IRANIAN PETROLEUM STANDARDS

IPS-E-PR-750 (1)

ENGINEERING STANDARD
FOR
PROCESS DESIGN OF COMPRESSORS

FIRST REVISION
FEBRUARY 2010

DEPUTY MINISTER
OF
ENGINEERING & LOCAL MANUFACTURING
RESEARCH & STANDARDS
FOREWORD

The Iranian Petroleum Standards (IPS) reflect the views of the Iranian Ministry of Petroleum and are intended for use in the oil and gas production facilities, oil refineries, chemical and petrochemical plants, gas handling and processing installations and other such facilities.

IPS is based on internationally acceptable standards and includes selections from the items stipulated in the referenced standards. They are also supplemented by additional requirements and/or modifications based on the experience acquired by the Iranian Petroleum Industry and the local market availability. The options which are not specified in the text of the standards are itemized in data sheet/s, so that, the user can select his appropriate preferences therein.

The IPS standards are therefore expected to be sufficiently flexible so that the users can adapt these standards to their requirements. However, they may not cover every requirement of each project. For such cases, an addendum to IPS Standard shall be prepared by the user which elaborates the particular requirements of the user. This addendum together with the relevant IPS shall form the job specification for the specific project or work.

The IPS is reviewed and up-dated approximately every five years. Each standards are subject to amendment or withdrawal, if required, thus the latest edition of IPS shall be applicable

The users of IPS are therefore requested to send their views and comments, including any addendum prepared for particular cases to the following address. These comments and recommendations will be reviewed by the relevant technical committee and in case of approval will be incorporated in the next revision of the standard.

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پیش گنگار

استانداردهای نفت ایران (IPS) منعکس کننده دیدگاه‌های وزارت نفت ایران است و برای ایجاد استفاده در تأسیسات تولید نفت و گاز، بالا بایشگاه‌های نفت، واحدهای شیمیایی و پتروشیمی، تأسیسات انتقال و فراورش گاز و سایر تأسیسات، مشابه به هدف است.

استانداردهای نفت، براساس استانداردهای قابل قبول بین‌المللی بهره و شرایط گرندیزی به اساس منابع مرجع در هر مورد می‌باشد. همچنین براساس تجربیات صممت نفت کشور و قابلیت تأمین کالا از بازار داخلی و نیز بررسی نیاز، مواردی بطور دکتری و یا اصلاحی در این استانداردها انجام شده است. مواردی از گرندیزی فنی که در متن استانداردها اورده شده است، در داده برگ یا به صورت شماره گزاری شده برای استفاده مناسب کاربران ارائه شده است.

استانداردهای نفت، به‌شکلی کامل اعمال پذیری تدوین شده است که کاربران به‌طور اندازه‌گیری یا ارتقاء خود را با آنها منطبق نمایند. با این حال ممکن است تمام نیازمندی‌های پژوهش را را پوشش ندهند. در این کشور موارد باید اصلاح شود و نیاز‌های خاص آنها را احتمالاً متغیب و پیش‌بینی نمایند. این حالیه ممکن است استانداردهای مربوط به مشخصات فنی آن پزوهش و یا کار خاص را تشکیل خواهد داد.

استانداردهای نفت، قرار گرفته روز و هر مورد بررسی قرار گرفته و روز و هر مورد بررسی ممکن است استانداردهای حفظ و یا اصلاحی به آن اضافه شود و بنابراین همگامه‌سازی و برای آنها مکانی عمل می‌باشد.

استانداردهای نفت، بر اساس استانداردها درخواست می‌شود نطفه نظاره بر پیشنهادات اصلاحی و یا که پیشنهاد کارهای نموده‌اند، به شکلی ارائه می‌شود. نظارت و پیشنهادات دریافتی در کمیته‌های فنی مربوطه بررسی و در صورت تصویب در تجدید نظر به‌دست استانداردهای منعکس خواهد شد.

ایران، تهران، خیابان کریمخان زند، خرمشند شمالی، کوچه چهاردهم، شماره 19
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General Definitions:
Throughout this Standard the following definitions shall apply.

Company:
Refers to one of the related and/or affiliated companies of the Iranian Ministry of Petroleum such as National Iranian Oil Company, National Iranian Gas Company, and National Petrochemical Company etc.

Purchaser:
Means the “Company” Where this standard is part of direct purchase order by the “Company”, and the “Contractor” where this Standard is a part of documents.

Vendor And Supplier:
Refers to firm or person who will supply and/or fabricate the equipment or material.

Contractor:
Refers to the persons, firm or company whose tender has been accepted by the company.

Executor:
Executor is the party which carries out all or part of construction and/or commissioning for the project.

Inspector:
The Inspector referred to in this Standard is a person/persons or a body appointed in writing by the company for the inspection of fabrication and installation work.

Shall:
Is used where a provision is mandatory.

Should:
Is used where a provision is advisory only.

Will:
Is normally used in connection with the action by the “Company” rather than by a contractor, supplier or vendor.

May:
Is used where a provision is completely discretionary.
ENGINEERING STANDARD
FOR
PROCESS DESIGN OF COMPRESSORS
FIRST REVISION
FEBRUARY 2010

استاندارد مهندسی
برای
طراحی فرآیندی کمپرسورها

ویرایش اول
بهمن 1388
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0. INTRODUCTION

"Process Design of Pressure Reducing/Increasing Machineries and or Equipment" are broad and contain various subjects of paramount importance. Therefore a group of process engineering standards are prepared to cover the subject.

This group includes the following standards:

<table>
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<th>Standard Code</th>
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<tr>
<td>IPS-E-PR-330</td>
<td>&quot;Engineering Standard for Process Design of Production &amp; Distribution Compressed Air Systems&quot;</td>
</tr>
<tr>
<td>IPS-E-PR-745</td>
<td>&quot;Engineering Standard for Process Design of Vacuum Equipment (Vacuum Pumps and Steam Jet Ejectors)&quot;</td>
</tr>
<tr>
<td>IPS-E-PR-750</td>
<td>&quot;Engineering Standard for Process Design of Compressors&quot;</td>
</tr>
<tr>
<td>IPS-E-PR-755</td>
<td>&quot;Engineering Standard for Process Design of Fans and Blowers&quot;</td>
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This Engineering Standard Specification covers:

"PROCESS DESIGN OF COMPRESSORS"
1. SCOPE

This Engineering Standard Specification covers the minimum requirements, basic reference data and necessary formulas for process calculations and proper selection of compressors to be used in the OGP industries.

Compressors are dealt within four groups; axial, centrifugal, reciprocating and rotary, and each covered in separate section.

Note 1:

This standard specification is reviewed and updated by the relevant technical committee on Feb 2005, as amendment No. 1 by circular No. 257.

Note 2:

This bilingual standard is a revised version of the standard specification by the relevant technical committee on Feb 2010, which is issued as revision (1). Revision (0) of the said standard specification is withdrawn.

Note 3:

In case of conflict between Farsi and English languages, English language shall govern.

2. REFERENCES

Throughout this Standard the following dated and undated standards/codes are referred to. These referenced documents shall, to the extent specified herein, form a part of this standard. For dated references, the edition cited applies. The applicability of changes in dated references that occur after the cited date shall be mutually agreed upon by the Company and the Vendor. For undated references, the latest edition of the referenced documents (including any supplements and amendments) applies.

API (AMERICAN PETROLEUM INSTITUTE)

API Std. 614  "Lubrication, Shaft-Sealing, and Control-Oil Systems and Auxiliaries" 5th. Ed

API Std. 617, "Axial & Centrifugal Compressors & Expander

API (موسسه نفت آمریکا)
Compressors for Petroleum, Chemical and Gas Industry Services, 7th Ed., July 2002

API Std. 618, "Reciprocating Compressors for Petroleum, Chemical and Gas Industry Services", 4th Ed. 1995

API Std. 619, "Rotary-Type Positive-Displacement Compressors for Petroleum, Petrochemical, and Natural Gas Industries", 3rd Ed. 1997


IPS (IRANIAN PETROLEUM STANDARDS)

IPS-E-GN-100 "Engineering Standard for Units" IPS-E-GN-100


IPS-M-PM-170 "Material and Equipment Standard for Centrifugal Compressors for Process Services" IPS-M-PM-170

IPS-M-PM-190 "Material and Equipment Standard for Axial Flow Centrifugal Compressors" IPS-M-PM-190

IPS-M-PM-200 "Material and Equipment Standard for Reciprocating Compressors for Process Services" IPS-M-PM-200

IPS-M-PM-220 "Material and Equipment Standard for Positive Displacement Compressors, Rotary" IPS-M-PM-220
3. DEFINITIONS AND TERMINOLOGY

3.1 Terms used in this Standard are in accordance with the relevant sections of definition of terms specified in API Standard 617, API Standard 618 and API Standard 619, unless otherwise stated in this Section.

3.2 Inlet Cubic Meters per Hour (Im³/h)

Refers to flow rate determined at the conditions of pressure, temperature, compressibility and gas composition, including moisture, at the compressor inlet flange (substitution to API Std. 617, 1.5.14).

3.3 Actual Cubic Meters per Hour (Am³/h)

Refers to the flow rate at flowing conditions of temperature and pressure at any given location. Because this term describes flow at a number of locations, it should not be used inter-changeably with inlet m³/h.

3.4 Standard Cubic Meter per Hour (Sm³/h)

Refers to the flow rate at any location corrected to a pressure of 101.325 kPa and at a temperature of 15°C with a compressibility factor of 1.0 and in a dry condition.

3.5 Normal Cubic Meters per Hour (Nm³/h)

Refers to a flow rate at any location corrected to the normal atmospheric pressure and a temperature of 0°C with a compressibility factor of 1.0 and in dry conditions.

3.6 Specific Volume

Is the volume per unit mass or volume per mole of material.
4. SYMBOLS AND ABBREVIATIONS

**C_p** = Specific heat at constant pressure,

**C_v** = Specific heat at constant volume,

**D** = Cylinder inside diameter,

**d** = Piston rod diameter,

**Ghp** = Gas horsepower, actual compression horsepower excluding mechanical losses,

**H** = Head,

**h** = Enthalpy,

**k** = Isentropic exponent, \( \frac{C_p}{C_v} \)

**MC_p** = Molar specific heat at constant pressure,

**MC_v** = Molar specific heat at constant volume,

**MW** = Molecular weight,

**N** = Speed, rpm

**Nm** = Molar flow, moles/min

**n** = Polytropic exponent or number of moles

**P** = Pressure,

**PD** = Piston displacement,

**Q** = Inlet capacity

**Qg** = Standard gas flow rate,

**r** = Compression ratio, \( \frac{P_2}{P_1} \)

**s** = Entropy,

**Stroke** = Length of piston movement,

**T** = Absolute temperature,

**t** = Temperature

**VE** = Volumetric efficiency,

**w** = Weight flow,

**Z** = Compressibility factor,

**Zavg** = Average compressibility factor,

**η** = Efficiency, expressed as a decimal,

**Subscripts**

**avg** = Average

**d** = Discharge

**g** = Gas

**is** = Isentropic process

**L** = Standard conditions used for calculation or contract

- **C_p**، گرمایی ویژه در فشار ثابت
- **C_v**، گرمایی ویژه در حجم ثابت
- **D**، قطر داخلی سیلندر
- **d**، قطر دسته پیستون
- **Ghp**، تویان اسب بخار گاز، تویان اسب بخار واقعی تراکم
- **H**، ارتفاع
- **h**، آنتالپی
- **k**، نمای ایزوتروپیک
- **MC_p**، گرمایی ویژه مولی در فشار ثابت
- **MC_v**، گرمایی ویژه مولی در حجم ثابت
- **MW**، جرم مولکولی
- **N**، سرعت، دور بر دقیقه
- **Nm**، جریان مولی، مول بر دقیقه
- **n**، نمای پلی تروپیک یا تعداد مولها
- **P**، فشار
- **PD**، جابجایی بیستون
- **Q**، ظرفیت ورودی
- **Qg**، شدت جریان استاندارد گاز
- **r**، نسبت تراکم \( \frac{P_2}{P_1} \)
- **s**، آنتروپی
- **Stroke**، طول حرکت بیستون
- **T**، دما مطلق
- **t**، دما
- **VE**، راندمان حجمی
- **w**، جریان جرمی
- **Z**، ضریب تراکم پذیری
- **Zavg**، ضریب تراکم پذیری میانگین
- **η**، راندمان، بستره اعشاری بیان می شود
- **avg**، میانگین
- **d**، خروجی
- **g**، گاز
- **is**، فرآیند ایزوتروپیک
- **L**، شرایط استاندارد مورد استفاده برای محاسبه یا قرارداد
p = Polytropic process
S = Standard conditions, usually 14.7 psia, 60°F
s = Suction
t = Total or overall
1 = Inlet conditions
2 = Outlet conditions
5. UNITS
This Standard is based on International System of Units (SI) as per IPS-E-GN-100, except where otherwise specified.

