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Synthesis of Mechanical Driver and Power Generation Configurations, Part 2: LNG Applications

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Optimization framework for the synthesis of power systems has been presented in Part 1 of this article, which systematically identifies the most cost-efficient number, type, and model of mechanical drivers, together with optimal arrangement for compressor stage, helper motors or generators, and power plants. The developed methodology is applied to an LNG case study in which optimal and near-optimal systems at various economic scenarios are identified. Also, a systematic methodology for the integrated design of refrigeration and power systems has been addressed to improve the overall design of low temperature processes. Additional key degrees of freedom such as stage pressure ratios and plant capacity are optimized, alongside other design variables, which provide greater flexibility in the matching of power supply and demands. This strategy is applied to an LNG case study and shows the convenience of this approach as the interactions between the refrigeration and power systems are systematically exploited. © 2010 American Institute of Chemical Engineers *AIChE J,* 56: 2377–2389, 2010

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Introduction

Provision of low temperature or cold energy to the processes requires refrigeration cycles to remove heat from hot streams or liquefy gas stream(s). Refrigerant compressor requires mechanical power to increase the pressure and condensing temperature of the refrigerant vapors. This mechanical energy is provided by steam turbines, gas turbines, or electric motors, which act as compressor drivers. These drivers, in turn, require thermal energy either directly or indirectly (e.g., electricity) to perform, which in some cases can be covered with heat recovered from hot process streams to some extent, although in general the heat of combustion from hydrocarbon fuels is used for this purpose.

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Compressors and drivers are normally the most expensive items in a refrigeration system. Their size, determined by the power requirement, has a considerable impact on capital investment. Likewise, fuel consumption, a main operating cost (especially nowadays with high energy prices), also increases with power demand. Therefore, there is a strong incentive in designing thermally efficient, yet cost-effective, refrigeration systems.

Although several approaches have been published to tackle the optimal design of refrigeration systems,[1–3] the cost and performance impact of driver selection has not been given proper consideration. Restricting the problem only to minimize the power consumption may in principle aim towards minimum total costs. However, this optimality may well be lost in a later stage if the resulting compressor sizes and electricity demand lead to a poor driver and power plant selection under the given economic scenario.

On the other hand, the problem of optimal driver selection has been in some cases addressed in the literature by procedures that tackle this in conjunction with electricity generation and process steam demand and supply as "utility systems."[4–9] However, practical considerations such as the existence of gas turbines only as discrete options, are often overlooked. Also, these methods do not address issues such as the need for gas turbine startup devices, the existence of a nonsteam solution space, and the flexibility of splitting and/or combining mechanical demands to exploit better machinery configurations. Furthermore, the power demands are fixed inputs for these approaches, which rules out the possibility of any systematic feedback to the refrigeration design stage and, hence, of adjusting the decisions there in search for a better driver/power plant selection (power system).

It is evident that there are important interactions between refrigeration and power systems, and it is the goal of this article to produce a method capable of systematically exploiting these opportunities in an integrated design context. Special attention is given throughout this study to liquefied natural gas (LNG) applications and to refrigeration systems that use component mixtures (mixed refrigerants) as working fluids.

First, the optimization framework for driver selections in power-dominated energy systems presented in Part 1 of this article[10] is now applied for an industrial LNG case study, with which the applicability and practicality of the proposed design method are demonstrated. Then, a novel approach for the simultaneous optimization of both power systems and refrigeration systems is explained, in which the interactions between both systems are fully exploited in a most integrated and systematic manner. The methodology is also illustrated with an LNG case study, in which the benefits of integrated design approach are clearly justified.

Case Study 1: Design of Power Systems

The described formulation from Part 1 is now applied to a case study. The best resulting designs are compared with each other and the sensitivity of the optimal solution to the fuel cost is explored, as well as the effect of considering sparing philosophy for electricity generation.

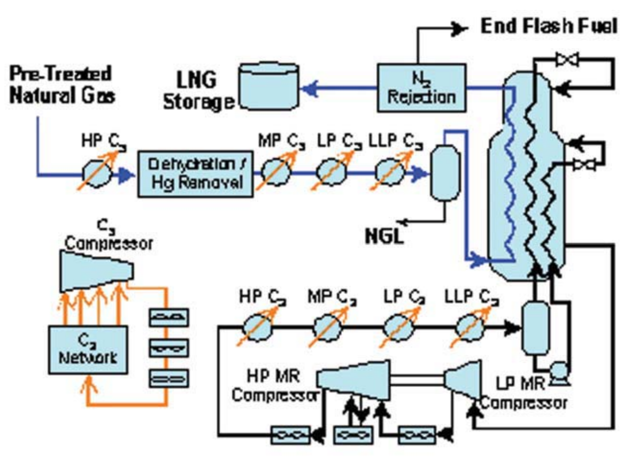


Figure 1. Simplified propane pre-cooled mixed refrigerant process.

[Color figure can be viewed in the online issue, which is available at wileyonlinelibrary.com.]

