

Article

Resource-Efficient Model of an Offshore Platform Compressor Station for the Declining Production Period of a Gas-Condensate Field

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Abstract

Under conditions of gradual depletion of gas-condensate field reserves and declining well production rates, one of the key challenges in operating offshore production facilities is improving the energy efficiency and reliability of process equipment, particularly compressor stations used for gas gathering, treatment, and transportation. Resource conservation and the adaptation of operating modes to changing production conditions require flexible compression systems capable of operating efficiently over a wide range of inlet flow rates and pressures. This article examines examples from three key regions where compressor stations have been adapted to declining production rates and reservoir pressures, and analyses the challenges encountered under actual operating conditions. A compressor model is proposed for the late stage of field development, during which the production rate of gas-condensate wells decreases from 5.0 million m³/day to 0.5 million m³/day over a ten-year period. The wellhead pressure and compressor suction pressure decline from 5.0 MPa to 0.1 MPa, while the discharge pressure remains at 5.5 MPa. Calculations were performed for the principal operating parameters, including gas production rate, wellhead and compressor suction pressures, the capacity of a single compressor under different configurations, the required number of compressors, and the total annual capacity of all operating compressors. The proposed compressor-unit reconfiguration scheme makes it possible to compress the produced gas for at least nine years using a single unit size, without replacing cylinders and without changing the type of compressor; over this period the number of units operating in parallel increases from three to six. Unlike a single oversized centrifugal machine, whose efficiency falls sharply once anti-surge recycling becomes necessary at part load, the proposed configuration is estimated to remain operable down to about 10% of the rated capacity of one unit. This lower limit is obtained by extrapolation of the model presented here and requires confirmation by test or manufacturer data.

Keywords: gas-condensate well; declining production; centrifugal compressor; reciprocating compressor; model; operating efficiency range

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1. Introduction

At the late stage of gas-condensate field development, reservoir pressure naturally declines, well production rates decrease, and the proportion of liquid phases, including condensate and water, as well as sand production, increases. These changes lead to unstable operation of conventional centrifugal compressors and a reduction in their efficiency. Centrifugal compressors, which are widely used on offshore platforms, have a limited operating range in terms of flow rate and pressure. When the production rate falls below 1.0 million m³/day, they operate outside their optimal performance range, resulting in lower efficiency, frequent shutdowns, the need to activate bypass lines, increased specific energy consumption, and higher operating costs.

One promising solution is the application of reciprocating gas-engine-driven compressors, which can maintain stable operation under variable flow rates ranging from 0.5 to 5.0 million m³/day, allow smooth adjustment of suction pressure within the range of 0.1–5.0 MPa, provide high efficiency of up to 0.85 over a broad load range [1,2], exhibit low sensitivity to gas composition and thermobaric conditions, and can be added to existing offshore platforms as self-contained modules [3,4].

Under current conditions, where energy efficiency and resource conservation are becoming key priorities for the sustainable development of the oil and gas industry, the development of a resource-efficient compressor station model adapted to declining production conditions on offshore platforms is of considerable practical and scientific importance.

Such a solution is directed at extending the operating life of the existing infrastructure, at maintaining the deliverability of the gas transportation system during the final stage of field development and, through this, at increasing the hydrocarbon recovery factor. The effects on energy consumption and on greenhouse-gas emissions are not quantified in the present work; the scope of the study is stated below.

The aim of this study is to develop and substantiate a resource-efficient model of an offshore platform compressor station that ensures energy-efficient operation under declining production conditions in a gas-condensate field. The model is based on the use of reciprocating gas-engine-driven compressors with adaptive suction-pressure control and is intended to improve the efficiency of energy and material resource utilisation, extend the service life of equipment and infrastructure, and maintain the required compression duty as production conditions change.

To achieve this aim, the study addresses four objectives. The first is to analyse the current state of compressor-station operation on offshore platforms, determining the operating characteristics of conventional centrifugal compressors at the late stage of field development and identifying the principal limitations and energy-efficiency problems associated with declining production rates and pressures. The second is to examine the feasibility of using reciprocating gas-engine-driven compressors under these conditions by means of a comparative review of the technical characteristics of units manufactured by AJAX (Cooper Machinery Services, Houston, TX, USA), Dresser-Rand (Houston, TX, USA), Ariel (Mount Vernon, OH, USA) and other companies [3,4], with particular attention to a flow-rate range of 0.5–5.0 million m³/day and a suction-pressure range of 0.1–5.0 MPa. The third is to develop a conceptual model of compressor-station operation with adaptive control, in which the compressor suction pressure is related to the current gas production rate. The fourth is to formulate a transparent calculation procedure that determines, for each year of the declining-production period, the capacity of a single unit, the required number of units and the total capacity of the station, and to derive on this basis recommendations for upgrading existing offshore compressor stations.

The scope of the present work is deliberately limited to this technological concept and to the associated capacity calculation. A quantitative comparison of energy

consumption with a conventional centrifugal scheme, and an assessment of fuel savings, greenhouse-gas emissions, operating expenditure and capital investment, require machine-specific performance maps and field-specific cost data; these belong to the detailed design stage for a particular field and are therefore not claimed in this article.

The problem addressed here continues a line of work by the present authors on the processing and transport of natural gas and produced fluids, which has included supersonic flow processes in gas treatment [5], the prevention of hydrate formation in subsea gas pipelines [6] and the blending of produced crude oils [7]. The importance of maintaining productivity at the late stage of development of large gas assets has also been demonstrated by integrated asset modelling studies [8].