6. GENERAL
- Compressors are generally divided into three major types, dynamic, positive displacement and thermal as shown in Fig. A.1 of Appendix A.
- For typical figures of three type of compressors see Appendix C
- The type of compressor to be used shall be the most suitable for the duty involved. See the compressor coverage chart in Fig. A.2 of Appendix A.
- Adequate knock out facilities including demister pads where necessary shall be provided to prevent damage by liquid carry over into the compressor.
- Compressors handling SO₂, HCl or other gases which are corrosive in the presence of water, shall not employ water as a cooling medium unless the water circuit is positively isolated from the gas side, e.g., by separate water jackets. It is not sufficient to rely on gaskets or seals for isolation.

Similar restrictions shall apply to the use of glycol as a coolant for machines handling corrosive gases plus hydrogen as the hydrogen can react with glycol to form water. The use of oil as a cooling medium will be acceptable as an alternative in special cases.
- Rotodynamic compressors are to be provided with anti-surge equipment. The response time for the control equipment shall be such as to prevent surge during any anticipated process condition, due consideration being given to the speed at which process changes or upsets can move the compressor operation towards surge.

For the more complicated installations with multiple stages and sidestreams, or multiple units (in series or parallel) or variable speed units, an analysis of the stability of the anti-surge control system is also necessary.

6.1 Type Selection Criteria

The choice of the type of compressor, whether axial, centrifugal, reciprocating or rotary, depends primarily on the required flow to be compressed, the density of the gas in conjunction with the total head (for a given gas, this is the compression ratio) and the duty which has to be performed. Table A.1 of Appendix A outlines the compression limits for the four types of compression equipment.

6.1.1 Axial compressors

Axial compressors can handle large volume flow and are more efficient than centrifugal compressors. However, centrifugals are less vulnerable and hence more reliable, have wider operating ranges and are less susceptible to fouling.

Axial compressors should be considered only for air, sweet natural gas or non-corrosive gases. Axial compressors shall be in accordance with Iranian Petroleum Standards IPS-M-PM-190, for "Axial Flow Centrifugal Compressors".

6.1.2 Centrifugal compressors

Providing a centrifugal compressor can handle the required flow with a reasonable efficiency, then this type is the preferred choice because it has the potential to operate continuously for long periods.

- A type of centrifugal compressor is the preferred choice because it can handle the required flow with a reasonable efficiency, and the type is expected to operate continuously for a long

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- A type of centrifugal compressor is the preferred choice because it can handle the required flow with a reasonable efficiency, and the type is expected to operate continuously for a long
Compressors for Process Services shall be in accordance with API Std. 619 as amended by Iranian Petroleum Standards IPS-M-PM-170, for "Centrifugal Compressors for Process Services".

6.1.3 Reciprocating compressors

Where the required flow is too small for a centrifugal compressor, or where the required head is so high that an undesirably large number of stages would be necessary, then generally the choice should be a reciprocating compressor.

As a reciprocating compressor cannot fulfill the minimum requirement of continuous uninterrupted operation for a twoyear period, due to fairly high maintenance requirements, a full-capacity spare shall be provided as general rule for reciprocating compressors in critical services. Alternatively, three half-capacity machines may be specified, two running in parallel with the third unit as a spare. Reciprocating compressors shall be in accordance with API Std. 618 as amended by IPS-G-PM-200, for "Reciprocating Compressors for Process Services".

6.1.4 Rotary compressors

Rotary compressor shall be considered only where there is proven experience of acceptable performance of this type of compressor in the duty concerned and only where there are advantages over a reciprocating compressor.

The application of oil flooded screw compressors for instrument air and of dry running rotary screw compressors, sliding vane compressors and rotary lube compressors for process duties, requires the explicit approval of the Company.

Rotary-type positive displacement compressors shall be in accordance with API Std. 619 as amended by IPS-G-PM-200, for "Reciprocating Compressors for Process Services".

6-1-3 Centrifugal Compressors

Centrifugal compressors shall be designed in accordance with API Std. 617 as amended by Iranian Petroleum Standards IPS-M-PM-170, for "Centrifugal Compressors for Process Services".

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Rotary-type positive displacement compressors shall be in accordance with API Std. 619 as amended by IPS-G-PM-200, for "Reciprocating Compressors for Process Services".

API 617 and API 619 are both standards for centrifugal compressors. API 617 covers the design and construction of centrifugal compressors, while API 619 covers the design and construction of positive displacement compressors. Both standards require the approval of the company for their application in specific duties.

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The application of oil flooded screw compressors for instrument air and of dry running rotary screw compressors, sliding vane compressors and rotary lube compressors for process duties, requires the explicit approval of the Company.

Rotary-type positive displacement compressors shall be in accordance with API Std. 619 as amended by IPS-G-PM-200, for "Reciprocating Compressors for Process Services".
amended by IPS-M-PM-220, for "Positive Displacement Compressors, Rotary”.

6.2 Atmospheric Pressure
The absolute pressure of the atmosphere at the site should be considered as the "absolute pressure" in the compressor calculations. The value of the absolute pressure is taken as 101.325 kPa at sea level and declines with increasing altitude as shown in Table A.2 of Appendix A.

6.3 Specification Sheets
Process information required to complete specification sheets for compressors are presented in Appendix B.

7. CENTRIFUGAL COMPRESSORS
7.1 General
7.1.1 The centrifugal (radial flow) compressor is well established for the compression of gases and vapors. It has proven its economy and uniqueness in many applications, particularly where large volumes are handled at medium pressures.

7.1.2 Centrifugal compressors shall conform to API Std. No. 617 for all services handling air or gas, except machines developing less than 35 kPa (0.35 bar) from atmospheric pressure, which may be classified as fans or blowers.

7.1.3 Performance
7.1.3.1 Compressors shall be guaranteed for head, capacity, and satisfactory performance at all specified operating points and further shall be guaranteed for power at the rated point.

7.1.3.2 a) The volume capacity at the surge point shall not exceed the specified percentage of normal capacity at normal speed, and normal (unthrottled) suction conditions. The rise in pressure ratio from normal capacity to the surge point at normal speed shall not be less than that specified.

b) The head developed at 115% of normal capacity at normal speed shall be not less than approximately 85% of the head developed at the normal operating point.

7-6 فشار جوی
نوعی می‌شود فشار مطلق محیط در محل واحد در محاسبات کمپرسور به عنوان "فشار مطلق" در نظر گرفته شود. فشار مطلق در سطح دریا 325/101 کیلو پاسکال و با افزایش ارتفاع مطبق شکل اف-20 یکی باشد یا بوده و با افزایش ارتفاع مطبق شکل اف-20 پیوست به کاهش می‌یابد.

7-4 گره‌های مشخصات
اطلاعات ابتدایی لازم برای تکمیل گره‌های مشخصات کمپرسور در پیوست (ب) نشان داده شده‌اند.

7-7 کمپرسورهای گریز از مرکز
1-7 عمومی
کمپرسور گریز از مرکز (جایگاه شوائی) برای مکان کردن گازها و بخارها کاملاً ساخته شده است. منحصر به فرد و اقتصادی بودن آن در اکثر کاربردها ثابت شده است مخصوصاً که همراهی با فشار متوسط مترکب می‌شود.

2-7 کارآیی
2-7 کمپرسورهای گریز از مرکز برای تمام کاربرد-7 API های حاوی یا گاز باید مطابق با استاندارد 617 باند به جز ماهین آتات که فشار کمتر از 35 کیلو پاسکال (500 بار) از فشار جوی تولید می‌کنند که به عنوان دندو بهادر طبقه بندی می‌شوند.

3-7 کارآیی
3-7 کمپرسورهای باید برای ارتفاع، ظرفیت و کارآیی رضایت بخش در تمام نقاط عملیاتی مشخص تضمین شوند و همچنین باید برای توان مصرفی در نقطه تعیین ظرفیت تضمین شوند.

4-7
7.1.3.3 The head-capacity characteristic curve shall rise continuously from the rated point to the predicted surge. The compressor, without the use of a bypass, shall be suitable for continuous operation at any capacity at least 10 percent greater than the predicted approximate surge capacity shown in the proposal.

7.1.3.4 For variable speed compressors, the head and capacity shall be guaranteed with the understanding that the power may vary ±4%.

7.1.3.5 For constant-speed compressors, the specified capacity shall be guaranteed with the understanding that the head shall be within ±5% and -0% of that specified; the power shall not exceed stated power by more than 4%. These tolerances are not additive.

7.1.4 The compressor manufacturer shall be responsible for checking the "k" (ratio of specific heats) and "Z" (compressibility factor) values specified against the gas analysis specified.

7.1.5 Compressor mach numbers shall not exceed 0.90 when measured at any point.

7.2 Design Criteria

7.2.1 This Section of Standard covers information necessary to select centrifugal compressors and to determine whether the selected machine should be considered for a specific job.

7.2.2 An approximate idea of the flow range that a centrifugal compressor will handle is shown in Table 1. A multistage centrifugal compressor is normally considered for inlet volumes between 850 and 340,000 Im³/h. A single stage compressor would normally have applications between 170 and 255,000 Im³/h. A multi-stage compressor can be thought of as series of single stage compressors contained in a single casing.

7.2.3 The head-capacity characteristic curve shall rise continuously from the rated point to the predicted surge. The compressor, without the use of a bypass, shall be suitable for continuous operation at any capacity at least 10 percent greater than the predicted approximate surge capacity shown in the proposal.

7.2.4 For variable speed compressors, the head and capacity shall be guaranteed with the understanding that the power may vary ±4%.

7.2.5 For constant-speed compressors, the specified capacity shall be guaranteed with the understanding that the head shall be within ±5% and -0% of that specified; the power shall not exceed stated power by more than 4%. These tolerances are not additive.

7.2.6 The compressor manufacturer shall be responsible for checking the "k" (ratio of specific heats) and "Z" (compressibility factor) values specified against the gas analysis specified.