Table 1. Mechanical Power Demands of the Refrigerant Compressors

Compressor/Stage	Power Demand (MW)
C3/1	2.50
C3/2	6.96
C3/3	12.35
C3/4	32.25
Total C3	54.06
MR/1	57.64
MR/2	18.74
MR/3	27.34
Total MR	103.72

The optimization model explained in Part 1 was implemented using GAMS IDE 2.0.13.0 by GAMS Development Corporation.[11] All the process data and input parameters were provided in an interface spreadsheet and transferred automatically to GAMS using a Visual Basic macro to centralize the information and to make easier the management of different scenarios.

Process data

This case study is based on the Propane Pre-cooled Mixed Refrigerant LNG process by Air Products and Chemicals. The process features a four level propane pre-cooling cycle, a mixed refrigerant cycle, and an end flash system for nitrogen rejection. The rejected stream will provide most of the fuel required.

A basic flowsheet and the set of process data for this case study were derived from a publication,[12] and natural gas feed is available at 60 bara and 25°C.* A simplified flowsheet is shown in Figure 1. The mixed refrigerant compressor has three stages, the first one being axial and the rest centrifugal. The propane compressor has four centrifugal stages with side loads.

To work from a consistent and relevant set of power demand data, the process as published by Vink and Nagelvoort[12] was simulated. The basis was a single liquefaction train producing 5 mtpa (millions of metric tons per year) of LNG, (please note that 2–6 mtpa is typical production capacity for Propane Pre-cooled Mixed Refrigerant LNG process by Air Products and Chemicals). The resulting refrigerant compressor demands are given in Table 1.

The basic electricity demand is estimated as 42.7 MW, which does not include any allowance for potential electric motors or steam support equipment within the power system. The process heating demands are estimated to be 34.2 t/h of medium pressure steam and 26.3 t/h of low pressure steam. That is, 36.31 MW of heat in total.

Other parameters of the base case scenario and candidate direct drive gas turbines (DDGTs) and power plants are summarized in Tables 2–5. Candidate DDGTs and power plants were de-rated to account for ambient temperature, ageing, fouling, and fuel composition. The fuel gas is initially assumed as a blend of 80% end flash gas and 20% pretreated natural gas. This has to be checked in the solution for consistency. If significantly different, an updated fuel derating factor must be applied.

*Feed composition: N_2 (1.5%); CO_2 (2.2%); C1 (85.1%); C2 (6.5%); C3 (3.0%); C4 (1.2%); C5+ (0.5%).

Table 2. Simple Cycle Power Plant Options for Case Study*

Model	Type	Output† (kW)	Heat Rate† (kJ/kWh)	Cost‡ (k$, FOB)
LM1600PA	Aeroderivative	13,750	10,153	8077
LM2000	Aeroderivative	18,000	9892	8027
LM2500PE	Aeroderivative	22,800	9783	11,106
LM6000PC	Aeroderivative	42,665	8779	14,236
LM6000PD	Aeroderivative	42,227	8698	15,145
PG5371PA	Industrial	26,300	12,649	7754
PG6581B	Industrial	42,100	11,223	14,640
PG6101FA	Industrial	70,140	10,529	22,213
PG7121EA	Industrial	85,400	10,993	21,203
PG7241FA	Industrial	171,700	9873	41,397
PG9171E	Industrial	123,400	10,656	25,848
PG9231EC	Industrial	169,200	10,305	34,884
PG9351FA	Industrial	255,600	9757	49,979
W251B11/12	Industrial	49,500	11,024	14,034
V64.3A	Industrial	67,000	10,371	20,597
W501D5A	Industrial	120,500	10,381	25,747
V94.2	Industrial	157,000	10,466	30,179
W501F	Industrial	186,500	9632	40,387
V94.2A	Industrial	190,000	10,227	36,449
W501G	Industrial	253,000	9241	48,262
V94.3A	Industrial	265,000	9347	51,897
Trent	Aeroderivative	51,190	8662	15,650

*Data from gas turbine world handbook, 2000–2001.[13]
†At ISO conditions.
‡Updated to 2002, using chemical eng plant cost index.

The plant is not allowed to trade electricity and consequently its import is set to zero. There is no benefit from exporting electricity but the electricity export has not been banned to let the overall electricity balance close. The lifetime of the plant for economic analysis purposes is 25 years. Three hundred fifty operation days per year are assumed initially. Regarding the cost of compressors, a 20% penalty is assumed per extra casing.

Base case

The base case corresponds to minimize the total cost of the system considering a fresh fuel cost of 1 \$/MMBtu ($\cong$ 3.41 \$/MWh). Only gas turbines and electric motors are considered as drivers in the first part of this case study. In this case, heat is supplied to the process through an implicit hot oil circuit. Also, availability is not participating in the objective function for the moment, although the N + 1 sparing philosophy is set to active for electricity generating units.