2. Operating Experience of Offshore Compressor Stations Under Declining Production

2.1. Offshore Compressor Stations and the Limitations of Centrifugal Machines

Compressor stations installed on offshore production platforms are key components of the technological system used for gathering and processing gas-condensate production. Their primary function is to provide the pressure required for transporting gas through subsea pipelines [6], while also stabilising production rates and maintaining optimal well operating conditions. During the early stages of field development, when reservoir pressure considerably exceeds the pressure required for gas transportation, compressor stations are either not used or operate at partial load. However, as the reservoir becomes depleted and system pressure declines, compression must be introduced gradually to maintain production rates and ensure the required discharge pressure, which is typically 5.0–6.0 MPa for offshore trunk pipelines.

A typical offshore compressor station includes one to three centrifugal compressor units operating in duty and standby modes; gas-turbine or electric drives, whose relative merits have been assessed for transmission compression duty [9]; separation equipment for removing condensate and moisture from the gas stream before compression; cooling and lubrication systems; and automated control and protection systems. Centrifugal compressors are used in most projects because of their high capacity, compact design, and reliability under stable-flow conditions. They are particularly efficient at high gas flow rates exceeding 2–3 million m³/day and within a relatively limited operating range, typically 70–100% of the rated capacity.

Offshore equipment is subject to stringent requirements, including limitations on weight and dimensions, high reliability and operational autonomy due to maintenance difficulties, resistance to vibration and corrosion, and minimal personnel involvement through remote operation [10].

Centrifugal compressors used on offshore platforms are designed with these requirements in mind. Nevertheless, they have several significant operational limitations when process conditions change.

During the declining-production stage, when the gas flow rate decreases several times below its initial level, centrifugal compressors begin operating outside their optimal performance range. The main problems include unstable compression at low flow rates, potentially leading to surge; the need for throttling and flow recycling through bypass lines, which reduces efficiency; increased specific energy consumption; unstable suction pressure; higher loads on bearings and seals; and reduced intervals between maintenance operations.

In addition, centrifugal compressors are sensitive to gas composition, particularly the presence of condensate and moisture, and require relatively stable flow conditions. Such stability is difficult to maintain when wells operate unevenly.

The stable operating range of centrifugal compressors is generally limited to approximately 60–100% of their rated capacity. When the gas flow rate declines further, it becomes necessary either to shut down some compressor units or to install additional compression stages. Under offshore-platform conditions, such measures are often technically difficult and economically unjustified.

Consequently, the operation of conventional compressor stations during the late stage of field development leads to reduced energy efficiency, excessive fuel consumption by compressor drives, increased operating costs, and accelerated equipment wear.

Thus, conventional centrifugal compressor stations have demonstrated reliable performance during the early and intermediate stages of field development but become inefficient when operating under low-flow and low-pressure conditions. This creates an objective need for new resource-efficient solutions capable of providing operational flexibility and adapting compression conditions to changes in field performance.

When a field enters the declining-production stage, centrifugal compressors exhibit several limitations, including a narrow stable operating range, a sharp decrease in efficiency at low flow rates, the need for continuous gas recycling through bypass lines, and increased specific energy consumption. These factors make centrifugal compressors economically unattractive for long-term operation at flow rates significantly below their rated capacity.

2.2. Part-Load Behaviour of Centrifugal and Reciprocating Machines

The part-load behaviour of centrifugal and reciprocating machines has been quantified in the peer-reviewed literature. For offshore gas-processing trains operating at continuously decreasing gas loads, Cruz et al. [11] report that a single-shaft centrifugal arrangement sized for the maximum load retains an exergy efficiency of about 83% at full gas load but falls to about 49% at 25% of the design gas load, because anti-surge recycling keeps the absorbed power almost constant irrespective of the actual throughput. In the same study, an arrangement of several smaller compressors operating in parallel remains above 80% over the whole load range and lowers the carbon dioxide emission intensity by about 34%. Exergy audits of North Sea platforms likewise identify gas compression as one of the largest single sources of exergy destruction in the topside process [12].

Reciprocating machines behave differently at part load, because the delivered volume is determined by the swept volume actually admitted to the cylinders rather than by an aerodynamic match between impeller and diffuser. Suction-valve unloading and stepless capacity regulation allow the delivered volume to be varied continuously while the absorbed power decreases in approximately the same proportion, so that neither recycling nor a surge limit is involved [1,2]. This behaviour is the physical basis of the configuration proposed in Section 3.4, in which the delivered volume is adjusted by changing the number of cylinders admitted to the first compression stage.

From a practical perspective, centrifugal compressors generally maintain stable operation at flow rates above approximately 60% of their rated capacity. At lower flow rates, costly modifications or a transition to alternative compression technologies may be required.

The practical implications and recommendations are as follows. When designing or upgrading compressor stations for mature offshore fields, it should be recognised that centrifugal compressors are most effective at high and stable flow rates, generally above 60–70% of their rated capacity. For periods involving prolonged operation at low gas flow rates of 0.5–2.0 million m³/day, reciprocating gas-engine-driven compressors or hybrid configurations, combining a centrifugal compressor with a reciprocating unit operating as a booster, should be considered.

Under offshore-platform conditions characterised by limited available power and high inspection and maintenance costs, reciprocating compressors frequently provide a more favourable balance between service life and economic performance.

2.3. International Operating Experience

In international practice, the development of offshore gas-condensate fields during their late production stage increasingly focuses on adapting compressor stations to declining gas flow rates and reservoir pressures. Examples from three major regions—the North Sea, the Caspian Sea, and the Gulf of Mexico—are considered below [13,14].