7.2.7 Compressor mach numbers shall not exceed 0.90 when measured at any point.
### TABLE 1 – CENTRIFUGAL COMPRESSOR FLOW RANGE

<table>
<thead>
<tr>
<th>SPEED TO DEVELOP 3048 m HEAD/WHEEL</th>
<th>AVERAGE ISENTROPIC EFFICIENCY</th>
<th>AVERAGE POLYTROPIC EFFICIENCY</th>
<th>NOMINAL FLOW RANGE (INLET m³/h)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>میانگین بارده ایزوتروپیک</td>
<td>میانگین بارده پلی تروپیک</td>
<td>محدوده جریان اساسی</td>
</tr>
<tr>
<td>170 - 850</td>
<td>0.63</td>
<td>0.60</td>
<td>20,500</td>
</tr>
<tr>
<td>850 - 12,743</td>
<td>0.74</td>
<td>0.70</td>
<td>10,500</td>
</tr>
<tr>
<td>12,743 - 34,000</td>
<td>0.77</td>
<td>0.73</td>
<td>8,200</td>
</tr>
<tr>
<td>34,000 - 56,000</td>
<td>0.77</td>
<td>0.73</td>
<td>6,500</td>
</tr>
<tr>
<td>56,000 - 93,400</td>
<td>0.77</td>
<td>0.73</td>
<td>4,900</td>
</tr>
<tr>
<td>93,400 - 135,900</td>
<td>0.77</td>
<td>0.73</td>
<td>4,300</td>
</tr>
<tr>
<td>135,900 - 195,400</td>
<td>0.77</td>
<td>0.73</td>
<td>3,600</td>
</tr>
<tr>
<td>195,400 - 246,400</td>
<td>0.77</td>
<td>0.73</td>
<td>2,800</td>
</tr>
<tr>
<td>246,400 - 340,000</td>
<td>0.77</td>
<td>0.73</td>
<td>2,500</td>
</tr>
</tbody>
</table>

### 7.2.3 Effect of speed

7.2.3.1 With variable speed, the centrifugal compressor can deliver constant capacity at variable pressure, variable capacity at constant pressure, or a combination of variable capacity and variable pressure.

7.2.3.2 Basically, the performance of the centrifugal compressor, at speeds other than design, follows the affinity (or fan) laws.

7.2.3.3 By varying speed, the centrifugal compressor will meet any load and pressure condition demanded by the process system within the operating limits of the compressor and the driver.

7.2.3.4 If speed is constant then Characteristic operating curve will be also constant. The following factors will increase suction pressure resulting in change of discharge pressure:

- a) Molecular weight of gas increases
- b) Suction pressure increases
- c) Inlet temperature decreases
- d) Compressibility factor decreases
- e) Ratio of specific heats, k decreases
7.2.4 Performance calculation

7.2.4.1 Determination of properties pertaining to compression

Compressibility factor (Z factor), ratio of specific heats \( \frac{C_p}{C_v} \) or \( k \) value) and molecular mass are three major physical properties for compressor which must be clarified.

7.2.4.2 Determination of suction conditions

The following conditions at the suction flange should be determined:

a) Temperature

b) Pressure

In case of air taken from atmosphere, corrections should be made for elevation. Air humidity should also be considered.

c) Flow rate

All centrifugal compressors are based on flows that are converted to inlet or actual conditions (Im³/h or inlet cubic meters per hour). This is done because centrifugal compressor is sensitive to inlet volume, compression ratio (i.e., head) and specific speed (see 6.2.7).

d) Fluctuation in conditions

Since fluctuations in inlet conditions will have large effects on the centrifugal compressor performance, owing to the compressibility of the fluid, all conceivable condition fluctuations must be taken into consideration in determination of design conditions.

7.2.4.3 Determination of discharge conditions

7.2.4.3.1 Calculation method

Discharge conditions of a centrifugal compressor can be calculated by the following procedure.

a) Calculate the polytropic exponent "n":

1) Using the equation:

\[
\frac{P_2}{P_1} = \left[ \frac{V_2}{V_1} \right]^n
\]

where:

- \( P_1 \) is the suction pressure,
- \( P_2 \) is the discharge pressure,
- \( V_1 \) is the suction volume,
- \( V_2 \) is the discharge volume,
- \( n \) is the polytropic exponent.

The polytropic exponent is determined using the polytropic equation:
if \( \eta_p \) (polytropic efficiency) is known from the manufacturer data.

\( \eta_p \) can also be estimated from Table 1 (k is the ratio of specific heats).

2) if \( \eta_i \) (isentropic or adiabatic efficiency) is known, then \( \eta_p \) can be found from Figs. 1 or 2 and the Equation 1 can be used to calculate "n".

3) Fig. 3 is useful for rough estimation of "n".

4) "n" can also be calculated iteratively from equation:

\[
(Eq. 2) \quad n = \frac{\log_{10} \left( \frac{P_2}{P_1} \right)}{\log_{10} \left( \frac{V_1}{V_2} \right)}
\]

Where \( V_1 \) and \( V_2 \) are specific volumes (actual) and \( P_1 \) and \( P_2 \) are absolute pressures at inlet and outlet conditions respectively.
Fig. 2-RELATIONSHIP BETWEEN ADIABATIC AND POLYTROPIC EFFICIENCIES

Fig. 3- RATIO OF SPECIFIC HEATS, $K = \frac{C_P}{C_v}$

$\frac{(n-1)}{n}$ VERSUS RATIO OF SPECIFIC HEATS
This equation applies with good accuracy for single wheels and the overall multistage compressor.

b) Calculate discharge temperature $T_2$, (kelvin) from equation:

$$T_2 = T_1 \left( \frac{P_2}{P_1} \right)^{(n-1)/n}$$

$$(T_1 \text{ and } T_2 \text{ are absolute temperatures})$$

These values are for polytropic compression in an uncooled compressor with no diaphragm cooling, no liquid injection and no external coolers. In the cases of internal cooling, the adiabatic exponent "k" approximates the actual condition.

In such cases:

$$\Delta T = T_1 \times \frac{\left( \frac{P_2}{P_1} \right)^{(k-1)/k} - 1}{\eta_{is}}$$

$$(\text{Eq. 4})$$

Where:

$\Delta T$ is the temperature increase, in °C.

Note:

The operating temperature should not exceed 190°C (375°F) at any point in the operating range, otherwise, difficulties will be encountered in the mechanical design, higher temperatures up to 232°C (450°F) are subject to Company’s approval.

c) Calculate adiabatic (isentropic) head $H_{is}$ (meters):

$$H_{is} = \frac{Z_{ave}RT_1}{gM(k-1)/k} \left[ \left( \frac{P_2}{P_1} \right)^{(k-1)/k} - 1 \right]$$

$$(\text{Eq. 5})$$

Where:

$Z_{ave}$ is the average molar mass of the gas.
$R$ is gas constant, in \( \text{J/kmol.K} \);

\[ R = \frac{8314.3}{\text{J/kmol.K}} \]

$Z_{ave}$ is average inlet and outlet compressibility factors;

$T_1$ is inlet absolute temperature, \( \text{kelvin (K)} \);

$M$ is molecular mass, \( \text{kg/kmol} \);

$g$ is acceleration of gravity, \( \text{9.80665 m/s}^2 \).

d) Calculate polytropic head \( H_p \):

\[
(Eq. 6) \quad H_p = \frac{Z_{ave}RT_1}{gM(n-1)/n} \left[ \left( \frac{P_1}{P_2} \right)^{(n-1)/n} - 1 \right]
\]

Note:

Polytropic and isentropic heads are related by:

\[
(Eq. 7) \quad \frac{H_s}{H_e} = \eta_e
\]

e) Calculate gas horse power in kilowatt (hp):

\[
(Eq. 8) \quad G_{hp} = \frac{W \cdot H_p}{6119.099 \eta_p}
\]

or:

\[
(Eq. 9) \quad G_{hp} = \frac{W \cdot H_{is}}{6119.099 \eta_{is}}
\]

Where:

$W$ is mass flow rate, \( \text{kg/min} \);

$H_p$ is polytropic head, \( \text{m} \).

Where:

$W$ is mass flow rate, \( \text{kg/min} \);

$H_p$ is polytropic head, \( \text{m} \).

f) Estimate head per stage:

1) Use the following equation based on molecular mass for calculation of maximum head per stage:

\[
(1) \quad \text{(معادله 7)}
\]
(Eq. 10) \[ H_{\text{max/stage}} = 4572 - 457.2 \ (M)^{0.35} \]  

2) Find uncorrected number of stages SN:

\[ SN = \frac{H_P}{H_{\text{max/stage}}} \]

3) Correct number of stages by choosing the next upper integer, S.

4) Find \( H' \), the head per impeller:

\[ H' = \frac{H_P}{S} \]

g) Estimate speed and wheel diameter:

1) Use Fig. 4 to find the size number;

2) Find the approximate wheel diameter, \( D \), from Table 2;

3) Choose pressure coefficient, \( \mu \) and find peripheral velocity, \( u \), from Fig. 5;

4) Calculate \( N \) from the equation.

\[ N = \frac{59809.42}{D} \sqrt{\frac{H'}{\mu}} = 19108.33 \times \frac{u}{D} \]

Where:

- \( N = r/\text{min} \ (\text{rpm}) \)
- \( H_{\text{total}} \) Is total compressor head (meters of fluid);
- \( S \) Is No. of stages S, can also be found by dividing \( H_p \) by \( H' \);
- \( D \) Is impeller diameter, (mm);
- \( H' \) Is head per stage, (meters of fluid);
- \( \mu \) (mu) Is pressure coefficient, values range from 0.5 to 0.6 (average 0.55, use Fig. 5);
- \( H_{\text{total}} \) Is total number of stages, \( S \), can also be found by dividing \( H_p \) by \( H' \);
- \( D \) Is impeller diameter, (mm);
- \( H' \) Is head per stage, (meters of fluid);
- \( \mu \) (mu) Is pressure coefficient, values range from 0.5 to 0.6 (average 0.55, use Fig. 5).
TABLE 2 - APPROXIMATE WHEEL DIAMETER vs SIZE NUMBER

<table>
<thead>
<tr>
<th>Size No.</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
<th>5</th>
<th>6</th>
<th>7</th>
</tr>
</thead>
<tbody>
<tr>
<td>Wheel diameter (mm)</td>
<td>375</td>
<td>450</td>
<td>600</td>
<td>800</td>
<td>1060</td>
<td>1350</td>
<td>1650</td>
</tr>
</tbody>
</table>

\( u \) is peripheral velocity, (m/s).

Fig. 4-CENTRIFUGAL COMPRESSOR SIZE vs CAPACITY

\( u \) is peripheral velocity, (m/s).

Fig. 5-PERIPHERAL VELOCITY (m/s)
h) Brake horse power (in kilowatt):

Is determined by addition of power losses due to friction in bearings, seals and speed increasing gears to the gas horse power in kilowatts (Ghp). The equation:

\[(\text{Eq. 14}) \quad \text{Mechanical losses} = 0.663 \times (\text{Ghp})^{0.4}\]

is a good estimation for these losses.

\[(\text{Eq. 15}) \quad \text{Bhp} = \text{hp} + \text{mechanical losses}\]

7.2.4.3.2 P-H diagram method

The use of an enthalpy diagram, when available, is the most accurate and an easy method for determining power. Fig. 7 represents a section of a typical P-H diagram. The following procedure should be followed: starting from point 1 (inlet conditions), follow the line of constant entropy to the required discharge pressure \(P_2\), locating the isentropic discharge state point (2\(_{is}\)). Now the differential isentropic enthalpy can be calculated from:

\[(\text{Eq. 16}) \quad \Delta h_{is} = h_{2is} - h_1\]

\(H_{is}\) (isentropic head) can then be found:

\[(\text{Eq. 17}) \quad H_{is} = \Delta h_{is} \times 101.978 \text{ if } h \text{ is in kJ/kg}\]

Where \(H_{is}\) is in meters of fluid.