The system in Figure 2a is the resulting optimal solution to the base case scenario (system BC-1 from now on). Three aeroderivative and one Frame 7 gas turbines drive the compressor stages whereas a Frame 5 gas turbine in simple cycle is selected as the only power plant (plus one spare). The total cost of this system is 245.87 MM\$ (millions of US\$), out of which 219.51 MM\$ are for capital expenditure and the remaining corresponds to the present value of the fuel costs along the plant lifetime. Considering the cost of fresh fuel as 1 \$/MMBtu has encouraged the solver to select three

Table 3. Combined Cycle Power Plant Options for Case Study*

Model	Total Output† (kW)	GT Power† (kW)	Heat Rate† (kJ/kWh)	Cost‡ (k$, FOB)
S106B	64,300	41,600	7340	20,900
S106FA	107,100	68,900	6794	39,478
S107EA	130,200	83,500	7174	35,339
S206B	130,700	83,200	7227	37,863
S109E	189,200	121,600	6931	45,435
S206FA	217,000	137,800	6704	53,159
S107FA	262,600	170,850	6425	58,511
S207EA	263,600	167,000	7068	58,435
GUD 1S.64.3A	100,000	65,000	6868	37,206
GUD 2.64.3A	202,000	130,000	6815	49,459
GUD 1.V94.2	232,500	152,000	6995	53,715
GUD 1S.84.3A	260,000	176,000	6309	57,501
GUD 1S.94.2A	293,500	198,500	6520	58,526
1.W251B11/12	71,500	48,000	7533	24,838
2.W251B11/12	143,500	96,000	7501	44,022
1.W501F	273,500	182,500	6489	57,536

*Data from gas turbine world handbook, 2000–2001.[13]
†At ISO conditions.
‡Updated to 2002, using the chemical engineering plant cost index. Cost reported in the source is for turnkey plants. It was assumed that turnkey price is 2.0 times the FOB cost.

Table 4. Gas Turbine Driver Options for Case Study*

Model	Type	Output† (kW)	Heat Rate† (kJ/kWh)	Cost‡ (k$, FOB)
LM1600PA	Aeroderivative	14,320	9750	6553
LM2500PE	Aeroderivative	23,270	9587	7875
LM6000	Aeroderivative	44,740	8460	12,621
M3142J	Industrial	11,290	13,440	3377
MS5002D	Industrial	32,590	11,898	7976
M5261RA	Industrial	19,690	13,270	5250
M5382C	Industrial	28,340	12,310	6967
M6511B	Industrial	37,810	11,120	10,299
M6581B	Industrial	38,290	11,060	10,400
M7111EA	Industrial	81,560	11,021	17,467
M7121EA	Industrial	86,230	10,922	18,578
COBERRA 6562	Aeroderivative	25,930	9472	7371
COBERRA 6761	Aeroderivative	32,590	8961	8380
Trent	Aeroderivative	52,549	13,024	8349

*Data from gas turbine world handbook, 2000–2001.[13]
†At ISO conditions.
‡The costs have been updated to January 2002, using the chemical engineering plant cost index and include speed adjusting gearbox when required.

aeroderivative gas turbines to improve the thermal efficiency of the system.

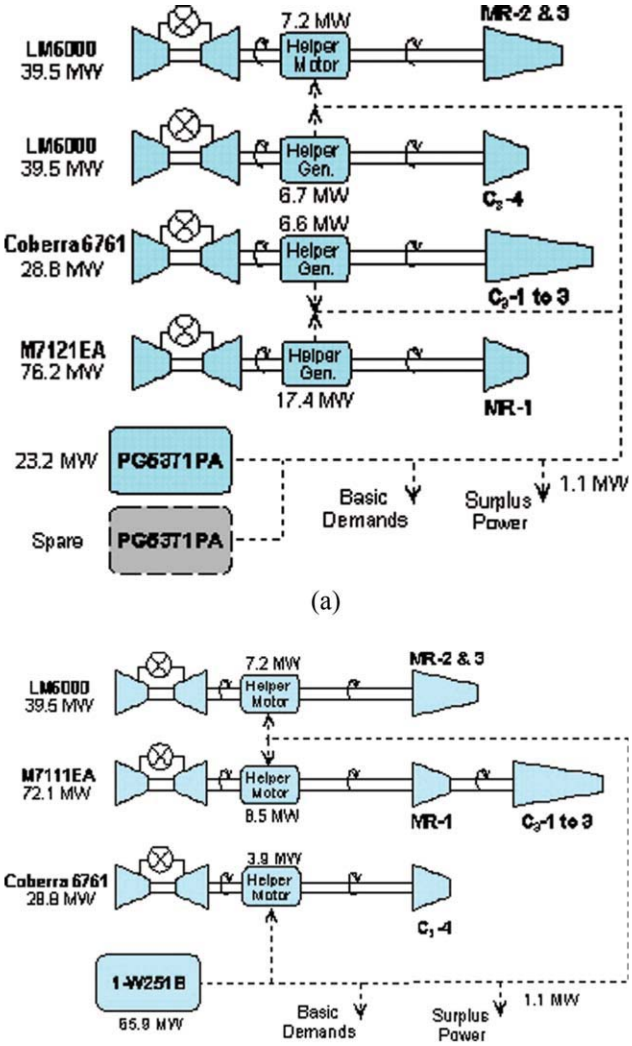
The MILP optimization to obtain system BC-1 was carried out using the Cplex 7.0 solver in GAMS with an optimality criterion of 0.01% (relative difference between the best integer solution found and the best relaxed solution). By considering a superstructure size as four potential gas turbines drivers, four potential electric motors, and four potential packaged power plants, the GAMS model generated 1096 equations and 1093 variables, out of which 464 were binary. The runtime was 508 CPU seconds using a Pentium® IV processor (3.0 GHz) with 512 Mb of RAM.

