margin in terms of capacity with respect to the nominal value. Such plants are inherently more expensive which negatively influences the plant profitability. On the other hand, application of safety factor enables an increase in production capacity over time, should the need arise. Both of these consequences need to be carefully evaluated during the planning and design phase, as well as during the plant operation. Further text will focus on each of these two consequences in detail.

It must not be forgotten that the safety factor also influences plant safety, potentially even negatively. For example, in [50] it is stated that water hammer can sometimes occur due to the oversized pipe diameters. Although this claim is doubtful, it points out that safety factor can potentially have a negative impact on plant safety.

6.1 Influence of safety factor on plant investment costs

Simplest estimate of capital expenditures of plant or equipment can be made based on the already available investment costs of similar plants or equipment. This is done by employing a method of capacity comparison, and so the capital expenditures can be estimated based on the following equation from [8]:

$$\frac{C_1}{C_2} = \left(\frac{S_1}{S_2}\right)^e \tag{26}$$

in which:

- C_1 and C_2, EUR , capital expenditures of plants 1 and 2, respectively;
- S_1 and S_2, capacities of plants 1 and 2, respectively.

Equation (6.1) can also be applied to single pieces of equipment, by replacing the capacities with some of the equipment–related technical parameters (e.g. a good example would be heat transfer surface area for heat exchangers or installed power for electric motors).

Value of the exponent *e* in above equation is typi–cally in the range 0.6÷0.75 and depends on the type of plant or equipment. If the safety margin based on the level of development of the technological process sho–wn in section 4.1 of this paper is directly and consistently applied to the entire plant, the increase in the capital expenditures shown in Table 7 is obtained.

Table 7 Increase of capital expenditures of a plant based on application of the safety margin depending on level of development of the technological process

Process	*SM*, %	Increase of CAPEX
New and insufficiently established process	15	1.10
Relatively new process	10	1.07
Redesigned process	7	1.05
Licensed process	5	1.03
Established process	3	1.02

6.2 Influence of safety factor on equipment investment costs

Besides having influence on the entire plant, safety factor also influences the cost of single pieces of equip–ment and process subsystems. Influence on various equipment is discussed in further text.

6.2.1 Piping systems

According to [24] an economically optimized pipe diameter for carbon steel pipes can be calculated using the following equation

$$D_{opt} = \sqrt[6.5]{A \cdot B \cdot \left(\xi \cdot \rho \cdot V^3\right)} \qquad (26)$$

By applying the above equation, the safety margin on pipe roughness of 20÷30% can be shown to result in the increase of the optimal pipe diameter of 2.8÷4.1%. Furthermore, application of an additional safety margin on the flowrate of 10÷20% results in additional increase of pipe diameter by 4.8÷8.8%. Finally, application of both of the above safety margins results in the overall diameter increase of 13.4÷36.1% - see example 8.

6.2.2 Pumps, fans, blowers and compressors

If the recommendations listed in Section 4.3 are applied on roughness and flowrate, and not on duct or pipe dia–meter, the increase in pressure drop can be shown to be

$$\begin{aligned} \Delta p &= \left[\left(1.2 \div 1.3\right) \cdot \xi\right] \cdot \frac{L}{D} \cdot \frac{\rho \cdot \left[\left(1.1 \div 1.2\right) \cdot w\right]^2}{2} = \\ &= \left(1.45 \div 1.87\right) \cdot \xi \cdot \frac{L}{D} \cdot \frac{\rho \cdot w^2}{2} \end{aligned} \qquad (27)$$

which means that the application of safety factor results in the increase of pressure drop by 45÷87%. In addition, due to the application of safety margin of 10÷20% on the flowrate, the power consumption of the machine is further increased and becomes

$$\begin{aligned} P_{des} &= \frac{\left(1.45 \div 1.87\right) \cdot \Delta p \cdot \left(1.1 \div 1.2\right) \cdot V}{E} = \\ &= \left(1.60 \div 2.25\right) \cdot \frac{\Delta p \cdot V}{E} \end{aligned} \qquad (28)$$

This means that the application of the recommended safety margins results in the increase of the power consumption of 60÷125% in comparison with the rea–listically required power. If realistic pump prices and above numbers are replaced into equation (6.1) with exponent e = 0.63, according to [10], the application of the recommended safety margins results in the total increase in capital expenditures of 34÷67%.