2.3.1. North Sea: Norway and the United Kingdom

Troll Project, Equinor, Norway. The field has been in operation since 1996. As reservoir pressure declined, projects were implemented to install additional compressor modules on the Troll A and Troll C platforms. Centrifugal compressors are used, together with the gradual reduction in suction pressure from approximately 12 MPa to 5–6 MPa and the upgrading of compression stages. At the late stage of field development, modular reciprocating booster compressors are introduced to compress low-rate tail-end gas production.

Åsgard Field Compression Project [15]. This was one of the world's first subsea compression projects. However, the project demonstrated that, under conditions of substantial production decline, the efficiency of centrifugal compression systems decreases and sophisticated control systems with high energy requirements are needed.

Where the gas flow rate falls substantially below the rated capacity of the installed machines, a transition to reciprocating compressors or to hybrid compression configurations may become appropriate.

Ormen Lange Project, Norway. Subsea compression has been installed on this field in order to counteract the decline in reservoir pressure; the system entered operation in 2025, after which the operator reported an increase in gas deliverability from the field [16,17]. The subsea multiphase compression solution applied there is estimated to raise the recovery factor of the field by about 10% [18,19]. The example confirms that added compression is the principal means of extending the productive life of a depleting offshore gas field, while at the same time illustrating the scale of investment that a purpose-built subsea compression facility requires.

In the North Sea, preference is generally given to centrifugal compressors combined with staged upgrading. However, modernisation costs are high, and efficiency decreases sharply under low-flow operating conditions. Hybrid systems and modular reciprocating solutions are therefore considered alternatives for extending the operating life of offshore facilities.

2.3.2. Caspian Sea: Kazakhstan and Azerbaijan

Kashagan Project, North Caspian Operating Company. At the late production stage, when the gas flow rate per well cluster declines substantially below the design value, centrifugal compressor stations experience operating problems, including a limited control range, increased specific energy consumption, and frequent shutdowns.

Reciprocating gas-engine-driven compressors are a technically appropriate option for well clusters operating at such reduced rates. This is stated here as an engineering assessment; the authors are not aware of a publicly documented installation of this type at the field.

Azeri–Chirag–Gunashli Project, BP and SOCAR. Compressor stations installed on the offshore platforms use centrifugal compressor units sized for the plateau production rate. During the late production stage, the efficiency of these units decreases and fuel

consumption increases. Compression of the tail-end gas stream by reciprocating modules is a technically appropriate option for facilities of this type, and is stated here as an engineering assessment rather than as a reported operator decision.

In summary, the Caspian region demonstrates the typical problem associated with the narrow operating range of centrifugal compressors. To ensure stable operation under declining production conditions, reciprocating compressors, including AJAX units, are considered an effective alternative.

2.3.3. Gulf of Mexico: United States and Mexico

Thunder Horse and Atlantis Projects. These facilities were initially designed with high-capacity centrifugal compressors sized for the plateau production rate.

During the late production stage, surge-related problems and declining compressor efficiency were observed. Additional compression stages and adjustable bypass systems were introduced. In some projects, auxiliary reciprocating compressors were used during the tail-end production phase.

On Gulf of Mexico platforms, the principal focus is placed on modularity and the adaptation of compressor-station configurations. Reciprocating booster compressors are used to extend the productive life of wells and reduce energy consumption.

The three regional examples given above are presented as context describing current industrial practice at the late production stage. They illustrate the operating difficulties encountered by centrifugal compressor stations under declining production, and the direction in which operators have moved, but they do not by themselves validate the specific six-cylinder configuration proposed in Section 3.4. That configuration is substantiated in the present work by calculation only; its experimental verification is a subject for further work.

The principal trends and conclusions are summarised in Table 1.

Table 1. General trends and conclusions from international compressor applications.

Region	Compressor Type	Challenges Under Declining Production	Solutions
North Sea	Centrifugal compressors	Limited operating range and increased energy consumption	Upgrading and installation of modular booster compressors
Caspian Sea	Centrifugal compressors	Efficiency loss and frequent shutdowns	Transition to reciprocating gas-engine-driven compressors
Gulf of Mexico	Centrifugal compressors	Surge and increased energy intensity	Auxiliary reciprocating compressors

Operational experience with compressor stations at the late stage of field development shows that centrifugal compressors provide high efficiency at or near their rated flow rates but become inefficient once the throughput falls substantially below the design value and anti-surge recycling has to be used [11]. Under late-life production conditions, frequent flow recycling through bypass lines, increased specific energy consumption, a reduction in overall system efficiency and an increased frequency of shutdowns and maintenance interventions are commonly observed.

Reciprocating gas-engine-driven compressors, such as AJAX, Dresser-Rand, and Ariel units [3,4], offer several advantages under these conditions. They can operate stably over a gas flow-rate range of 0.5–5.0 million m³/day, provide a wider control range in terms of suction pressure and flow rate, and offer high maintainability and modularity.

2.4. Depletion Scenario Adopted in the Study

The depletion scenario adopted as the basis for the model developed in the following section is shown in Figure 1. Over a ten-year period, the gas flow rate of the well cluster

decreases tenfold, from 5.0 to 0.5 million m³/day, while the wellhead pressure, which is also the compressor suction pressure, decreases fiftyfold, from 5.0 to 0.1 MPa. The cumulative recovery over the same period amounts to about 8.2×10^9 m³, so that the recoverable reserves of the cluster remaining to be produced fall from about 8.2×10^9 m³ at the beginning of the period to about 0.18×10^9 m³ in the final year. The shaded bands in Figure 1b indicate the compressor operating mode required in each interval, as derived in Section 3.4.