If sparing philosophy was not taken into account, the system in Figure 2b (system BC-2) would have been the optimal solution instead. A Frame 7 and two aeroderivative gas turbines drive the compressor stages whereas a combined cycle is the selected power plant. The total cost of this system is 211.00 MM$, out of which 206.81 MM$ are capital expenditure. This is considerably cheaper than solution BC-1 (total cost decreased by 14.2% and capital by 5.8%) and is partly due to the fact that there is a significant capital saving by not investing in a spare power plant. Also, the efficiency improvement due to the introduction of a combined cycle power plant contributes towards fuel savings. However, a system without electricity generation redundancy is very unlikely to be considered in practice.

Systems BC-1 and BC-2 feature three dissimilar types of gas turbine drivers. This is often not desired since it would imply different maintenance regimes and more spare parts to be kept in the warehouse, which is also a capital investment. Without great difficulty, extra constraints can be added to the formulation to limit the gas turbine driver types to a predefined maximum. The same can be done regarding power plants. Figure 3 shows the optimal power system for the base case scenario subject to identical gas turbine drivers and power plants (BC-3). The solution features two identical Frame 7 gas turbine drivers and a combined cycle power plant with its corresponding spare unit. The total cost is now 267.78 MM$, out of which 235.19 MM$ are capital cost. This is 8.9% more expensive than BC-1 in total cost, due to the lack of aeroderivative units. Aeroderivative gas turbine drivers cannot participate in this solution as identical drivers are being enforced, and the largest aeroderivative unit (LM6000 with 39.54 MW de-rated output) is not large enough to drive the first stage of the mixed refrigerant compressor alone (57.64 MW) even with a helper motor. System BC-3 tackles thermal efficiency by using a combined cycle power plant instead. The remainder of this case study will assume no constraints on the number of dissimilar equipment.

Table 5. Parameter Values for the Base Case

Parameter	Value
Equipment	
Maximum main motor driver	60 MW
Maximum steam turbine driver	60 MW
Maximum helper motor size	25% of GT
Maximum helper generator size	25% of GT
Power required for DDGT start up	15% of GT
Motor efficiency	95%
Helper generator efficiency	95%
Mechanical transmission losses	1.5%
Steam	
Boiler and HRSG blowdown ratio	3.0%
Steam losses	2.0%
VHP Steam Level	490/85°C/bar
HP Steam Level	390/40°C/bar
MP Steam Level	280/15°C/bar
LP Steam Level	140/3°C/bar
Condensing mains pressure	0.20 bar
Minimum temperature difference	20°C
Minimum stack temperature	130°C
Electricity	
Electricity distribution losses	2%
Maximum electricity export	25 MW
Maximum electricity import	0 MW
Price of sold electricity	0 $/kWh
Cost of imported electricity	0 $/kWh
Fuel and CO_2	
Specific CO_2 emissions	0.199 kg/kWh LHV
CO_2 emissions penalty	0 $/ton
Cost of end flash fuel	0 $/kWh LHV
Cost of fresh fuel	1 $/kWh LHV
Maximum end flash fuel available	540.3 MW LHV
Other	
Plant lifetime	25 yr
Interest rate	4%



(a)

(b)

Figure 2. Optimal no-steam solution for the base case.

(a) Optimal solution to the base case scenario (system BC-1), (b) Optimal solution to the base case scenario without sparing philosophy (system BC-2). [Color figure can be viewed in the online issue, which is available at wileyonlinelibrary.com.]

Sensitivity to fuel costs

A sensitivity analysis was performed considering five different fuel cost scenarios. The detailed results are reported in Table 6. As it was expected, the higher the fuel penalties, the more efficient the system becomes, as indicated by the decreasing total fuel consumptions (with the exception of systems S-3 and S-4 which are identical). This reduction is achieved with the progressive use of aeroderivative gas turbines and combined cycle power plants. As discussed before, motor drivers increase significantly the electricity demand, which opens the opportunity of using larger and more efficient power plants. However, these are not featured until the fuel incentive is large enough to overcome the associated increase in capital costs. Such is the case in Scenarios 3, 4, and 5. The capital costs are higher for the more efficient systems, as also expected.

Figure 4 illustrates how the systems in Table 6 perform economically under the five scenarios considered. The differences in capital cost and thermal performance between these systems lead to a number of break-even situations. For instance, it can be seen that system S-1 is the best choice for the capital only scenario but, as fuel cost increases, system S-2 (BC-1) takes over around half way between Scenarios 1 and 2. In a similar way, system S-3/S-4 takes over as the best choice from slightly before Scenario 3 until slightly after Scenario 4, when system S-5 becomes the one with lowest total cost among the options. It can also be appreciated how system S-1 becomes uncompetitive relatively quickly after Scenario 2, whereas the gap between systems S-3/S-4 and S-5 is small due to their similar fuel consumption.

Availability considerations

When estimating the present value of lost production due to the scheduled and unscheduled downtime of major rotating equipment, the system BC-1, which is in series, incurs 225.65 MM$ of losses throughout the plant lifetime. Considering this figure as an additional cost alongside capital and fuel costs, then the total cost incurred by system BC-1 would be 471.52 MM$.