6.2.3 Heat exchangers

Price of heat exchangers depends predominantly on the heat exchange surface area and an exponent of e = 0.68 is defined in [10] to be used with equation (6.1). When recommendation from [36] is applied and the heat exchange surface is increased by 15% for heat transfer coefficient and by additional 10% for heat duty, the total increase of heat exchanger cost becomes 17%.

6.2.4 Trayed columns

If the recommendation from [39] is respected and the number of trays is increased by 10% together with the recommendation from [31] to increase the flowrate by 10%, based on the column prices defined in [8], the increase of column cost becomes 12÷15%.

In [51] the influence of column oversizing on both capital and operating expenditures has been evaluated in details for several cases of binary systems. Conclusions drawn therein can be summarized in a single recom–mendation that the maximal safety margin for the columns should be SM_{Nt} = 30%.

6.3 Possibility of increase in plant capacity

Once the plant is operational, the owner/manager constantly thinks about several points: what is the realistic plant capacity, what is the demand for the final product, how is the plant amortization proceeding, is there room for increase in plant capacity etc.

In [31] an example is given for a heat exchanger, similar to the one from example 5, which explains what can practically happen if the installed heat exchangers have a significant margin in capacity. Similar conclu–sions can be drawn for pumps and pipelines, as shown in sections 4.3 and 4.4. It is clear that process, energy, and HVACplants designed following the safety margin recommendations typically have a minimal safety margin of 10%, or in other words that the plant can operate with the capacity exceeding 100% the nominal capacity.

These margins can be used once the plant owner decides to revamp his plant in order to increase its capacity. Revamping typically starts by performing a study including a global assessment and identifying major obstacles to obtain an increase in plant capacity. These bottlenecks can be piping or ducting systems having inadequate flow section, heat exchangers having smaller heat transfer surface areas than required, colu–mns which have excessive pressure drop or are ope–rating inadequately under increased flowrates etc. Prob–lems may also occur due to the regulation valves which cannot ensure the required flowrate even when comp–letely open. In most cases, the problem in increasing plant capacity is not reduced to a single issue but to a combination of some or all of the above factors.

6.4 Reliability in plant operation

In general, application of safety factor leads to a certain comfort and convenience in plant operation. However, there are also certain cases which need to be mentioned, in which the excessive safety factor may lead to operating problems. With this respect, the following three examples will be mentioned [52–53]:

- pump is oversizing in terms of head may lead to issues on the downstream equipment not designed to withstand this excessive pressure;
- oversized regulation valve may lead to the irregular flow regulation when the valve is only slightly opened;
- oversized heat exchanger may lead to overheating or excessive cooling of the process fluids (which can be very bad, especially in the food industry). In extreme cases, excessive cooling may even lead to the freezing of the process fluid which results in plant stopping and downtime.

7. CONCLUSIONS AND PERSONAL RECOMMEN–DATIONS

Successfully designed process, energy, and HVAC plants ensure both quality and quantity of the final product. Every piece of equipment must perform its function safely and efficiently. Technical documentation typically includes a safety factor to account for uncer–tainties such as future capacity increases, process cont–rol flexibility, upset conditions, alternative feedstocks, and limitations of process simulations.

In stress analyses, safety factors depend on uncer–tainties in material properties, construction, fabrication, operating conditions, and calculation models. Human factors also play a significant role and must be considered. Technological safety factors address uncer–tainties in parameters not fully defined in project spe–cifications, including operating conditions, calculation models, and market fluctuations. Greater uncertainty requires a larger safety factor. Engineers must evaluate the role of each component, considering the conse–quences of undersizing or malfunction.

Oversizing affects both plant and company profi–tability. Selecting a safety factor requires balancing reliability and economic efficiency. Applied safety factors influence the sizing of equipment, piping, val–ves, and fittings. Once set, safety factors must be clearly indicated in technical documentation to prevent incon–sistent application across engineering teams. Uncoor–dinated use can compromise the entire project. Ideally, safety factors are contractual agreements between clients, designers, and vendors.

Excessive safety factors, sometimes justified by potential capacity increases, can lead to uneconomical oversizing. Each safety factor must be based on con–crete data. Engineers should adjust or disregard safety factors if required by the situation. The chosen factor must ensure capacity and safe operation while avoiding economic inefficiency. Balancing these objectives is complex, making safety factor selection a critical and often underestimated engineering task.