From Eqs. 17, 7, 8 and 9:

\[(\text{Eq. 18}) \quad \Delta h = \frac{\Delta h_{is}}{\eta_{is}} = \frac{\Delta \text{hp}}{\eta_p}\]

Where \(\Delta h\) is the actual differential enthalpy.
Fig. 6-REQUIRED NUMBER OF WHEELS

شکل 6- تعداد چرخهای مورد نیاز

Fig. 7-P-H DIAGRAM CONSTRUCTION

شکل 7- ساختار نمودار آنتالپی، فشار
To find the discharge enthalpy:

\[
\text{Eq. 19} \quad h_2 = \frac{\Delta h_{\text{is}}}{\eta_{\text{is}}} + h_1
\]

The actual discharge temperature can now be obtained from the P-H diagram.

**7.2.5 Sonic or acoustic velocity, in any gas may be calculated from:**

\[
\text{Eq. 20} \quad V_S = \left( \frac{k RT}{M} \right)^{1/2}
\]

Where:

- \( V_S \) is in meters per second;
- \( k \) is the ratio of specific heats;
- \( T \) is suction absolute temperature of gas, in (Kelvin);
- \( M \) is gas velocity at any point.

**General design practice avoids using gas velocities near or above the sonic velocity.**

**7.2.6 The ratio of gas velocity at any point to the sonic velocity of the gas is known as "mach number, \( M' \).”**

\[
\text{Eq. 21} \quad M' = u'/V_S
\]

Where \( u' \) is gas velocity at any point.

**7.2.7 Specific speed**

At a given point, the "specific speed", correlates the important performance factors of adiabatic head, capacity and r/min for geometrically similar wheels. The specific speed of all geometrically similar wheels is the same and does not change when the speed of the wheel is changed.

\[
\text{Eq. 22} \quad N_S = \frac{0.315 \times r}{(Ha)^{0.75}} \sqrt{V_1}
\]

Where:

- \( N_S \) is specific speed, in r/min (rpm);
- \( V_1 \) is flow rate, in m³/h at suction condition;
7.2.8 Flow limits

Two conditions associated with centrifugal compressors are surge (pumping) and stone-wall (choked flow).

7.2.8.1 Surge point

At some point on the compressors operating curve there exists a condition of minimum flow/maximum head where the developed head is insufficient to overcome the system resistance. This is the "surge point". When the compressor reaches this point, the gas in the discharge piping back-flows into the compressors. Without discharge flow, discharge pressure drops until it is within the compressor’s capability, only to repeat the cycle. The repeated pressure oscillations at the surge point should be avoided since it can be detrimental to the compressor. Surging can cause the compressor to overheat to the point the maximum allowable temperature of the unit is exceeded. Also, surging can cause damage to the thrust bearing due to the rotor shifting back and forth from the active to the inactive side.

7.2.8.2 Stone-Wall or choked flow

Stone-wall-or choked flow occurs when sonic velocity is reached at any point in the compressor. When this point is reached for a given gas, the flow through the compressor can not be increased further without internal modifications.

7.2.9 Interstage cooling

Multistage compressors rely on intercooling whenever the inlet temperature of the gas and the required compression ratio are such that the discharge temperature of the gas exceeds about...
150°C. Performance calculations indicate that the head and power are directly proportional to the absolute gas temperature at each impeller.

When interstage coolers are furnished, the vendor shall provide the following:

a. Drawing showing cooling system details
b. Data for purchasers heat and material balances.

c. Details of provisions for separating and withdrawing condensate.

d. Vendor’s recommendations regarding provision for support and piping expansion.

7.2.10.2 Instrumentation

Compressor controls can vary from the very basic manual recycle control to the elaborate ratio controllers. The driver characteristics, process response and compressor operating range must be determined before the right controls can be selected.

7.2.10.1 Control systems

Control system shall be designed for start-up operation, for all specified operating conditions, and for surge prevention. The method of control, the source of the control signal, its sensitivity and range, and the equipment to be furnished by vendor should be specified. Compressor control may be accomplished by suction throttling, variable inlet guide vanes, variable stator vanes, speed variation, a cooled bypass from discharge to suction, discharge blowoff or discharge throttling.

In cases where constant speed drivers are used, the inlet gas density can be reduced by throttling or by adjusting the compressor guide vanes. Different features of the centrifugal compressor control systems are described in GPSA.

7.2.10.2 Instrumentation

All instruments shall be located and arranged to permit easy visibility by the operators, as well as accessibility for tests, adjustments, and maintenance.

Unless otherwise specified, all instruments other than shut-down sensing devices shall be
installed with sufficient valving to permit their replacement while the system is in operation. When shut-off valves are specified for shut-down sensing devices, the vendor shall provide a means of locking the valves in the open position.

Except for instrument air service, bleed valves are required between instruments and their isolation valves. Combination isolation/bleed valves may be used.

Refer to API Std. 614 for details on instrumentation.

7.2.10.3 Anti-Surge control

It is essential that all centrifugal compressor control systems be designed to avoid possible operation in surge which usually occurs below 50% to 70% of the rated flow.

The surge limit line can be reached by reducing flow or decreasing suction pressures and/or increasing discharge to suction pressure ratio. An anti-surge system senses conditions approaching surge, and maintains the unit pressure ratio below the surge limit by recycling some flow to the compressor suction. A volume-controlled anti-surge system is shown in Fig. 8. As the flow decreases to less than the minimum volume set point, a signal will cause the surge control valve to open, to keep a minimum volume flowing through the compressor.

A pressure-limiting anti-surge control system is shown in Fig. 9. A process pressure increase over the pressure set point will cause the blow-off valve to open. The valve opens as required to keep the pressure limited to a minimum of gas or air flowing through the compressor.
8. AXIAL COMPRESSORS

8.1 General

8.1.1 Axial compressor is usually a single inlet, uncooled machine consisting essentially of blades mounted on a rotor turning between rows of stationary blades mounted on the horizontally split casing.
8.1.2 All requirements and recommendations specified in this Section are amendments or additions to those of Section 6 of this Standard.

8.1.3 Performance guarantee

a) Compressors shall be guaranteed for head; capacity and satisfactory performance at all specified operating points and further shall be guaranteed for power at the normal operating point.

b) For variable-speed compressors, the head and capacity shall be guaranteed with the understanding that the power may vary ±4%.

c) For constant-speed compressors, the specified capacity shall be guaranteed with the understanding that the head shall be specified for 100.0 and 105.0 percent; the power consumption shall not exceed stated power by more than 4%. These tolerances are not additive.

d) For compressors handling side loads or for two or more compressors driven by a single drive, the required performance guarantee for each compressor "section" shall be agreed upon by the Company and the Vendor.

8.2 Design Criteria

8.2.1 Performance

8.2.1.1 The minimum head rise to surge of an axial machine should be specified. The normal operating point shall be at least 10% removed in flow from surge point.

8.2.2 Gas velocities

General guideline for good design practice indicates an axial velocity for air of 91 to 137 meters per second. For other gases, the axial velocity range is in direct proportion to the speed of sound of the gas compared to air. The internal shape of the machine is usually arranged to give constant gas velocity as the gas travels through.
8.2.3 Volume

The size is determined by the inlet volume. The lower volume limit is approximately 8500 m³/h but the upper limit practically does not exist, units have been built to handle well above 1,700,000 m³/h.

9. RECIPROCATING COMPRESSORS

9.1 General

9.1.1 The reciprocating compressor is a positive displacement unit with the pressure on the fluid developed within a cylindrical chamber by the action of a moving piston. It may consist of one or more cylinders each with a piston or plunger that moves back and forth, displacing a positive volume with each stroke.

9.1.2 Reciprocating compressors shall conform to API 618 for all services except portable air compressors, and standard utility air compressors of 400 kW or less with not more than 900 kPa (9 bar) discharge pressure. This latter group will generally be purchased as packaged units.

9.1.3 Reciprocating compressors normally should be specified for constant-speed operation to avoid excitation of torsional and acoustic resonances. Rated speed (in revolutions per minute) is the highest speed required to meet any of the specified operating conditions. When variable-speed drivers are used, all equipment shall be designed to run safely to the trip speed setting.

9.1.4 When considering the use of a single frame for cylinders on different services particular attention shall be given to the means of independently controlling the different process streams. Care shall also be taken to ensure that the frame, transmission and driver can accept the wide variety of loadings that occur during all operating modes including start-up and shut-down.

9.1.5 Speed ranges

The vendor shall inform the purchaser of all critical speeds from zero to trip speed or

9.2 Compressors of 400 kW or less with not more than 900 kPa (9 bar) discharge pressure. This latter group will generally be purchased as packaged units.

9.3 API 618 for all services except portable air compressors.
9.1.6 Capacity control

9.1.6.1 Capacity control for constant-speed units normally will be obtained by suction valve unloading (depressors or lifters), clearance pockets, a combination of both pockets and unloaders, or bypass. Operation of controls shall be automatic. Unless stated otherwise, five-step unloading shall provide capacities of 100, 25, and 0 percent; three-step unloading shall provide capacities of 100, 50, and 0 percent; and two-step unloading shall provide capacities of 100 and 0 percent.

9.1.6.2 Capacity control on variable-speed units generally is by speed control.

9.1.6.3 Clearance pockets may be either the two-position type (pocket either open or closed) or the variable-capacity type. If not specified, the Vendor shall propose on the data sheet the type recommended for the Purchaser’s or Company’s operating conditions.

9.1.6.4 When unloading for startup is necessary, unloading arrangement shall be stated on the data sheet or shall be mutually agreed upon between the Purchaser’s and the Company’s and the Vendor.

9.2 Design Criteria

9.2.1 This Section covers information necessary for process engineers to determine the synchronous speed that occur during acceleration or deceleration.

<table>
<thead>
<tr>
<th>Low speed unit-up to 330 r/min</th>
<th>Medium speed unit 330 to 700 r/min</th>
<th>High speed unit-over 700 r/min</th>
</tr>
</thead>
<tbody>
<tr>
<td>300 r/min</td>
<td>500 r/min</td>
<td>700 r/min</td>
</tr>
<tr>
<td>250 r/min</td>
<td>750 r/min</td>
<td>900 r/min</td>
</tr>
<tr>
<td>200 r/min</td>
<td>1000 r/min</td>
<td></td>
</tr>
<tr>
<td>150 r/min</td>
<td>1500 r/min</td>
<td></td>
</tr>
<tr>
<td>100 r/min</td>
<td>2000 r/min</td>
<td></td>
</tr>
<tr>
<td>50 r/min</td>
<td>3000 r/min</td>
<td></td>
</tr>
<tr>
<td>25 r/min</td>
<td>5000 r/min</td>
<td></td>
</tr>
<tr>
<td>0 r/min</td>
<td>10000 r/min</td>
<td></td>
</tr>
</tbody>
</table>

Generally high speed units are preferred for units under 1865 kW. For larger units the choice is between low and medium speed.

The compressor vendor shall provide the necessary lateral and torsional vibrations that may hinder the operation of the complete unit within the specified operating speed range in any specified loading step.

9-1-9 حجم وظائف

9-1-9 كنترول قطاع

9-1-9 كنترول طريقة برای واحدی از واحدی در نتای عاملی با برداشتی برای شریفکش (کاهدها و یا بالابر) فاصله لی شیارها، ترکبی از برداشتی و فاصله لی شیارها برای گذر به دست می‌آید. عملیات کنترل با باید خودکار باشد. به غیر از موارد مشخص، برداشتی برای پنجم مدل‌های واحد و تریفاتهای 1000، 100 و 50 و 25 و 0 درصد و با ربارداری دو مرحله‌ای 100 و 0 درصد را تأیین نماید.