System BC-P1, featuring parallel compression, was obtained as an alternative to BC-1 when minimizing capital and fuel costs for the base case scenario. Having compressors and drivers in parallel allows BC-P1 to reduce the production losses by 37.8% with respect to BC-1. Featuring all-aeroderivative gas turbines and a smaller power plant to spare also allow system BC-P1 to contribute modest fuel and capital savings, leading to an overall 17.8% of cost reduction relative to BC-1. Such a saving was obtained merely by considering parallel compression in the problem formulation. More details are presented in Table 7.

After introducing the lost profit as an additional cost in an extended objective function, the systems AV-S1 and AV-P1 were obtained as the optimal systems in series and in parallel, respectively, for the base case scenario. The details of these are shown in Table 7. For the series case, AV-S1 results an all-motor solution with three combined cycle power plants and one aeroderivative generator. Although

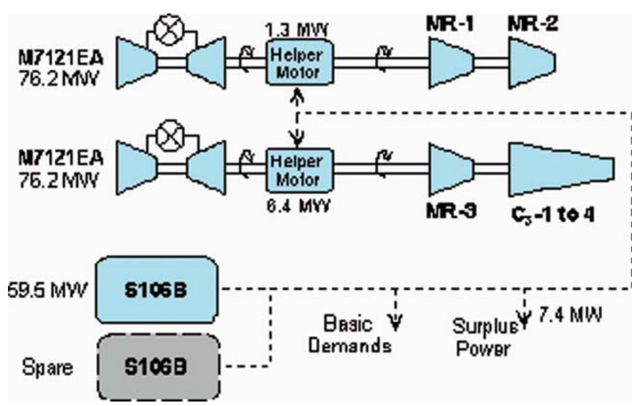


Figure 3. Optimal no-steam solution for the base case with identical units.

[Color figure can be viewed in the online issue, which is available at wileyonlinelibrary.com.]

Table 6. Optimal Power Systems for Different Fuel Cost Scenarios

Scenario	1	2 (base)	3	4	5
Fresh fuel ($/MMBtu)	–	1.0	2.0	3.0	5.0
End flash fuel ($/MMBtu)	–	–	1.0	2.0	3.0
System ID	S-1	S-2 (BC-1)	S-3	S-4	S-5
Driver arrangement*	M7111EA+13.4 MW HG for MR-1 M7121EA+2.3 MW HG for C_3-1 to 4 & MR-2 M6581B+6.0 MW HG for MR-3	M7121EA+17.4 MW HG for MR-1 Coberra6761+6.6 MW HG for C_3-1 to 3 LM6000+6.7 MW HG for C_3-4 LM6000+7.2 MW HM for MR-2 & 3	Coberra6761+3.9 MW HM for C_3-4 Coberra6562+0.8 MW HG for C_3-1 to 3 LM6000+7.2 MW HM for MR-2 & 3 58.52 MW Motor for MR-1	Coberra6761+3.9 MW HM for C_3-4 Coberra6562+0.8 MW HG for C_3-1 to 3 LM6000+7.2 MW HM for MR-2 & 3 58.52 MW Motor for MR-1	LM6000+7.2 MW HM for MR-2 & 3 58.52 MW Motor for MR-1 54.88 MW Motor for C_3-1 to 4
Electricity generation	1xPG5371PA	1xPG5371PA	2xS106B	2xS106B	1xS109E
Spare power plant	PG5371PA	PG5371PA	S106B	S106B	S109E
Total fuel (MW)	669.7	599.21	491.01	491.01	451.70
Capital cost (MM$)	201.69	219.51	281.77	281.77	318.97
Fuel cost (MM$)	–	26.36	219.72	439.43	606.38
Total cost (MM$)	201.69	245.87	501.49	721.20	925.35

*HM, helper motor; HG, helper generator.

significantly more expensive in capital than BC-1 (45.9%), this solution brings a dramatic improvement in lost production (78.6% less), which leads to an overall cost reduction of 21.9% against BC-1. Even when system AV-S1 is a series system, such an improvement in lost production makes it more economically competitive than BC-P1, which is in parallel. The significant advantage of AV-S1 is due to the fact that motors are more reliable than gas turbine drivers, and that the minimum number of compressor casings (three) is achieved by this design. Clearly, an all-motor system would relocate nearly all the availability issues to the electricity generation equipment, but this has already been taken care of by the power plant sparing philosophy.

The runtime to obtain solution AV-S1 was 327 CPU seconds using a Pentium IV processor (3.0 GHz) with 512 Mb of RAM and using DICOPT as the MINLP solver with the CONOPT and Cplex solvers for the NLP and MIP subproblems, respectively. However, it was necessary to run the 100% availability subproblem first to initialize the MINLP solver with a feasible initial solution. As reported before, this takes around 508 additional CPU seconds for a problem of this size. The reason why the variable availability problem was solved faster than the 100% availability one is due to the fact that electric motors are especially attractive in this case and, as they involve less configuration decisions than gas turbine drivers (i.e., there is a unique type of motor but 14 types of DDGTs considered in this case study), the optimization task is likely to require fewer internal operations. By considering a superstructure size as four potential gas turbines drivers, four potential electric motors, and four potential packaged power plants, the MINLP GAMS model generated 1133 equations and 1221 variables, out of which 464 were binary.