The manuscript provides a comprehensive analysis of safety factors, unprecedented in its scope, connecting each factor to its causes, consequences, and practical application. It integrates classical design approaches with reliability- and risk-based (PRA) perspectives, offering engineers both theoretical and applied guidance.

Key contributions include:

- Comprehensive literature review with precise defi–nitions;
- Methodological advances, including equations for design pressure and temperature;
- Integration of classical safety factors with relia–bility-based interpretations.

In conclusion, this manuscript serves practicing engineers by providing a complete framework to under–stand, select, and justify safety factors. It ensures safe, reliable, and economically optimized plant operation while connecting classical engineering principles with modern risk assessment methods.

After all which has been said it must not be forgotten that, in the end, you as an engineer need to provide the “guarantee” of plant capacity, productivity and safety while achieving an economically optimized technical solution. When all of the above is put together and looked upon, it is clear that this engineering task is by no means simple. At the end, the Authors would like to provide their own recommendations and usual procedure when developing technical documentation:

- client is usually asked if he wants to keep a safety margin of approximately 5% or 10%;
- if we estimate that the client is not confident in the supplied inputdata we apply a minimal safety margin of 5% and often even 10% or more;
- if the client is absolutely certain of the supplied input data (meaning that he has supplied a signed technical specification), we usually apply an inter–nal safety margin of approximately 2% to 5% and keep this value to ourselves, without having it shown in the technical documentation.

ACKNOWLEDGMENT

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ENGINEERS ASSOCIATIONS AND STANDARDS MENTIONED IN TEXT

ASME	American Society of Mechanical Engineers
API	American Petroleum Institute
TEMA	Tubular Exchanger Manufacturers Association
EN	Europen norm or Euronorm

NOMENCLATURE

a_r, m	absolute roughness
AS, N/m^2	allowable stress
B_w, m	weir width
c_p, J/(kg·K)	isobaric specific heat
C_1,C_2, EUR	capital expenditures of plants 1 and 2
CR	correlation ratio
D, m	pipe diameter
DC	design characteristic
d_h, m	hydraulic diameter
h_L, m	clear liquid height
e	exponent
DS, N/m^2	design stress
DSF	design safety factor
E	eficiency
F_{ls}	partial safety factor for the load stress
F_{ft}	partial safety factor for the failure theory
F_g	partial safety factor for the geometry
F_m	partial safety factor for the material
F_r	partial safety factor for the reliability
H_w, m	weir height
k_c,W/(m^2·K)	clean heat transfer coefficient
k_d,W/(m^2·K)	design heat transfer coefficient
k_s,W/(m^2·K)	service heat transfer coefficient
L, m	pipe length
m, kg/s	mass flow rate
MC	maximal characteristic
MS, N/m^2	maximal stresses
Nt_d	adopted number of trays
Nt_c	calculated number of trays
Nu	Nusselt number
p_{des}, barG	design gauge pressure
p_{ope}, barG	maximum operating gauge pressure
p_{set}, barG	set pressure of the relief device
Pr	Prandtl number
Q_d, W	heat duty based on k_d
Q_s, W	heat duty based on k_s
R_c, m^2·K/W	fouling factor for cold fluid side
R_h, m^2·K/W	fouling factor for hot fluid side
Re	Reynolds number
S_1, S_2	capacities of plants 1 and 2
SD	standard deviation
SF	safety factor
SM	margin of safety
t_{des}, °C	design temperature
t_{ope}, °C	maximal operating temperature
V, m^3/s	volumetric flow rate
V_c, m^3/s	calculated phase flow rate in the column
V_d, m^3/s	design phase flow rates in the column
V_L, m^3/s	volumetric flow rate
w, m/s	velocity
Δp, Pa	pressure drop
ε	froth porosity
η, Pa·s	dynamic viscosity
η_w, Pa·s	dynamic viscosity at the wall temperature
ρ, kg/m^3	density
ξ	friction factor

SUBSCRIPTS

max	maximal value
min	minimal value
opt	optimal

APPENDIX – NUMERICAL EXAMPLES

Example 1

The following example demonstrates how the use of excessive capacity margin on trayed columns may actu–ally lead to the reduction of tray efficiency [27] and con–sequently the unproper quality of distillate (ie malfunction of the column).