The average reservoir pressure shown in Figure 1b is included in order to place the scenario in its reservoir context. It is obtained from a lumped deliverability relation of the usual quadratic form, $Q = C(p_r^2 - P_1^2)$, with $C = 0.455$ million m³/(day·MPa²) and an initial average reservoir pressure of 6.0 MPa, which together correspond to a drawdown of 1.0 MPa at the initial production rate. The curve is illustrative: it does not enter the compressor calculation of Section 3, in which only the suction pressure P_1 is used.

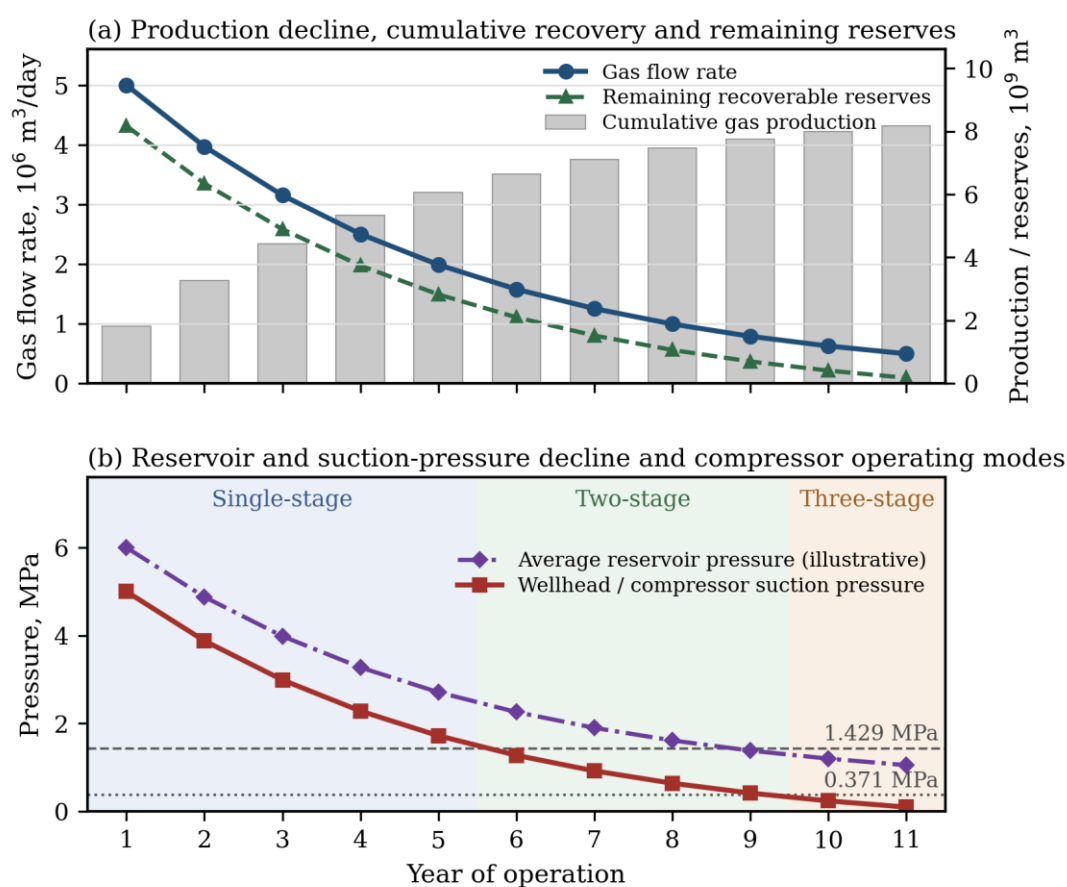


Figure 1. Depletion scenario of the gas-condensate field adopted in this study: (a) gas flow rate, cumulative gas production and remaining recoverable reserves; (b) average reservoir pressure, wellhead/compressor suction pressure and the corresponding compressor operating modes. Plotted from the input data of Table 2 and the results of Table 3; the reservoir-pressure curve is illustrative and is obtained as described in the text.

Table 2. Input data adopted for the calculation and their status.

Parameter	Symbol	Value	Status
Initial gas flow rate of the well cluster	Q_1	5.000 million m ³ /day	Assumed (scenario)
Final gas flow rate (year 11)	Q_{11}	0.500 million m ³ /day	Assumed (scenario)
Annual decline factor	λ	0.7943	Derived from Q_1 and Q_{11}
Initial wellhead/suction pressure	$P_{1,max}$	5.0 MPa	Assumed (scenario)
Final wellhead/suction pressure	$P_{1,min}$	0.1 MPa	Assumed (scenario)

Required discharge pressure	P_2	5.5 MPa	Assumed (scenario)
Maximum pressure ratio per stage	$\Pi_{\max}$	3.85	Selected (temperature-limited)
Cylinder bores, two of each size	D	381; 228.6; 114.3 mm	Selected (compressor unit)
Piston stroke	S	0.1397 m	Selected (compressor unit)
Rotational speed	N	700 min^{-1}	Selected (compressor unit)
Relative clearance volume	c	0.14	Selected (compressor unit)
Polytropic exponent	n	1.28	Assumed (gas)
Effective volumetric efficiency	η_v	0.739	Calculated from c , n and $\Pi_{\max}$
Suction temperature	T_1	293.15 K	Assumed (gas)
Compressibility factor at suction	Z_1	0.861	Assumed (gas)
Standard reference conditions	$P_{\text{st}}; T_{\text{st}}$	0.101325 MPa; 288.15 K	ISO 13443 [20]

Table 3. Calculation Results for Natural Gas Compression Parameters. Columns 2 and 4 define the assumed depletion scenario and follow from Equations (14) and (15); columns 3, 5, 6 and 7 are calculated from Equations (16)–(18). The single-stage configuration applies in years 1–5, the two-stage configuration in years 6–9 and the three-stage configuration in years 10–11.