The lowest total cost in Table 7 is achieved by solution AV-P1 (350.64 MM$), which is the optimal parallel compression system obtained with the extended objective function. It features two aeroderivative gas turbine drivers and two Frame 5 generators in simple cycle. The reductions in lost production and total cost reach 51.8% and 25.6%, respectively, against the reference system in series (BC-1), while capital and fuel costs are not too different. Besides the

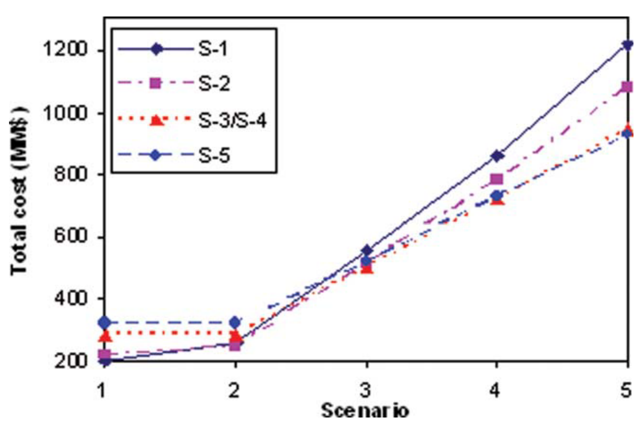


Figure 4. Fuel cost sensitivity plot for the systems in Table 4.

[Color figure can be viewed in the online issue, which is available at wileyonlinelibrary.com.]

Table 7. System Comparison Considering Lost Production

System ID	Base Case in Series BC-1	Base Case in Parallel BC-P1	Extended Objective Function in Series AV-S1	Extended Objective Function in Parallel AV-P1
Driver arrangement*	M7121EA+17.4 MW HG for MR-1 Coberra6761+6.6 MW HG for C_3-1 to 3 LM6000+6.7 MW HG for C_3-4 LM6000+7.2 MW HM for MR-2 & 3	Two identical sets of: LM6000+5.4 MW HG for C_3-1 & 2 & MR-1 Coberra6761+6.1 MW HG for C_3-3 & 4 Coberra6761+5.3 MW HG for MR-2 & 3	58.52 MW Motor for MR-1 46.78 MW Motor for MR-2 & 3 54.88 MW Motor for C_3-1 to 4	Two dentical sets of: LM6000+0.8 MW HG for MR-1 & 2 LM6000+1.8 MW HM for C_3-1 to 4 & MR-3
Electricity generation	1xPG5371PA	1xLM2000	3xS106B 1xLM6000PC	2xPG5371PA
Spare power plant	PG5371PA	LM2000	S106B	PG5371PA
Capital cost (MM$)	219.51	237.87	320.16	232.13
Fuel cost (MM$)	26.36	9.40	–	9.72
Lost profit** (MM$)	225.65	140.38	48.33	108.78
Total cost (MM$)	471.52	387.65	368.49	350.64

*HM, helper motor; HG, helper generator.
**Assuming profit of $50 per metric ton of LNG.

step improvement introduced by parallel compression alone, the reduction in the number of drivers from four to two in each compression set also contribute towards the availability improvement. When compared against parallel system BC-P1, AV-P1 also shows a clear cost reduction, although of a minor extent. Namely, the lost profit is reduced by 22.5% and the overall cost by 9.5%. In this case, the savings are due to a reduction in the number of drivers from three to two in each compression set.

Steam vs. no-steam

So far, steam-based drivers and generators have been neglected and only the base superstructure (as shown in Figure 5a in the Part 1) has been used. Extending such a superstructure to include all the steam-related options, as shown in Figure 5b in the Part 1, results in a new optimal solution to the base case scenario. This system is named BC-S1 and is shown in Figure 5. It has a total present cost of 233.04 MM$ (capital, fuel, and water treatment), which is 5.2% lower than the optimal no-steam solution BC-1. The capital cost (231.71 MM$), however, is slightly higher (5.6%). Also, the system is now considerably more complex. It features one simple cycle gas turbine generator as the main source of electricity and two Frame 7 gas turbine drivers for the first stage of the mixed refrigerant compressor and the whole of the propane compressor. Heat is recovered from all gas turbines to raise 216.9 t/h of VHP steam, which is