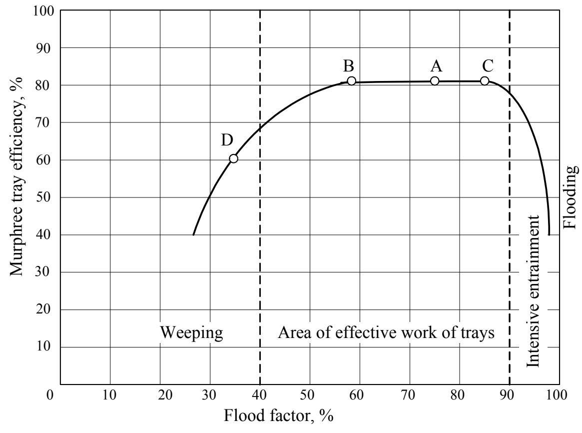

Figure A.1 General diagram of tray efficiency

Figure A.1 shows a simplified representation of dependency of tray efficiency on liquid flooding (ratio of actual gas velocity and gas flooding velocity). Central part of the graph shows practically constant tray efficiency, while in the zones of either excessively high or low gas velocities the efficiency drops sharply. If the column is oversized and is operating at the lower capacity than designed, due to excessively low gas velocity it may be impossible to form a two–phase layer on the tray. This may lead to excessive holdup and liquid weeping through the tray thus drastically reducing tray and column efficiency.

Consequently, the quality of separation process will be significantly reduced.

Column diameter is typically selected so that the actual gas velocity is limited to 65÷75% of flooding point velocity (point A shown in Figure A.1 is at 75% of flooding point). This column diameter allows efficient column operation between points B (at ~70% of point A) and C (at ~115% of point A). In other words, the ratio between the maximal and minimal gas flowrate for which the column can operate with acceptable efficiency is approximately 1.6 and not 2.5 as stated in [13] (point D is the lower capacity limit according to 2.5 ratio).

Example 2

An insight into the inaccuracy of the criterial equations can be shown on an example of helical tube heat exchanger used in the district heating system. In this example, the exchanger is installed in the heating substation and is operating in the temperature regime 140/75°C on the hot side and 70/90°C on the cold side. Exchanger manufacturer has recommended installation of his standard heat exchanger which has heat exchange surface area of 28.8 m^2. Considering equation (31) and a fouling factor of 0.22 $m^2 \cdot K/kW$ (obtained based on the measurements published in [54]) results in the values shown in Table A.1.

Table A.1 Data for Example 2

Quantity	Minimum	Mean	Maximum
Heat duty, kW	915	966	1011
Tube side heat transfer coefficient, W/(m^2·K)	6058	6353	6611
Shell side heat transfer coefficient, W/(m^2·K)	4417	4960	5505
Heat conductivity, W/(m^2·K)	1628	1719	1800
Heat duty difference with respect to mean value, %	–5.3	–	+4.7

Results shown in Table A.1 show that the variation of a single parameter (shell side heat transfer coeffi–cient) within the range ±8.25% yields heat duty vari–ation in the range from –5.3% to +4.7%. If other para–meters were to be examined/varied as well, the range of heat duty (minimal to maximal value) would be even greater.

Example 3

If equation (3.2) is analyzed in detail, it can be shown that even when there is no gas flow (i.e. when $\varepsilon = 0$) and there is no liquid flow (i.e. $V_L = 0$) there is a certain level of liquid on the column trays, which is, of course, nonsense. If, for example, a column having 20 trays has an average pressure drop of 800 Pa per tray (i.e. overall average pressure drop for the entire column 16kPa), using equation (3.2) would result in the calculated pressure drop in the range 11 kPa to 21 kPa. This means that the maximal fan pressure drop obtained by using equation (3.2) is significantly higher than the design value based on the average results, thus resulting in excessive fan and motor oversizing. It is obvious that the form of equation (3.2) is inadequate and that it can lead to significant errors in the engineering calculations and consequently in the design documentation.

Example 4

Some detailed absolute pipe roughness tables contain data the roughness values at different phases of ageing. For example, [55] provides data for carbon steel pipes shown in Table A.2. As it can be seen, during the life–span the absolute roughness (a_r, m) changes within the ratio 100:1. This does not mean that each piping system will result in this drastic increase of absolute roughness over its lifetime, but does raise a flag that there may be a potential problem which needs to be addressed. If the cleaning of piping systems is not performed regularly (and relatively often), it can be expected that the ratio between the minimal and maximal pipe roughness be within the ratio $a_{r,max} / a_{r,min} \approx 10$.