Year	Gas Flow Rate, million m^3/day	Gas Production, million m^3/year	Wellhead/Compressor Suction Pressure, MPa	Capacity of One Compressor, million m^3/day	Required Number of Compressors	Total Capacity of All Compressors, million m^3/day
1	5.000	1825.00	5.000	1.938	2.58 (3)	5.814
2	3.972	1449.65	3.880	1.504	2.64 (3)	4.512
3	3.155	1151.50	2.991	1.159	2.72 (3)	3.477
4	2.506	914.67	2.284	0.885	2.83 (3)	2.655
5	1.991	726.55	1.723	0.668	2.98 (3)	2.004
6	1.581	577.12	1.277	0.466	3.39 (4)	1.864
7	1.256	458.42	0.923	0.336	3.73 (4)	1.344
8	0.998	364.14	0.642	0.234	4.26 (5)	1.170
9	0.792	289.24	0.418	0.152	5.21 (6)	0.912
10	0.630	229.75	0.241	0.065	9.68 (10)	0.650
11	0.500	182.50	0.100	0.027	18.52 (19)	0.513

The following section presents a complete conceptual model of a compressor station with adaptive, or dynamic, control developed for the conditions considered in this study.

3. Materials and Methods

3.1. Problem Statement and Input Conditions

High-rate condition: flow rate $Q_{\text{high}} = 5.0$ million m^3/day ($\sim 5 \times 10^6 \text{ m}^3/\text{day}$).

Low-rate condition (late stage): $Q_{\text{low}} = 0.5$ million m^3/day .

Target suction pressure: at $Q = Q_{\text{high}}$, $P_{1,\text{high}} = 5.0$ MPa; at $Q = Q_{\text{low}}$, the pressure is gradually reduced to $P_{1,\text{min}} = 0.1$ MPa.

Required minimum discharge pressure: $P_2 \geq 5.5$ MPa.

Equipment: reciprocating gas-engine-driven compressors in a modular configuration, with several identical modules arranged in parallel and/or in series.

3.2. General Concept of the Model

1. The control unit continuously measures the current total flow rate $Q(t)$ and calculates the target suction pressure:

$$P_{1,\text{set}}(t) = f(Q(t)). \quad (1)$$

2. The compressor or compressors operate to maintain:

$$P_1 \approx P_{1,\text{set}} \quad (2)$$

subject to the following constraints:

$$P_{1,min} \leq P_{1,set} \leq P_{1,high} \quad (3)$$

and

$$P_2 \geq 5.5 \text{ MPa} \quad (4)$$

3. Control is achieved through a combination of the following methods: starting and stopping compressor modules or cylinder blocks, controlling the driver speed or operating mode where technically possible, changing the number of active cylinders or valves, and using intercoolers and bypass lines.
4. The main objective of the control system is to minimise energy consumption over the operating period and/or minimise total equipment wear while maintaining the specified technological constraints.

3.3. Basic Mathematical Relationships

3.3.1. Density and Mass Flow Rate

The mass flow rate $\dot{m}$ is determined from the total volumetric flow rate q_v :

$$\dot{m} = \rho_1 q_v, \quad (5)$$

where

$$\rho_1 = \frac{P_1 M}{Z(P_1, T_1) R_u T_1} \quad (6)$$

is the gas density at the compressor suction for a real gas; R_u is the universal gas constant, M is the molar mass, and Z is the compressibility factor.

The suction volumetric flow rate q_v , expressed in m³/s, is calculated from the total flow rate Q , expressed in m³/day:

$$q_v = \frac{Q}{86400} \quad (7)$$

3.3.2. Specific Work of Polytropic Compression

For a polytropic process with exponent n :

$$w = \frac{n}{n-1} R_s T_1 \left[\left(\frac{P_2}{P_1} \right)^{\frac{n-1}{n}} - 1 \right], \quad (8)$$

where

$$R_s = \frac{R_u}{M} \quad (9)$$

is the specific gas constant, J/(kg·K), and w is the specific compression work, J/kg. For a real gas, the compressibility factor Z and appropriate corrections to n should be taken into account.

The useful compression power is:

$$P_{useful} = \dot{m} \cdot w \quad (10)$$

The required electrical power or equivalent fuel input is:

$$P_{el} = \frac{P_{useful}}{\eta_{comp} \eta_{mech}} \quad (11)$$

where η_{comp} is the compressor efficiency, representing the conversion from indicated to useful power, and η_{mech} is the mechanical efficiency of the driver.

3.3.3. Multistage Compression

The total pressure ratio is:

$$\Pi = \frac{P_2}{P_1} \quad (12)$$

For k compression stages with ideal intercooling, the optimum pressure ratio per stage is $\Pi^{1/k}$.

A practical estimate of the required number of stages is:

$$k \approx \frac{\ln \Pi}{\ln r_{opt}} \quad (13)$$

where r_{opt} is the optimum pressure ratio per stage. For reciprocating compressor units with intercooling, it is generally $r_{opt} \sim 2 \div 4$. At $P_1 = 0.1$ MPa and $P_2 = 5.5$ MPa, the total pressure ratio is $\Pi = 55$. For $r_{opt} = 3.85$, three compression stages are required according to the logarithmic estimate, which is exactly the number of stages provided by the configuration described below.