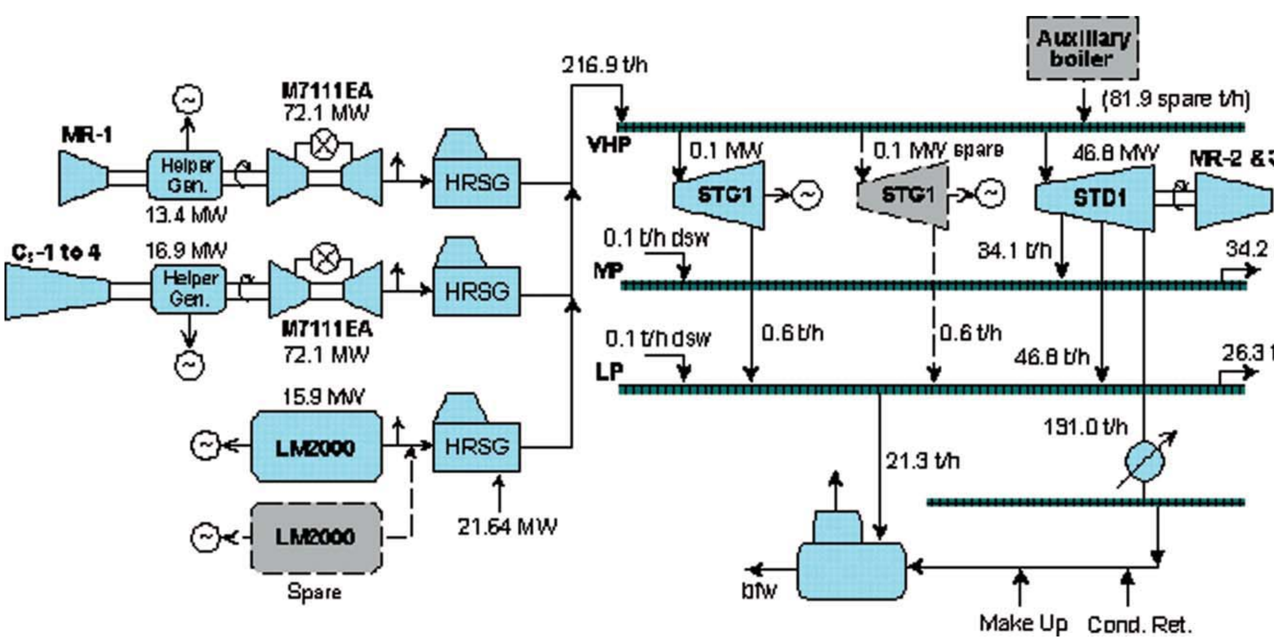


Figure 5. General optimal system for the base case scenario (BC-S1).

[Color figure can be viewed in the online issue, which is available at wileyonlinelibrary.com.]

enough for running one steam turbine driver, one steam turbine generator, and providing the required process heating through MP and LP steam. Three redundant elements are considered; a simple cycle gas turbine generator, a steam turbine generator, and an auxiliary boiler with enough capacity to cover for the breakdown of any one of the heat recovery units. The steam system features three steam headers (VHP, MP, and LP) and a condensing main. The steam turbine driver has two extraction streams to intermediate steam levels and the remaining steam is expanded down to the pressure in the condensers. A very small amount of desuperheating water is added to the MP and LP headers to be consistent with the fixed steam conditions assumed initially. In a practical application, however, this water injection is likely to be eliminated by slightly adjusting the extraction flows from the steam turbine driver. Also, in practice, the very small steam turbine generator and its corresponding spare are likely to be eliminated, for example, by shifting a small amount of compression duty from the first to the second and third MR compressor stages, so that the electricity exported from the associated helper generator can be increased by the required 0.1 MW. Facilities for the production of demineralized water are also considered in the cost of this system.

For each of the fuel cost scenarios defined in the previous sensitivity analysis, the optimal general solution, the optimal no-steam solution, and the optimal pure-steam solution were obtained. The comparison of the results is shown in Figure 6 in the form of relative total costs. The first thing to notice is how expensive the pure-steam optima are when compared to the no-steam optima and the general optima in all the scenarios considered. They feature total costs between 50 and 90% higher than their respective general optima. As expected, the general optimal solution to the Scenario 1 turned out to be a no-steam solution, since only capital costs are considered in the objective function. The details for Scenario 2 (base case scenario) have already been discussed previously, where a hybrid gas turbine/steam turbine driver solution was the best option. A hybrid gas turbine/steam turbine driver solution still dominates in Scenarios 3–5 but each time the advantage over the no-steam solution becomes smaller. Finally, an additional sixth scenario was added (both fuels at 6.5 $/MMBtu) to show that no-steam solutions regain optimality at high fuel costs, this time with all-motor drivers.

The overall trend in Figure 6 is that no-steam solutions result the best choice either at very low or at high fuel costs. In the first case, they are dominant because they can provide a relatively low capital cost. In the latter, it is because, with the increased use of motor drivers, the sought thermal efficiency improvement is provided by the use of larger and more efficient packaged combined cycle power plants, sometimes with dual and even triple pressure HRSGs. Besides, as discussed before, the pre-engineered nature of these power plants makes them more cost-effective than custom combined cycles. It is at fuel costs as high as 6.5 $/MMBtu that the trade-off between capital and efficiency favors all-motor solutions. Although for the intermediate scenarios (2–5) hybrid gas turbine/steam turbine systems prove optimal, the saving against nonsteam systems does not exceed 7.5% at most (Scenario 3). A more detailed risk and availability