Table A.2 Absolute pipe roughness

State of pipes	Absolute pipe roughness, mm
New	0.04 (0.02 ÷ 0.1)
Cleaned after extensive operation period	0.15 ÷ 0.20
Moderately corroded or fouled	0.40
Extremely corroded or fouled	3.0

In case of rough pipe flow Shifrinson's type of equ–ation for friction factor can be used in modified form [56]

$$\xi = 0.203 \cdot \left(a_r / D\right)^{0.36} \tag{A1}$$

which yields the following ratio between the maximal and minimal friction factors

$$\frac{\xi_{max}}{\xi_{min}} = \left(\frac{a_{r,max}}{a_{r,min}}\right)^{0,36} = 10^{0,36} = 2.29 \tag{A2}$$

On the other hand, according to Weisbach equation pressure drop can be calculated using

$$\Delta p = 8 \cdot \xi \cdot \rho \cdot \frac{L}{D^5} \cdot \frac{V^2}{\pi^2} \tag{A3}$$

For a constant pressure drop minimal absolute rou–ghness corresponds to maximal volumetric flow rate

$$V_{max} = \sqrt{\frac{\pi^2}{8} \cdot \frac{D^5}{L} \cdot \frac{\Delta p}{\xi_{min} \cdot \rho}} \tag{A4}$$

and maximal absolute roughness to minimal flow rate

$$V_{min} = \sqrt{\frac{\pi^2}{8} \cdot \frac{D^5}{L} \cdot \frac{\Delta p}{\xi_{max} \cdot \rho}} \tag{A5}$$

A 10–fold increase of relative pipe roughness (as shown above) results in flowrate reduction (ratio of maximal and minimal flow rate) of 51%.

$$\frac{V_{max}}{V_{min}} = \sqrt{\frac{\xi_{max}}{\xi_{min}}} = \left(\frac{a_{r,max}}{a_{r,min}}\right)^{0,18} = 10^{0,18} = 1.51 \tag{A6}$$

This once again points to the fact that the state of the piping system needs to be accounted for even in the design phase in order to avoid serious issues during plant lifetime and normal operation.

Example 5

Standard tube lengths used in construction of heat exc–hangers are typically 6 m, 12 m or 18m and are relati–vely easily found on the market. If an engineer during the sizing phase adopts and specifies in the specification sheet a tube length of 5.4 m, the manufacturer will most probably use 6 m tubes cut to the desired length for cons–truction of the exchanger. Having in mind the labor costs, an increase of the exchanger length from 5.4 m to 6m may result in a decrease of the overall equipment cost by eliminating the need for cutting the tubes to the desired length. On the other hand, an exchanger with 6 m tubes has not only ~10% greater heat transfer surface area, but also a greater mass. This needs to be taken into account during the erection and construction of the foundation.

Example 6

In the short text [57], the author has managed to de–monstrate fundamental lack of knowledge while dis–cussing the oversizing of heat exchangers. Namely, he states that the increase of heat exchange surface area of 10% is equal to the increase of exchanger capacity (i.e. fluid flowrate) of 20%. With this regard, we cannot resist to mention a problem which has been presented for over 20 years to the students of the department of Process Engineering of the Mechanical Faculty in Belgrade at the introductory session of the subject "Heat transfer operations and equipment". This problem states as follows: In the stationary heat exchanger a 24 kg/s of oil having inlet temperature of 25°C is heated by using saturated steam which is being condensed without sub–cooling of liquid. Steam temperature is 111°C. If the overall heat transfer coefficient equals 65 W/(m^2·K) and the total heat exchange surface area is 300 m^2 determine the heat duty of the exchanger. What would the heat duty of the exchanger be under the same operating conditions if the heat exchange surface is increased by 10%?

Solution is as follows: heat duty is 1.36 MW. If the heat exchange surface area is increased by 10% the heat duty becomes 1.46 MW, which is an increase of 7.4%.

Example 7

Let us consider a simple heat exchanger station shown on the left of Figure A.2. Fluid A is hot technological flow which flows through both exchangers. There are two separate cold technological flows B and C flowing through exchangers E–1 and E–2, respectively. Heat equivalents are $(m{\cdot}c_p)_A$ = 30 kW/K, $(m{\cdot}c_p)_B$ = 40 kW/K and $(m{\cdot}c_p)_C$ = 50 kW/K. Table 4.2 shows the design temperature regime. If both exchangers are oversized by 10%, the values which are also shown in Table A.3 are obtained. Furthermore, let us presume that the exc–hanger E–2 is critical for plant operation.