9-2-9 کنترول طرق در واحدی دو مرحله با واحدی دو مرحله

9-2-9 میزان از راه اندازی با ربارداری لازم برای آرامی با ربارداری با برای آرامی با برای

9-2-9 مقایسه بین خریدار و یا کارفرما و فروشنده باشد.
approximate power required to compress a certain volume of gas at some intake conditions to a given discharge pressure, and estimate the capacity of an existing reciprocating compressor under specified suction and discharge conditions.

9.2.2 Reciprocating compressors are furnished either single-stage or multi-stage. The number of stages is determined by the overall compression ratio.

9.2.3 On multistage machines, intercoolers may be provided between stages. Such cooling reduces the actual volume of gas going to the high pressure cylinders, reduces the power required for compression, and keeps the temperature within safe operating limits.

9.2.4 Reciprocating compressors should be supplied with clean gas as they cannot satisfactorily handle liquids and solid particles that may be entrained in the gas.

9.2.5 In evaluating the work of compression, the enthalpy change is the best way. If a P-H diagram is available, the work of compression should always be calculated by the enthalpy change of the gas in going from suction to discharge conditions.

9.2.6 The k value of a gas is associated with adiabatic compression or expansion. The change in gas properties at different states is related by:

\[
\frac{P_1}{V_1^k} = \frac{P_2}{V_2^k} = \frac{P_3}{V_3^k}
\]

(Eq. 23)

For a polytropic compression the actual value of \( n \) (polytropic exponent) is a function of the gas properties such as specific heats, degree of external cooling during compression and operating features of the cylinder. Usual reciprocating compressor performance is evaluated using adiabatic \( C_p/C_v \).

9.2.7 "Power Rating" or kilowatt rating of a compressor is the measure of the ability of the supporting structure and crankshaft to withstand torque and the ability of the bearings, to dissipate frictional heat. The rated power includes the effect of equipment such as pulsation suppression devices, process piping, intercoolers, after coolers, and separators.
9.2.8 "Rod Loads" are established to limit the static and internal loads on the crankshaft, connecting rod, frame, piston rod, bolting and projected bearing surfaces.

9.2.9 Performance calculation

9.2.9.1 Determination of properties pertaining to compression

Compressibility factor (Z factor), ratio of specific heats (C_p/C_v or k value), and molecular mass are three major physical properties for compression which must be clarified. Mollier diagrams should be used if available.

The k value, may be calculated from the ideal gas equation:

\[ K = \frac{M_C P}{C_v} \left( \frac{M_C P}{M_C P - R} \right) \]  

Where:

- \( M_C P \) is molar heat capacity at constant pressure (kJ/kmol.K);
- \( R \) is gas constant, 8.3143 kJ/kmol.K.

Method presented in Tables A.3 and A.4 of Appendix A can be used for calculation of k value of hydrocarbon gases and vapors.

9.2.9.2 Determination of suction conditions

The following conditions at the suction flange should be determined:

a) Temperature

b) Pressure

In case of air taken from atmosphere, corrections should be made for elevation. Air humidity should also be considered.

c) Flow rate

Since the maximum and minimum flow rates are important parameters in compressor selection in some cases, studies must be conducted carefully.

For the purpose of performance calculations, compressor capacity is expressed as the
actual volumetric quantity of gas at inlet to each stage of compression on a per hour basis (Im³/h).

1) Inlet volume

From mass (weight) flow W, (kg/h):

\[ Q = \frac{R}{M} \left( \frac{W \cdot T_1 \cdot Z_1}{P_1 \cdot Z_L} \right) \]  

(Eq. 25)

Where:

- \( R \) is gas constant; 
- \( M \) is molecular mass, in (kg/kmol);
- \( P_1 \) is absolute pressure, in (kPa);
- \( T_1 \) is absolute temperature, in (K);
- \( Z_1 \) is gas compressibility factor at inlet condition;
- \( Z_L \) is gas compressibility factor at standard condition.

From (Sm³/h):

\[ Q = q(Sm^3/h) \left( \frac{101.34}{288.15} \right) \left( \frac{T_1 \cdot Z_1}{P_1 \cdot Z_L} \right) \]  

(Eq. 26)

From (Nm³/h):

\[ Q = q(Nm^3/h) \left( \frac{101.34}{273.15} \right) \left( \frac{T_1 \cdot Z_1}{P_1 \cdot Z_L} \right) \]  

(Eq. 27)

2) Piston displacement

Piston displacement is equal to the net piston area multiplied by the length of the piston sweep in a given period of time:

For a single-acting piston compressing on the outer end only:

\[ PD = \frac{S(r/min)(D^2)}{4 \times 10^9} \times 60 \times 47.124 \times 10^{-9} \times S(r/min)D^2 \]  

(Eq. 28)

For a single acting piston compressing on the
crank end only:

$$PD = \frac{S \pi (D^2 - d^2) r}{4 \times 10^9}$$  
(معادلة 29)

and for a double acting piston (other than rod tail type):

$$PD = \frac{S \pi (2D^2 - d^2) r}{4 \times 10^9}$$  
(معادلة 30)

Where:

- **PD** = جابجایی پیستون (م³/س)  
- **S** = طول ضره (م);  
- **D** = قطر داخلي سیلندر (م);  
- **d** = قطر دسته پیستون (م);  
- **r/min** = سرعت کمپرسور (دور به دقیقه).

3) Cylinder clearance

In a reciprocating compressor the piston does not travel completely to the end of the cylinder at the end of the discharge stroke. Some clearance volume is necessary and it includes the space between the end of the piston and the cylinder head when the piston is at the end of its stroke. It also includes the volume in the valve ports, the volume in the suction valve guards and the volume around the discharge valve seats.

Clearance volume is usually expressed as percent of piston displacement and referred to as percent clearance, or cylinder clearance, **C**.

$$C = \frac{\text{clearance volume}}{\text{piston displacement}} \times 100$$  
(معادلة 31)

For double acting cylinders, the percent clearance is based on the total clearance volume for both the head end and the crank end of a cylinder. These two clearance volumes are not the same due to the presence...
of the piston rod in the crank end of the cylinder. Sometimes additional clearance volume (external) is intentionally added to reduce cylinder capacity.

4) Volumetric efficiency

The term "volumetric efficiency" refers to the actual pumping capacity of a cylinder compared to the piston displacement. Without a clearance volume for the gas to expand and delay the opening of the suction valve(s), the cylinder could deliver its entire piston displacement as gas capacity. The effect of the gas contained in the clearance volume on the pumping capacity of a cylinder can be represented by:

\[
V_E = 100 - C \left[ \frac{Z_s}{Z_d} (r) \right]^{1/3} - 1
\]

Where:

- \( V_E \) is volumetric efficiency;
- \( R = \frac{P_2}{P_1} \) is compression ratio;
- \( C \) is percent clearance;
- \( Z_s \) and \( Z_d \) are gas compressibility at suction and discharge.

Note that volumetric efficiencies as determined by the above equation are theoretical in that they do not account for suction and discharge losses. One method for accounting for losses is to reduce the volumetric efficiency in the following manner:

- 4% for valve losses,
- 5% for gas slippage (for non-lubricated compressors),
- 4% for heavy gases (propane and similar).

5) Actual capacity (actual flow rate)

This is the volume of gas measured at intake to the first stage of a single or multistage compressor, at stated intake temperature and
pressure.

\[ V_a = PD \times VE/100 \text{ m}^3/\text{h} \]  

(Eq. 33)  

(36)  

6) Equivalent capacity

The net capacity of a compressor, in m³/h at 101.325 kPa and suction temperature, may be calculated by equation:

\[ Q_{eq} = \frac{PD \times VE \times P_s}{100 \times 101.325 \times Z_{ave}} \]  

(Eq. 34)  

(37)  

Where:

- \( Q_{eq} \): Equivalent capacity at 101.325 kPa (abs), in (m³/h);
- \( PD \): Piston displacement, in (m³/h);
- \( VE \): Volumetric efficiency (see 4 above);
- \( P_s \): Suction pressure, in [kPa (abs)];
- \( Z_{ave} \): Average compressibility.

If compressibility is not used as a divisor in calculating m³/h, then the statement "not corrected for compressibility" should be added. The figure 101.325 kPa (abs) is used in Eq. 34, because it is the common base pressure for the compression power charts (see 8.2.9.4.2).

9.2.9.3 Determination of discharge conditions

9.2.9.3.1 Discharge temperature

The temperature of the gas discharged from the cylinder can be estimated from the equation:

\[ T_a = T_s \left( r^{(k-1)/k} \right) \]  

(Eq. 35)  

(38)  

Although the result of this equation is theoretical value and heat from friction, irreversibility effects etc., are neglected, use of it has been recommended by many sources. Polytropic exponent "n" may be used instead of "k" in the above equation and will give better results.
9.2.9.3.2 Limitations on discharge temperature

Limitations on the discharge temperature are as shown in API Std. 618, 2.3.

9.2.9.3.3 Estimation of number of compression stages

a) Number of compression stages is a function of compression ratio per stage ($R_{cs}$), these parameters are related by the equations:

\[
R_{cs} = \left(\frac{P_f}{P_1}\right)^{1/\gamma}
\]

and

\[
R_{CO} = \left(\frac{P_f}{P_1}\right)
\]

Where:

- $R_{co}$ is overall compression ratio;
- $P_1$ & $P_f$ are absolute initial and final pressures respectively;
- $\gamma$ (gamma) is number of compression stages.

b) The maximum ratio of compression permissible in one stage is determined considering and limited by the discharge temperature (see 8.2.9.3.2), or by rod loading (see 8.2.8), particularly in the first stage. Economic considerations are also involved, because a high ratio of compression will mean a low volumetric efficiency.

c) In multi-stage operation, equal ratio of compression per stage shall be used, unless otherwise stated by process design. This will result minimum power requirement.

9.2.9.3.4 Interstage cooling

a) Interstage cooling operations affect the cumulative power required to do the work of compression.

Note that if condensate forms in interstage
coolers and is to be removed, the flow rate and properties of the fluid will vary.

b) Cooling water system, where necessary, shall be designed as per Section 2.1.3 of API Standard 618.

c) Compression ratio across stage with intercooling.

Pressure drop in the interstage cooler shall be regarded to be 20 to 35 kPa. Ratio of compression per stage (R) may be calculated by:

\[
P_f = P_1 \cdot R \cdot \frac{1}{R} - (\Delta P_1) \cdot \frac{1}{R^2} - (\Delta P_2) \cdot \frac{1}{R^3} - \Delta (P_3) \cdot \frac{1}{R^4}.
\]

Where:

- \( \Delta P_1, \Delta P_2, \ldots \) are interstage cooler pressure drops of different stages.

Other variables are defined earlier.

Number of terms on right side of this equation should be equal to the number of stages. This equation shall be solved by trial and error method for the case of multi-stage compression.

9.2.9.4 Determination of power required

Brake horse power is the actual power input at the crankshaft of the compressor drive. It does not include the losses in the driver itself, but is rather the actual power which the driver must deliver to the compressor crankshaft. There are three methods for determination of power required for compression. These methods are described in the following section.