3.4. Compressor Configuration and Operating Modes

The compressor model was selected based on the following operating conditions: during the late stage of field development, the production rate of gas-condensate wells decreases from 5.0 million m³/day to 0.5 million m³/day over a period of 10 years. The wellhead pressure and the compressor suction pressure decrease from 5.0 MPa to 0.1 MPa. The required discharge pressure is 5.5 MPa.

The compressor design is intended to operate in three different modes and configurations. Therefore, an Ariel-type horizontally opposed compressor with six cylinders of different diameters was selected:

- Two first-stage cylinders with a diameter of 381 mm (15.0 in);
- Two second-stage cylinders with a diameter of 228.6 mm (9.0 in);
- Two third-stage cylinders with a diameter of 114.3 mm (4.5 in).

The main feature of this configuration is that different combinations of cylinders can be brought into operation, allowing the system to operate over a wide range of flow rates and suction pressures. The cylinder arrangement for each operating mode, the reasons for the selected configuration, and its effectiveness are described below.

3.4.1. Single-Stage Operation at High Suction Pressure

Configuration: All six cylinders operate in parallel (2L + 2M + 2S). All cylinders have suction connections opening into the same suction manifold and discharge connections opening into the same discharge manifold.

Effect: This configuration provides the maximum total working-cylinder displacement.

Operating limitation: This mode is applicable only when the suction pressure is approximately:

$$P_{wc} \geq 1.429 \text{ MPa}$$

This limitation is based on a maximum allowable pressure ratio per stage of 3.85. This value is not arbitrary: for the gas and the suction temperature considered here it corresponds to a discharge temperature of approximately 150 °C, which is the practical limit recommended for reciprocating compressor cylinders operating under API 618 conditions [4]. The temperature at the outlet of each stage is therefore the physical criterion that fixes the switching points between the operating modes.

3.4.2. Two-Stage Operation in the Balanced Configuration

Configuration: Two 381 mm cylinders and two 228.6 mm cylinders operate as the first stage, while two 114.3 mm cylinders operate as the second stage. For this purpose, the cylinders are separated by valves installed in the manifold system.

Operating range: This configuration is suitable for suction pressures ranging from 1.429 MPa to 0.371 MPa.

3.4.3. Three-Stage Operation at Low Suction Pressure

Configuration: Two 381 mm cylinders operate as the first stage, two 228.6 mm cylinders operate as the second stage, and two 114.3 mm cylinders operate as the third stage.

Capacity: The gas flow rate through all three stages is the lowest among the three configurations; however, this arrangement provides the high overall compression ratio required under low-suction-pressure conditions.

Operating range: Three-stage operation is required when the suction pressure is: $P_{wc} < 0.371$ MPa.

With a discharge pressure of 5.5 MPa and a maximum pressure ratio of 3.85 per stage, the lowest suction pressure that the three-stage configuration can handle is $5.5/3.85^3 = 0.096$ MPa. The three-stage arrangement therefore covers the whole suction-pressure range considered in this study, down to 0.1 MPa, and no fourth compression stage is required. The estimate of the number of stages obtained from Equation (13) and the number of stages provided by the proposed configuration are thus in agreement.

With a properly designed valve and piping arrangement, the compressor can operate in any of the required single-stage, two-stage, or three-stage modes without physically replacing the cylinders.

The principal engineering requirement is to maintain an appropriate volumetric balance between the compression stages in each operating mode. The cylinder groupings described above are intended to optimise this balance.

3.5. Input Data and Calculation Procedure

The input data used in the calculation are collected in Table 2. They are of three kinds, identified in the last column of the table: values assumed for the depletion scenario, values selected for the compressor unit, and values adopted for the gas and for the reference conditions. Every quantity reported in Table 3 is calculated from these inputs by Equations (14)–(18); no quantity in Table 3 is an independent measurement.

The gas flow rate is assumed to decline by the same percentage each year, so that

$$Q_i = Q_1 \cdot \lambda^{(i-1)}, \quad \lambda = (Q_{11}/Q_1)^{1/10} = 0.7943 \quad (14)$$

The wellhead pressure, which is also the compressor suction pressure, is taken to vary linearly with the current gas flow rate between the two end points of the scenario:

$$P_1 = a \cdot Q + b, \quad a = 1.0889 \text{ MPa per million m}^3/\text{day}, \quad b = -0.4444 \text{ MPa} \quad (15)$$

The effective volumetric efficiency of the cylinders is obtained from the relative clearance volume and the maximum pressure ratio per stage:

$$\eta_v = 1 - c \cdot [(\Pi_{\max})^{1/n} - 1] = 0.739 \quad (16)$$

The capacity of one compressor unit, expressed in standard cubic metres per day under the reference conditions specified in ISO 13443 [20], follows from the swept volume of the cylinders admitted to the first compression stage and from the gas density at the suction:

$$q = A_1 \cdot S \cdot N \cdot 1440 \cdot \eta_v \cdot (P_1/P_{st}) \cdot (T_{st}/T_1)/Z_1 \quad (17)$$

where $A_1 = (\pi/4) \cdot \Sigma D_i^2$ is the total piston area of the cylinders forming the first stage. This area is the only quantity that changes between the three operating modes: it comprises all six cylinders in single-stage operation, the two 381 mm and the two 228.6 mm cylinders in two-stage operation, and the two 381 mm cylinders in three-stage operation, which gives the relative values 1.000, 0.938 and 0.690. The required number of units and the total capacity of the station are then