Based on calculation results, it can be concluded that oversizing of heat exchangers results in the increase of the heat duty of exchanger E–1 by 2.5% with respect to the design value, while the heat duty of exchanger E–2 is decreased by 7.2%. Calculation results are such that an inexperienced observer may easily be misled if presented only with results regarding the heat exchanger E–2. In this case, the exchanger E–2 is in fact undersized, and so the heat transfer surface area must be increased. In order to achieve the required temperature of 110°C a further increase of heat exchange surface area of 46% is needed, resulting in total oversizing with respect to the design value of 56%.

In order to compensate the oversizing of heat exc–hanger E–1 the correct solution is to install a by–pass line which would adjust the heat duty of both exc–hangers to the desired values (Figure A.2 – right).

Table A.3 Values related to example 7

Temperatures, °C	Design operation	Oversized exchangers
t_{A1}	330	
t_{A2}	130	124.9
t_{A3}	85	83.1
t_{B1}	100	
t_{B2}	150	151.3
t_{C1}	75	
t_{C2}	110	107.5

Example 8

If the required flowrate of the cold water through a pipeline is 50 kg/s (13.9 m^3/h) the economically opti–mized pipe diameter based on [24] would be 234 mm. Increasing this diameter by 13.4÷36.1% (values shown above) results in the diameter of 265÷318 mm. In first case the nearest standard pipe diameter is DN250 and in second case DN300 so the application of safety factor leads to the increase of capital expenditures.

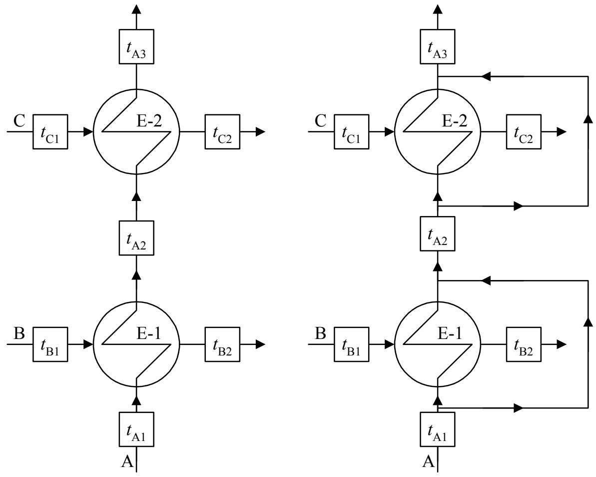

Figure A.2 Heat exchanger station for example 8

ФАКТОРИ СИГУРНОСТИ – ПРИНЦИП КОНЗЕРВАТИВНОСТИ У ПРОЈЕКТОВАЊУ И ИЗГРАДЊИ ПРОЦЕСНИХ, ЕНЕРГЕТСКИХ И HVAC ПОСТРОЈЕЊА И ОПРЕМЕ

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Уопштено речено успешно пројектована процесна, енергетска и термотехничка постројења се одликују истовремено високим квалитетом и задовољава–јућом количином финалног производа. Основни принцип пројектовања сваког процесног, енергет–ског и термотехничког система подразумева да сваки појединачни елемент опреме буде димензи–онисан тако да своју функцију обавља поуздано, безбедно и ефикасно. У том смислу, услед бројних неизвесности, у инжењерској пракси пројектовања и изградње често се примењује фактор сигурности.

Иако је овај појам широко присутан у свакодневној пракси, његов принцип се неретко поједностављује и погрешно интерпретира.
У раду је дат свеобухватан преглед примене фактора сигурности у свим аспектима пројектовања постро–јења и опреме. Опсежан преглед литературе, у ком–бинацији са значајним практичним искуством аутора у овој области, не служи само да читаоцима пружи јасну основу основних принципа и препорука за избор фактора сигурности, већ и да им омогући дубље разумевање свих сложености и утицајних фактора који учествују у његовом дефинисању.
Примери приказани у посебном поглављу имају за циљ да потврде изложене концепте, док препоруке аутора могу послужити као смернице за правилан и рационалан избор фактора сигурности.