9.2.9.4.1 Calculation method

a) Single stage compression

Use basic equation to determine brake horse power:

\[
Bhp = \frac{PVk}{3600(k-1)} \left( \frac{P_1}{P_0} \right)^{(k-1)/k} - 1 \left( L_e (F_e)(Z_1) \right).
\]

Where:

- \( L_e, F_e, Z_1 \) are coefficients.
**b) Multi-Stage compression**

Multi-Stage power is the sum of the power requirements of the individual cylinders on the compressor unit.

\[
\begin{align*}
Bhp & \quad \text{is brake horse power, in kilowatt (kW);} \\
P_1 & \quad \text{is suction pressure, in (kPa);} \\
P_2 & \quad \text{is discharge pressure, in (kPa);} \\
V_1 & \quad \text{is suction volume, in (m³/h), at suction conditions;} \\
L_o & \quad \text{is loss factor, comprised of losses due to pressure drop through friction of piston rings, rod packing, valves, and manifold (see Fig. 10);} \\
F_L & \quad \text{is frame loss for motor driven compressors only, values range from 1.0 to 1.05;} \\
Z_1 & \quad \text{is compressibility factor, based on inlet conditions.}
\end{align*}
\]
(Eq. 40)
\[
\text{Actual Bhp} = \frac{F_k}{3600(k-1)} \left( P_i \left( \frac{P_n}{P_i} \right)^{\frac{k-1}{k}} \right) - 1 \left( L_{o1} + P_i \left( \frac{P_n}{P_i} \right)^{\frac{k-1}{k}} \right) - L_{o2} - \ldots - \sum P_i \left( \frac{P_n}{P_i} \right)^{\frac{k-1}{k}} - L_{od}.
\]

Where:

- \(P_{ni}\): discharge pressure of stage \(i\), in (kPa);
- \(P_i\): inlet pressure to stage \(i\), in (kPa);
- \(V_i\): inlet volume to stage \(i\), in (Am³/h);
- \(L_{o1}, L_{o2}, \ldots, L_{of}\): loss factors designated by cylinder stages. Correction for compressibility may be incorporated as described for the single stage cylinder.

9.2.9.4.2 Power determination by chart

Detailed compressor power calculations can be made through the use Fig. 13-9 of GPSA.

9.2.9.4.3 Power calculation by Mollier diagram

Power calculations can be worked out most easily and that accurately if the P-H (Mollier) diagram is available. The procedure is as follows:

a) The enthalpy at the inlet pressure and temperature shall be calculated. The enthalpy at the outlet pressure shall be found from the diagram following the line of constant entropy:

(Eq. 41) \[
\text{amount of work} = h_2 - h_1
\]

Where \(h_1\) and \(h_2\) are enthalpies at inlet and outlet conditions respectively in (kJ/kg).

b) Brake horse power in kW (Bhp) is calculated from the equation:

(Eq. 42) \[
\text{Bhp} = 2.78 \times 10^{-4} \times W(h_2 - h_1)(L_o)(F_L)
\]

Where:

- \(W\): amount of work (kJ/kg);
- \(L_o\): loss factor designated by cylinder stages.

9.2.9.4.4 Power determination by Mollier diagram

The procedure is as follows:

a) The enthalpy at the inlet pressure and temperature shall be calculated. The enthalpy at the outlet pressure shall be found from the diagram following the line of constant entropy:

(Eq. 43) \[
\text{amount of work} = h_2 - h_1
\]

Where \(h_1\) and \(h_2\) are enthalpies at inlet and outlet conditions respectively in (kJ/kg).

b) Brake horse power in kW (Bhp) is calculated from the equation:

(Eq. 44) \[
\text{Bhp} = 2.78 \times 10^{-4} \times W(h_2 - h_1)(L_o)(F_L)
\]

Where:

- \(W\): amount of work (kJ/kg);
- \(L_o\): loss factor designated by cylinder stages.

9.2.9.4.5 Power calculation by Mollier diagram

Power calculations can be worked out most easily and that accurately if the P-H (Mollier) diagram is available. The procedure is as follows:

a) The enthalpy at the inlet pressure and temperature shall be calculated. The enthalpy at the outlet pressure shall be found from the diagram following the line of constant entropy:

(Eq. 45) \[
\text{amount of work} = h_2 - h_1
\]

Where \(h_1\) and \(h_2\) are enthalpies at inlet and outlet conditions respectively in (kJ/kg).

b) Brake horse power in kW (Bhp) is calculated from the equation:

(Eq. 46) \[
\text{Bhp} = 2.78 \times 10^{-4} \times W(h_2 - h_1)(L_o)(F_L)
\]

Where:

- \(W\): amount of work (kJ/kg);
- \(L_o\): loss factor designated by cylinder stages.
Where:

\[ Bhp \] is in kilowatts;
\[ W \] is mass flow rate of gas, in (kg/h).

9.2.10 Reciprocating compressor control devices

Output of compressors must be controlled (regulated) to match system demand. Compressor capacity, speed, or pressure, may be varied in accordance with the requirements. The nature of the control device will depend on the regulating variable; whether pressure, flow, temperature, or some other variable; and on type of compressor driver.

Compressor control system may be pneumatic, hydraulic, Electrical or electronic, and they may be operated either manually or automatically. When variable-speed driver is specified the speed of the drive shall vary linearly with the control signal. And an increase in signal will increase driver speed. Unless otherwise specified, the full range of the purchaser’s signal shall correspond to the required operating range of the compressor for all specified operating conditions. Reciprocating compressors are usually specified for constant-speed operation.

9.2.10.1 Capacity control

The most common requirement is regulation of capacity. Many capacity controls, or unloading devices, as they are usually termed, are actuated by the pressure on the discharge side of the compressor.

A common method of controlling the capacity of a compressor is varying the speed. This method is applicable to steam driven compressors and to units driven by internal-combustion engines.

On reciprocating compressors up to about 75 kW, two types of control are usually available. These are automatic start-and-stop control and...
constant-speed control.

9.2.10.1.1 Step control

Motor-driven reciprocating compressors above 75 kW in size are usually equipped with a step control. This is in reality a variation of constant-speed control in which unloading is accomplished in a series of steps, varying from full load down to no load.

9.2.10.1.2 Manual control

Although control devices are often automatically operated, manual operation is satisfactory for many services.

9.2.10.2 Instrumentation

When specified, a panel shall be provided and shall include all panel-mounted instruments for the compressor and the driver. The instruments on the panel shall be clearly visible to the operator from the driver control point; A lamp-test push bottom shall be supplied.

A tachometer shall be provided when specified. The type, range, and indicator provision shall be stated by the purchaser. When a turbine driver is to be used, the turbine vendor shall furnish the speed sensor and indicator.

9.2.10.3 Control by spill-back

In some cases, spill-back from discharge to suction will be required for flow or pressure control. This flow rate depends on the design of the unloader. Minimum flow rates are as shown below.

<table>
<thead>
<tr>
<th>Unloader Design</th>
<th>Spill-Back Flow Rate</th>
</tr>
</thead>
<tbody>
<tr>
<td>طرائحي بر پدیدار</td>
<td>میزان برگشت جریان</td>
</tr>
<tr>
<td>5-stage</td>
<td>100</td>
</tr>
<tr>
<td>4-stage</td>
<td>100</td>
</tr>
<tr>
<td>3-stage</td>
<td>100</td>
</tr>
<tr>
<td>2-stage</td>
<td>100</td>
</tr>
</tbody>
</table>

9.2.11 Gas pulsation control

Pulsation is inherent in reciprocating compressors because suction and discharge
Valves are open during only part of the stroke.

Pulsation must be damped (controlled) in order to:

a) Provide smooth flow of gas to and from the compressor;

b) Prevent overloading or underloading of the compressors, and;

c) Reduce overall vibration.

9.2.11.1 Pulsation dampeners (snubbers)

A pulsation dampener is an internally-baffled device. The design of the pulsation dampening equipment is based on acoustical analog evaluation which takes into account the specified operating speed range, conditions of unloading, and variations in gas composition. Detailed discussion of recommended design approaches for pulsation suppression devices is presented in API Std. 618.

10. ROTARY COMPRESSORS

10.1 General

10.1.1 Rotary compressors are positive displacement gas (or vapor) compressing machines. Rotary compressors cover lobe-type, screw-type, vane-type and liquid ring type, each having a casing with one or more rotating elements that either mesh with each other such as lobes or screws, or that displace a fixed volume with each rotation.

10.1.2 Rotary compressors shall conform to API Std. No. 619 for all services handling air or gas, except those machines which this Standard does not cover.

10.1.3 Performance

10.1.3.1 Compressor shall be guaranteed for satisfactory performance at all specified operating conditions.

10.1.3.2 Compressor performance shall be guaranteed at the rated point unless otherwise specified. At this point no negative tolerance is permitted on capacity and power may not exceed 104% of the quoted power.
10.1.4 Cooling water

The compressor cooling water jacket shall be designed for the specified cooling water pressure but not less than 618 kilopascals (absolute). The maximum pressure drop shall be 70 kPa and provisions shall be included for complete draining and venting of the jackets (Modification to API Std. 619). The cooling water design conditions shall be in accordance with API Std. 619.

10.2 Design Criteria

10.2.1 The vendor shall assume unit responsibility for all equipment and all auxiliary systems included in the scope of the order.

10.2.2 Equipment shall be designed to run without damage to the relief valve set pressure, specified maximum differential pressure, and trip speed, simultaneously.

10.2.2.1 There may be insufficient driver power to operate under these conditions.

10.2.2.2 For machines operating with variable suction and discharge pressure levels, maximum allowable temperature can occur before maximum allowable pressure or maximum allowable differential pressure occurs. In such cases the manufacturer and the purchaser should jointly consider and apply suitable safeguarding controls to avoid any damage. Controls may include but are not limited to discharge temperature or differential pressure.

10.2.3 Unless otherwise specified, cooling water systems shall be designed for the following conditions:

- Water velocity over heat exchange surfaces: 1.5-2.5 m/s
- Maximum allowable working pressure (MAWP): >7.0 barg
- Test pressure (1.5 times MAWP): >10.5 barg
- Maximum pressure drop: 1 bar
- Maximum inlet temperature: 32°C
- Maximum outlet temperature: 50°C
- Maximum temperature rise: 17K
- Minimum temperature rise: 10K
- Fouling factor on water side: 0.35 m-K/kW
- Shell corrosion allowance: 3.0 mm
The vendor shall notify the purchaser if the criteria for minimum temperature rise and velocity over heat exchange surfaces result in a conflict. The criteria for velocity over heat exchange surfaces is intended to minimize water-side fouling; the criterion for minimum temperature rise is intended to minimize the use of cooling water. If such a conflict exists, the purchaser will approve the final selection.

10.2.4 All equipment shall be designed to permit rapid and economical maintenance. Major parts such as casing components and bearing housings shall be designed and manufactured to ensure accurate alignment on reassembly. This may be accomplished by the use of shouldering, cylindrical dowels or keys.

10.2.5 The equipment’s maximum continuous speed shall be not less than 105% of the rated speed for variable speed machines and shall be equal to the rated speed for constant speed motor drives.

10.2.6 The equipment’s trip speed shall not be less than the values in Table 3.

10.2.7 Oil reservoirs and housings that enclose moving lubricated parts such as bearings, shaft seals, highly polished parts, instruments, and control elements shall be designed to minimize contamination by moisture, dust, and other foreign matter during periods of operation and idleness.