$$N_c = [Q/q], \quad Q_{\text{tot}} = N_c \cdot q \quad (18)$$

The procedure is illustrated for year 9. Equation (14) gives $Q = 5.000 \times 0.7943^8 = 0.792$ million m^3/day , and Equation (15) gives $P_1 = 1.0889 \times 0.792 - 0.4444 = 0.418$ MPa. This value lies below $P_2/\Pi_{\text{max}} = 1.429$ MPa and above $P_2/\Pi_{\text{max}}^2 = 0.371$ MPa, so the unit operates in the two-stage configuration and $A_1 = (\pi/4) \times (2 \times 0.381^2 + 2 \times 0.2286^2) = 0.3100$ m^2 . Equation (17) then gives $q = 0.3100 \times 0.1397 \times 700 \times 1440 \times 0.739 \times (0.418/0.101325) \times (288.15/293.15)/0.861 = 0.152$ million m^3/day , and Equation (18) gives $N_c = \lceil 0.792/0.152 \rceil = \lceil 5.21 \rceil = 6$ units with a total capacity of $6 \times 0.152 = 0.912$ million m^3/day . These four values form the ninth row of Table 3. Recalculation of the remaining rows from Equations (14)–(18) reproduces the tabulated values to within 0.5%, the residual difference arising from the rounding of intermediate results.

4. Results

Table 3 below presents the calculated values of the following parameters:

- Daily and annual gas production rates by year, million m^3/day and million m^3/year ;
- Wellhead pressure, corresponding to the compressor suction pressure, by year, MPa;
- Capacity of one compressor under different operating configurations by year, million m^3/day ;
- Required number of compressors by year;
- Total capacity of all operating compressors by year, million m^3/day .

Changes in the natural gas compression parameters of the gas-condensate field over the ten-year period are presented in Figure 2.

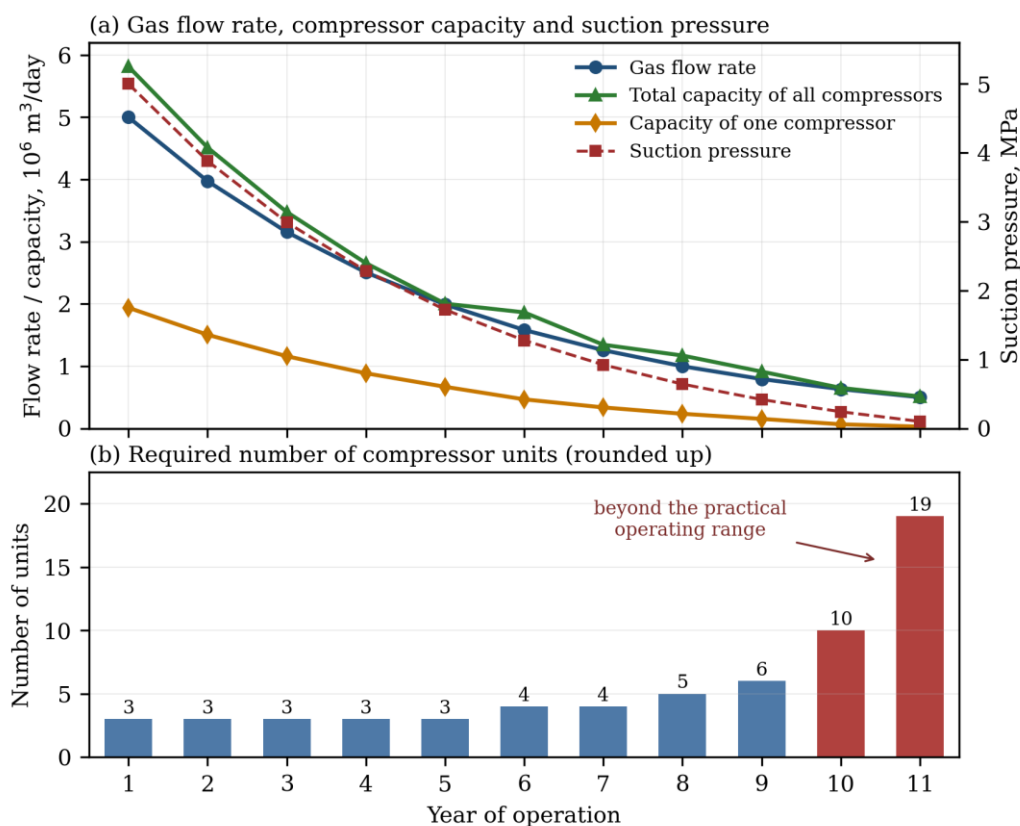


Figure 2. Changes in the natural gas compression parameters of the gas-condensate field over the ten-year period: (a) gas flow rate, capacity of a single compressor unit, total capacity of the operating units and compressor suction pressure; (b) required number of compressor units. Years 10 and 11 are shown in a different colour because they lie outside the practical operating range of the proposed configuration.

5. Discussion

5.1. Interpretation of the Calculation Results

Based on the analysis of the calculation results presented in Table 3 and the graphs shown in Figure 2, the following conclusions can be drawn:

1. The proposed compressor-unit reconfiguration scheme enables the compression of natural gas produced from gas-condensate wells during the late stage of field development for at least nine years using a single unit size, without replacing the cylinders and without changing the type of compressor. Over this period, the number of units operating in parallel increases from three to six, which corresponds to the installation of additional identical modules rather than to the replacement of the compression facility.
2. Within the nine-year period, the capacity of a single unit falls from 1.938 to 0.152 million m³/day, that is, to about 8% of its value in the first year, and the configuration remains operable at that level; the lower operating limit of the configuration is therefore of the order of 10% of the rated capacity of one unit. This figure is obtained by extrapolation of the model and should be regarded as an estimated lower operating limit rather than as a validated one; its confirmation requires cylinder-level performance data or a manufacturer performance map.
3. After the ninth year of operation, the required number of units rises steeply, to ten in year 10 and to nineteen in year 11 (Figure 2b). These two years therefore lie outside the practical operating range of the configuration considered, and continued production beyond year 9 requires the installation of booster compressors upstream of the station rather than a further increase in the number of parallel units.

At the same time, several practical considerations must be taken into account:

- Valve losses and pulsations. Pressure losses and flow pulsations increase when multiple cross-connection valves are used. Pulsation dampers and silencers should therefore be incorporated into the system.
- Increased compression work. Multistage operating modes may require interstage cooling. Interstage temperatures must be continuously monitored and maintained within the allowable limits.
- Uniform load distribution. The load should be distributed evenly among all compressor cylinders. Uneven loading may cause dynamic imbalance, accelerated wear, and reduced equipment life. Balanced flow distribution is therefore essential when designing the manifold system.
- Safety standards. The design must comply with the applicable safety standards and requirements governing valve selection, pressure-relief devices, instrumentation, and control systems.
- Maximum allowable working pressure. All compressor cylinders and associated components must be designed for the maximum operating pressure specified for the system, which is 5.5 MPa in the present study.

5.2. Comparison with Previous Studies

The result obtained here may be compared with published work on part-load compression offshore. Cruz et al. [11] showed, for an offshore gas-processing train under a decreasing gas load, that replacing a single oversized centrifugal machine by several smaller units operating in parallel raises the exergy efficiency at 25% gas load from about 49% to above 80% and lowers the carbon dioxide emission intensity by about 34%. The configuration proposed here follows the same principle of matching installed capacity to the current load, but achieves it in two steps rather than one: the swept volume admitted to the first stage of a single unit is first reduced by transferring cylinders to the higher

stages, and only afterwards is the number of parallel units increased. Compared with an arrangement of several fixed-capacity parallel machines, this additional degree of freedom keeps the number of installed units at three throughout the first five years of the declining-production period, as shown in Figure 2b.

Reconfiguring a reciprocating compressor as the suction pressure falls is not in itself new. Cylinder unloading, clearance pockets, restaging and stepless suction-valve control are established means of capacity regulation, and their thermodynamic behaviour has been examined in detail [1,2]. What is proposed here is therefore not a new regulating device, but a rule for selecting the cylinder set of a single frame so that the same machine can be operated as a one-, two- or three-stage unit over a fiftyfold decrease in suction pressure, together with the calculation procedure that converts a given production-decline profile into the number of units required in each year of operation.

5.3. Limitations of the Model

The limitations of the model should be stated explicitly. The relationship between the wellhead pressure and the current flow rate, Equation (15), is postulated as a linear interpolation between the two end points of the scenario and is not derived from an inflow–performance relationship or from a material balance for a specific reservoir. The effective volumetric efficiency is treated as constant across the three operating modes, whereas in a real machine it varies with the actual pressure ratio of each stage; the compressibility factor and the suction temperature are likewise held constant. Pulsation, valve losses and the dynamic behaviour of the manifold system are discussed qualitatively above but are not calculated. The lower operating limit of about 10% of rated capacity follows from the model and has not been verified against cylinder-level performance data. Finally, no experimental verification of the proposed configuration has been carried out, and the regional examples of Section 2 provide context rather than validation. The average reservoir pressure shown in Figure 1b is likewise an illustrative construction obtained from a lumped deliverability relation; it is not derived from a reservoir simulation and does not enter the compressor calculation.

5.4. Range of Applicability

The applicability of the model is bounded accordingly. It is intended for a compressor suction pressure between 0.1 and 5.0 MPa, a discharge pressure not exceeding 5.5 MPa, a gas flow rate between 0.5 and 5.0 million m³/day, and a gas stream that has passed through separation, so that liquid carry-over remains within the limits admissible for reciprocating cylinders. The scenario examined here, in which the flow rate falls tenfold and the suction pressure fiftyfold over ten years, was chosen deliberately as a severe case; in real gas-condensate fields these parameters, together with water and sand production, change more gradually. The unit used for the demonstration was also chosen deliberately small so that the dynamics would be visible over the whole period. For a specific field the designer would select a unit of higher rated capacity—for example, three times the capacity used here—in which case the number of units required over the same period would range from one to seven rather than from three to nineteen.

6. Conclusions

A model and a configuration of a horizontally opposed reciprocating compressor have been developed in order to adapt the compression process to declining gas production. The proposed model allows the same unit size to remain in service throughout the declining-production period, so that adaptation to the falling suction pressure is achieved by reconfiguring the existing cylinders and by adding identical modules rather than by replacing the compression facility. A quantitative assessment of the associated energy and

cost effects requires machine-specific and field-specific data and was not undertaken in this study.

Unlike a single oversized centrifugal machine, whose efficiency degrades sharply once anti-surge recycling becomes necessary at part load [11], the proposed configuration is estimated to remain operable down to approximately 10% of the rated capacity of one unit. This value is an extrapolation of the model and requires experimental or manufacturer confirmation. Subject to that qualification, the configuration has the potential to be implemented at existing offshore production facilities, and its verification on an operating unit is the natural continuation of the present work.

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