**TABLE 3—DRIVER TRIP SPEEDS**

<table>
<thead>
<tr>
<th>Driver Type</th>
<th>Trip Speed (% of maximum Continuous Speed)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Steam Turbine</td>
<td>Torbin بخار</td>
</tr>
<tr>
<td>Nema Class Aa</td>
<td>Antifreeze بندي Aa</td>
</tr>
<tr>
<td>Nema Class B,C,Da</td>
<td>Antifreeze بندي B,C,Da</td>
</tr>
<tr>
<td>Gas Turbine</td>
<td>Torbin غاز</td>
</tr>
<tr>
<td>Variable Speed Motor</td>
<td>موتر دور متغير</td>
</tr>
<tr>
<td>Constant Speed Motor</td>
<td>موتر دور ثابت</td>
</tr>
<tr>
<td>Reciprocating Engine</td>
<td>موتر رف و برکشی</td>
</tr>
</tbody>
</table>

*Indicates Governor class as specified in NEMA SM 23.*

The vendor shall notify the purchaser if the criteria for minimum temperature rise and velocity over heat exchange surfaces result in a conflict. The criteria for velocity over heat exchange surfaces is intended to minimize water-side fouling; the criterion for minimum temperature rise is intended to minimize the use of cooling water. If such a conflict exists, the purchaser will approve the final selection.

10.2.4 All equipment shall be designed to permit rapid and economical maintenance. Major parts such as casing components and bearing housings shall be designed and manufactured to ensure accurate alignment on reassembly. This may be accomplished by the use of shouldering, cylindrical dowels or keys.

10.2.5 The equipment’s maximum continuous speed shall be not less than 105% of the rated speed for variable speed machines and shall be equal to the rated speed for constant speed motor drives.

10.2.6 The equipment’s trip speed shall not be less than the values in Table 3.

10.2.7 Oil reservoirs and housings that enclose moving lubricated parts such as bearings, shaft seals, highly polished parts, instruments, and control elements shall be designed to minimize contamination by moisture, dust, and other foreign matter during periods of operation and idleness.
10.2.8 The power at the certified point shall not exceed 104% of the quoted value with no negative tolerance on required capacity.

10.2.9 The purchaser shall specify gas composition(s). The purchaser may also specify molecular weight, ratio of specific heats (Cp/Cv), and compressibility factor (Z).

10.2.10 Unless otherwise specified, the vendor shall use the specified values of flow, the specified gas composition, and the gas conditions to calculate molecular weight, ratio of specific heats (Cp/Cv), and compressibility factor (Z). The compressor vendor shall indicate his values on the data sheets with the proposal and use them to calculate performance data.

10.2.11 The equipment, including all auxiliaries, shall be suitable for operation under the environmental conditions specified by the purchaser. These conditions shall include whether the installation is indoors (heated or unheated) or outdoors (with or without a roof), maximum and minimum temperatures, unusual humidity, and dusty or corrosive conditions.

10.2.12 The equipment, including all auxiliaries, shall be suitable for operation, using the utility stream conditions specified by the purchaser.

10.2.13 The allowable tensile stress used in the design of the pressure casing (excluding bolting) for any material shall not exceed 0.25 times the minimum ultimate tensile strength for the material at the maximum specified operating temperature. For cast materials, the allowable tensile stress shall be multiplied by the appropriate casting factor as shown in Table 4.

10.2.14 The maximum allowable working pressure of the casing shall be at least equal to the specified relief valve set pressure. If a relief valve set pressure is not specified by the vendor, the purchaser shall indicate his values.

<table>
<thead>
<tr>
<th>جدول 4 — ضرایب ریخته گری</th>
<th>Casting factor</th>
</tr>
</thead>
<tbody>
<tr>
<td>Type of NDE</td>
<td>عامل ریخته گری</td>
</tr>
<tr>
<td>Visual, magnetic particle and/or liquid penetrant</td>
<td>0.8</td>
</tr>
<tr>
<td>رادیوگرافی افقی</td>
<td>0.9</td>
</tr>
<tr>
<td>Full radiography</td>
<td>1.0</td>
</tr>
</tbody>
</table>

10.2.9.2 یک مدلی از فشارهای تغییری در حال حرارتی پوشته فشاری (هیچ یک) برای هر مدلی باید از 1/4 برای حداکثر مقاومت تنشی نهایی برازش در جدول دامی عملیاتی تجاوز نماید. برای مواد ریخته گری شده، نشان دهنده کششی مجاز باید در ضریب ریخته گری مناسب نشان داده شده در جدول 4 ضریب شود.

10.2.10 یک مدلی از فشارهای تغییری در حال حرارتی پوشته فشاری (هیچ یک) برای هر مدلی باید از 1/4 برای حداکثر مقاومت تنشی نهایی برازش در جدول دامی عملیاتی تجاوز نماید. برای مواد ریخته گری شده، نشان دهنده کششی مجاز باید در ضریب ریخته گری مناسب نشان داده شده در جدول 4 ضریب شود.
purchaser, it must be specified by the vendor.

10.2.15 Casings designed for more than one maximum allowable working pressure are not permitted. When a cooling jacket is utilized, this jacket shall have only external connections between the upper and lower housings.

10.2.16 Rotor stiffness shall be adequate to prevent contact between the rotor bodies and the casing and between gear-timed rotor bodies at the most unfavorable specified conditions. Rotor bodies not integral with the shaft shall be permanently attached to the shaft to prevent relative motion under any condition. Structural welds on rotors shall be full-penetration continuous welds and shall be stress relieved, with appropriate ASTM heat treatment procedure.

10.2.17 Shafts shall be forged steel unless otherwise approved by the purchaser.
### TABLE A.1 – GENERAL COMPRESSOR LIMITS

<table>
<thead>
<tr>
<th>COMPRESSOR TYPE</th>
<th>APPROX. max. COMMERCIAL DISCH. PRESS. kPa</th>
<th>APPROX. max. COMPRESSION RATIO PER STAGE</th>
<th>APPROX. max. COMPRESSION RATIO PER CASE OR MACHINE</th>
</tr>
</thead>
<tbody>
<tr>
<td>Reciprocating</td>
<td>240,000-345,000</td>
<td>10</td>
<td>As required</td>
</tr>
<tr>
<td>Centrifugal</td>
<td>20,600-34,500</td>
<td>3-4.5</td>
<td>8-10</td>
</tr>
<tr>
<td>Rotary displacement</td>
<td>690-896</td>
<td>4</td>
<td>4</td>
</tr>
<tr>
<td>Axial flow</td>
<td>550-896</td>
<td>1.2-1.5</td>
<td>5-6.5</td>
</tr>
</tbody>
</table>

### Diagram:

```
Compressors
 Positive Displacement Type
    Reciprocating
      Single-Stage
      Multi-Stage
      Integral Gas-Engine Driven Separable Balanced/Opposed
      Diaphragm
    Rotary
      Straight Lobe Helical Lobe (Screw)
      Sliding Vane
      Liquid-Ring
  Dynamic Type
    Radial Flow (Centrifugal)
    Axial Flow
      Single-Stage Multi-Stage Horizontally Split Vertically Split (Barrel) Integral Gear
      Multi-Stage Fixed Stator Vanes Variable Stator Vanes
      Mixed Flow
  Thermal Type
    Ejectors
      Single-Stage Multi-Stage
```
کمپرسورها

- نوع جابجایی مثبت
  - رفت و برگشت
  - دورانی

- نوع متحرک
  - جریان شعاعی (گریز از مرکز)
  - جریان محوری

- نوع حرارتی
  - مکنده

- جریان مختلط

Fig. A.1-TYPES OF COMPRESSORS
شکل الف-1- انواع کمپرسورها
TABLE A.2 – ATMOSPHERIC PRESSURE vs ELEVATION

<table>
<thead>
<tr>
<th>ALTITUDE (meters)</th>
<th>AVERAGE ATMOSPHERIC PRESSURE [kPa (abs)]</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>101.325</td>
</tr>
<tr>
<td>100</td>
<td>99.97</td>
</tr>
<tr>
<td>200</td>
<td>98.84</td>
</tr>
<tr>
<td>300</td>
<td>97.93</td>
</tr>
<tr>
<td>400</td>
<td>96.60</td>
</tr>
<tr>
<td>500</td>
<td>95.44</td>
</tr>
<tr>
<td>600</td>
<td>94.54</td>
</tr>
<tr>
<td>700</td>
<td>93.49</td>
</tr>
<tr>
<td>800</td>
<td>92.04</td>
</tr>
<tr>
<td>1000</td>
<td>90.03</td>
</tr>
<tr>
<td>1200</td>
<td>87.77</td>
</tr>
<tr>
<td>1400</td>
<td>85.51</td>
</tr>
<tr>
<td>1600</td>
<td>83.42</td>
</tr>
<tr>
<td>2000</td>
<td>79.41</td>
</tr>
<tr>
<td>2500</td>
<td>74.58</td>
</tr>
<tr>
<td>3000</td>
<td>70.06</td>
</tr>
<tr>
<td>3500</td>
<td>65.54</td>
</tr>
<tr>
<td>4000</td>
<td>61.40</td>
</tr>
<tr>
<td>4500</td>
<td>57.71</td>
</tr>
<tr>
<td>5000</td>
<td>54.31</td>
</tr>
</tbody>
</table>
Fig. A.2-COMPRESSOR COVERAGE CHART

1- recip- multi stage
2- recip single stage
3- center single stage
4- center multi stage
5- rotary-screw
6- axial
7- rotary liquid ring
8- rotary straight lobe
9- rotary sliding vane
10- diaphragm

1- رفت و برگشتی چند مرحله ای
2- رفت و برگشت تک مرحله ای
3- گوییز از مرکز تک مرحله ای
4- گوییز از مرکز چند مرحله ای
5- چرخشی- عارضه
6- محوری
7- دورانی حلقوی- منابع
8- بادامک دورانی مستقیم
9- دورانی پره ای لنز
10- دیافراگم
### TABLE A.3 - CALCULATION OF k

<table>
<thead>
<tr>
<th>Component Name</th>
<th>Mole Fraction</th>
<th>Mol. Mass</th>
<th>Individual Component Mol. Mass (M × Y)</th>
<th>Individual Component MCp at 70°C (y × MCp)</th>
<th>Component Critical Pressure (Pc, kPa (abs))</th>
<th>y × Pc</th>
<th>Component Critical Temperature (Tc)</th>
<th>y × Tc</th>
</tr>
</thead>
<tbody>
<tr>
<td>Methane</td>
<td>0.9216</td>
<td>16.04</td>
<td>14.782</td>
<td>37.471</td>
<td>4640.4</td>
<td>460.4</td>
<td>190.6</td>
<td>175.6</td>
</tr>
<tr>
<td>Ethane</td>
<td>0.0488</td>
<td>30.07</td>
<td>1.467</td>
<td>58.39</td>
<td>4944.4</td>
<td>494.4</td>
<td>241.3</td>
<td>305.6</td>
</tr>
<tr>
<td>Propane</td>
<td>0.0185</td>
<td>44.10</td>
<td>0.816</td>
<td>82.85</td>
<td>4256.4</td>
<td>425.6</td>
<td>370.0</td>
<td>6.8</td>
</tr>
<tr>
<td>i-Butane</td>
<td>0.0039</td>
<td>58.12</td>
<td>0.227</td>
<td>109.397</td>
<td>3749.0</td>
<td>374.9</td>
<td>406.9</td>
<td>1.6</td>
</tr>
<tr>
<td>n-Butane</td>
<td>0.0055</td>
<td>58.12</td>
<td>0.320</td>
<td>109.497</td>
<td>3658.6</td>
<td>365.8</td>
<td>425.2</td>
<td>2.3</td>
</tr>
<tr>
<td>i-Pentane</td>
<td>0.0017</td>
<td>72.15</td>
<td>0.123</td>
<td>134.379</td>
<td>3333.2</td>
<td>333.3</td>
<td>460.9</td>
<td>0.8</td>
</tr>
<tr>
<td>Total=</td>
<td>1.0000</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

\[ MC_v = MC_p - 8.3143 = 31.859 \]
\[ k = MC_p/MC_v = 40.173/31.859 = 1.261 \]

**Note:**
For values of MCp other than at 70°C refer to Table A.4.
<table>
<thead>
<tr>
<th>Gas</th>
<th>Chemical formula</th>
<th>Mol wt</th>
<th>Temperature</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td>0°F</td>
</tr>
<tr>
<td>Methane</td>
<td>CH₄</td>
<td>16.043</td>
<td>8.23</td>
</tr>
<tr>
<td>Ethane</td>
<td>C₂H₆</td>
<td>30.070</td>
<td>11.44</td>
</tr>
<tr>
<td>Propene (Propylene)</td>
<td>C₃H₆</td>
<td>42.081</td>
<td>13.63</td>
</tr>
<tr>
<td>Propane</td>
<td>C₃H₈</td>
<td>44.097</td>
<td>15.65</td>
</tr>
<tr>
<td>1-Butene (Butylyene)</td>
<td>C₄H₈</td>
<td>56.108</td>
<td>17.96</td>
</tr>
<tr>
<td>cis-2-Butene</td>
<td>C₄H₈</td>
<td>56.108</td>
<td>16.54</td>
</tr>
<tr>
<td>trans-2-Butene</td>
<td>C₄H₈</td>
<td>56.108</td>
<td>18.84</td>
</tr>
<tr>
<td>iso-Butane</td>
<td>C₄H₁₀</td>
<td>58.123</td>
<td>20.40</td>
</tr>
<tr>
<td>n-Butane</td>
<td>C₄H₁₀</td>
<td>58.123</td>
<td>20.80</td>
</tr>
<tr>
<td>iso-Pentane</td>
<td>C₅H₁₂</td>
<td>72.150</td>
<td>24.94</td>
</tr>
<tr>
<td>n-Pentane</td>
<td>C₅H₁₂</td>
<td>72.150</td>
<td>25.64</td>
</tr>
<tr>
<td>Benzene</td>
<td>C₆H₆</td>
<td>78.114</td>
<td>16.41</td>
</tr>
<tr>
<td>n-Hexane</td>
<td>C₆H₁₄</td>
<td>86.177</td>
<td>30.17</td>
</tr>
<tr>
<td>n-Heptane</td>
<td>C₇H₁₆</td>
<td>100.204</td>
<td>34.96</td>
</tr>
<tr>
<td>Ammonia</td>
<td>NH₃</td>
<td>17.0305</td>
<td>8.52</td>
</tr>
<tr>
<td>Air</td>
<td></td>
<td></td>
<td>8.94</td>
</tr>
<tr>
<td>Water</td>
<td>H₂O</td>
<td>18.0153</td>
<td>7.98</td>
</tr>
<tr>
<td>Oxygen</td>
<td>O₂</td>
<td>31.9988</td>
<td>6.97</td>
</tr>
<tr>
<td>Hydrogen sulfide</td>
<td>H₂S</td>
<td>34.08</td>
<td>8.00</td>
</tr>
<tr>
<td>Carbon monoxide</td>
<td>CO</td>
<td>28.019</td>
<td>6.95</td>
</tr>
<tr>
<td>Carbon dioxide</td>
<td>CO₂</td>
<td>44.010</td>
<td>8.38</td>
</tr>
</tbody>
</table>
### APPENDIX B

#### B.1 Process Specification Sheet for Rotodynamic Compressors:

The minimum required information to be included in process specification sheet shall be as follows:

| 
| 
| **a) Process requirements** | 
| - Flow at 1.013 bar (abs) and 0°C, جریان در 13/1 بار (مطلق) و صفر درجه سانتی‌گراد | $m^3/h$, (Normal & Design). 
| - Suction Pressure, فشار مکش | kPa(abs), (Normal & Design). 
| - Discharge Pressure, فشار خروجی | kPa(abs), (Normal & Design). 
| - Discharge Temperature Limitation (if any) محدوده دمای خروجی (در صورت وجود) |  
| - Compression Ratio نسبت تراکم |  
| - Approx. $C_p/C_v$ (at Suction) تقریبی $C_p/C_v$ (در مکش) |  
| - Compressibility Factor (at Suction) ضریب تراکم پذیری (در مکش) |  
| - Estimated Polytropic Head فشار معادل ارتفاع تقریبی پلی تروپیک | m, (Normal & Design). 
| - Estimated Bhp Required نیروی اسب بخار ترمیمی | kW, (Normal & Design). 
| - Recommended Driver kW توان نصب دندنه گردانده | kW, (Normal & Design). 
| - Compressor Speed سرعت کمپرسور | r/min, (Normal & Design). |

| 
| 
| **b) Service** |
| - Approximate Gas Composition (vol% or mol%). ترکیب تقریبی گاز (درصد مولی یا درصد حجمی) |  

---

**طراحی و طراحی**

دوانی متحرک

حداکثر اطلاعات لازم که به صفحه مشخصات فرآیندی افزوده

میشوین بايد مطابق زیر باشد:

الف) الامات فرآیندی
c) Site informations

- Elevation of Plant Site from Sea Level, m.
- Minimum Winter Temperature, °C.
- Maximum Summer Temperature, °C.
- Normal Barometer, kPa.
- Relative Humidity for Process Design %

<table>
<thead>
<tr>
<th>Metric</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Elevation of Plant Site from Sea Level</td>
<td>m.</td>
</tr>
<tr>
<td>Minimum Winter Temperature</td>
<td>°C.</td>
</tr>
<tr>
<td>Maximum Summer Temperature</td>
<td>°C.</td>
</tr>
<tr>
<td>Normal Barometer</td>
<td>kPa.</td>
</tr>
<tr>
<td>Relative Humidity for Process Design</td>
<td>%</td>
</tr>
</tbody>
</table>

d) Available utilities

<table>
<thead>
<tr>
<th>Metric</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cooling water: max. Inlet Temperature</td>
<td>°C.</td>
</tr>
<tr>
<td>max. Outlet Temperature</td>
<td>°C.</td>
</tr>
<tr>
<td>Pressure</td>
<td>kPa</td>
</tr>
<tr>
<td>Fouling Factor</td>
<td></td>
</tr>
<tr>
<td>Instrument Air Pressure</td>
<td>kPa</td>
</tr>
<tr>
<td>Electric Power for Instruments, volts, Phase</td>
<td>Hz.</td>
</tr>
</tbody>
</table>

e) Compressor Location (Outdoor, Indoor).

f) Instrument Graduation System.

g) Remarks on Control System.

B.2 Process Specification Sheet for Positive Displacement Compressors

The minimum required information to be included in process specification sheet shall be as follows:
### a) Process requirements

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Unit</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>Flow at 1.013 bar (abs) and 0°C,</td>
<td>m³/h</td>
<td>Normal</td>
</tr>
<tr>
<td>Flow at Suction Conditions,</td>
<td>m³/h</td>
<td>Rated</td>
</tr>
<tr>
<td>Suction Temperature</td>
<td>°C</td>
<td></td>
</tr>
<tr>
<td>Suction Pressure</td>
<td>kPa (abs)</td>
<td></td>
</tr>
<tr>
<td>Discharge Pressure</td>
<td>kPa (abs)</td>
<td>Normal</td>
</tr>
<tr>
<td>Discharge Pressure</td>
<td>kPa (abs)</td>
<td>Rated</td>
</tr>
<tr>
<td>Differential Pressure</td>
<td>kPa, rated</td>
<td></td>
</tr>
<tr>
<td>Compression Ratio</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Approx. Cₚ/Cᵥ, (at Suction).</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Compressibility Factor (at Suction).</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Mass Flow Molecular Mass,</td>
<td>kg/h</td>
<td>Normal</td>
</tr>
<tr>
<td>Mass Flow Molecular Mass,</td>
<td>kg/h</td>
<td>Rated</td>
</tr>
<tr>
<td>Estimated Power Required,</td>
<td>kW</td>
<td>Normal</td>
</tr>
<tr>
<td>Estimated Power Required,</td>
<td>kW</td>
<td>Rated</td>
</tr>
<tr>
<td>Estimated Gear loss,</td>
<td>kW</td>
<td></td>
</tr>
<tr>
<td>Recommended Driver Bhp.,</td>
<td>kW</td>
<td></td>
</tr>
<tr>
<td>Compressor Speed Limitation (if any),</td>
<td>r/min</td>
<td></td>
</tr>
<tr>
<td>Piston Speed Limitation, (if any),</td>
<td>m/s</td>
<td></td>
</tr>
<tr>
<td>Approximate Gas Composition</td>
<td>(vol% or mol%).</td>
<td></td>
</tr>
<tr>
<td>Average Molecular Mass.</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Relative Density (Specific Gravity).</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Corrosiveness and Relevant Remarks.</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

### b) Service

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Unit</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>Approximate Gas Composition</td>
<td>(vol% or mol%).</td>
<td></td>
</tr>
<tr>
<td>Average Molecular Mass.</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Relative Density (Specific Gravity).</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Corrosiveness and Relevant Remarks.</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
c) Site informations

<table>
<thead>
<tr>
<th>Description</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Elevation of Plant Site From See Level</td>
<td>m</td>
</tr>
<tr>
<td>Minimum Winter Temperature</td>
<td>°C</td>
</tr>
<tr>
<td>Maximum Summer Temperature</td>
<td>°C</td>
</tr>
<tr>
<td>Normal Barometer</td>
<td>kPa</td>
</tr>
<tr>
<td>Relative Humidity for Process Design</td>
<td>%</td>
</tr>
</tbody>
</table>

d) Available utilities

<table>
<thead>
<tr>
<th>Description</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cooling water: max. Inlet Temperature</td>
<td>°C</td>
</tr>
<tr>
<td>max. Outlet Temperature</td>
<td>°C</td>
</tr>
<tr>
<td>Fouling Factor</td>
<td>kPa</td>
</tr>
<tr>
<td>Instrument Air Pressure</td>
<td>kPa</td>
</tr>
<tr>
<td>Fuel Gas for Engine:</td>
<td>mol. mass</td>
</tr>
<tr>
<td>- Gas composition by mole</td>
<td></td>
</tr>
<tr>
<td>Rel. Density</td>
<td></td>
</tr>
<tr>
<td>- kJ/m³, Gross</td>
<td></td>
</tr>
<tr>
<td>Calorific value</td>
<td></td>
</tr>
<tr>
<td>Pressure</td>
<td>kPa</td>
</tr>
<tr>
<td>Electric Power for Instruments</td>
<td>volt, Phase, Hz</td>
</tr>
</tbody>
</table>

e) Compressor Location, Outdoor, Indoor.

f) Instrument Graduation System.

g) Remarks on Control System.
Fig. C.1-TYPICAL SCREW COMPRESSOR

Key:
1. Casing
2. Male rotor
3. Female rotor
4. Shaft seal
5. Radial/thrust bearing
6. Timing gear
7. End cover
8. Drive shaft

كلید:
1- بدنه 2- گردنه نری 3- گردنه مادگی 4- نشت بندی محور 5- پاتاقان محوری 6- دندی زمانی 7- پوشش انتهایی 8- محور گردانندگی
Fig. C.2-TYPICAL CENTRIFUGAL COMPRESSOR

تشکل ج-2- نوع کمپرسور گریز از مرکز
Fig. C-3-TYPICAL RECIPROCATING COMPRESSOR

شکل ج-3-نمونه کمپرسور رفته و برگشتی