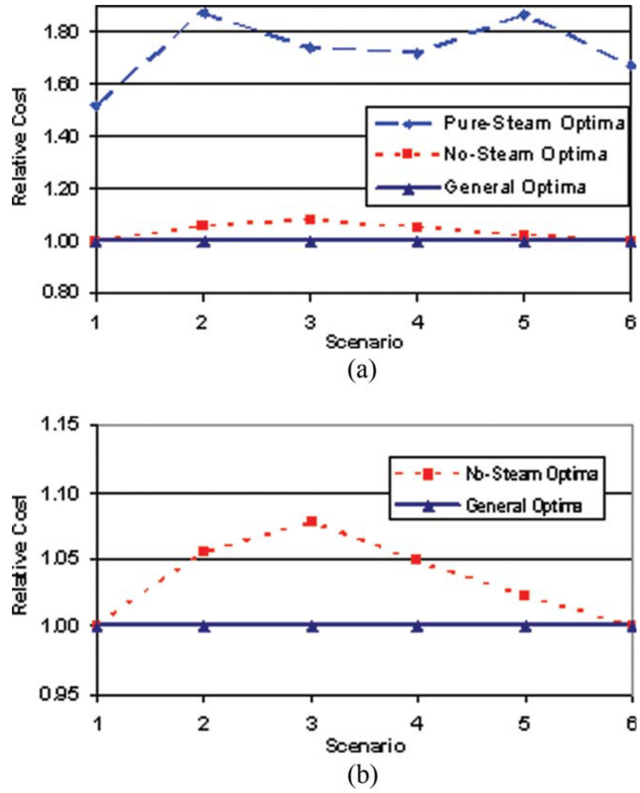


Figure 6. Steam vs. no-steam systems: relative cost comparison.

(a) Full scale, (b) Zoomed scale. [Color figure can be viewed in the online issue, which is available at wileyonlinelibrary.wiley.com.]

analysis would be required to ascertain whether this amount of savings is worth the extra complexity.

An Integrated and Simultaneous Design of Refrigeration and Power Systems

An integrated approach

The use of vapor-compression refrigeration in low temperature processes creates significant demands for mechanical power. The power system supplying this energy will have an impact on the performance and economics of these processes. However, the interactions between refrigeration and power systems are strong and must be exploited to achieve a full and efficient capital utilization. For instance, the power system design approach departs from a given set of energy demands (heat, electricity, and mechanical power) and assumes that the refrigeration system has already been designed, without challenging the convenience of its features. On the other hand, although the approach for optimizing refrigeration systems is able to consider some capital costs (including compressors) it does not foresee the economic and thermal performance implications that the resulting solution will have on the driver and power plant selection, as illustrated in Figure 7.

Usually, these systems are designed in sequence; refrigeration system first and then the power system, in a sort of ''manual'' iteration loop, until the designer is satisfied with

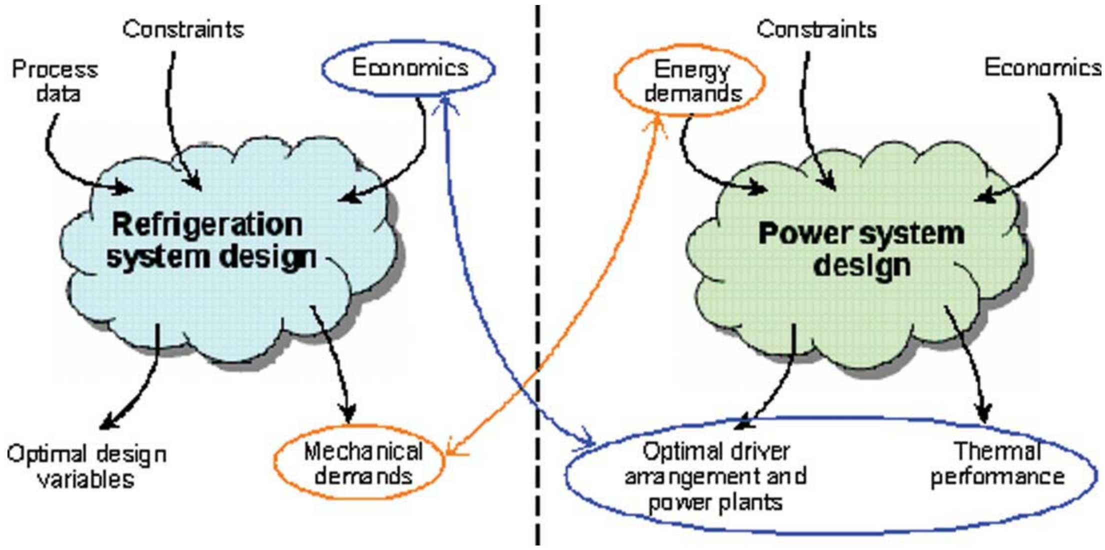


Figure 7. Interactions between refrigeration system design and power system design.

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the outcome. Such a nonsystematic procedure is unlikely to produce truly optimal solutions (or near-optimal, for practical purposes), unless a very significant amount of effort and time are invested. Ideally, both systems should be optimized simultaneously as one single design problem. Although the methods discussed in the first part of the article have been successfully applied to the design of power systems, this only focus on solving one part of a bigger problem. Therefore, a systematic methodology for the integrated design of refrigeration and power systems has been developed.

Design of refrigeration systems

Subambient operating conditions in process industries are mainly provided by a refrigeration cycle based on the recompression of vapor, either using a pure fluid or a mixture of refrigerants. For the refrigeration cycle with a pure fluid, the cooling in the evaporator is carried out at a constant temperature. However, this is not energy-efficient when cooling at very low temperature is required. The energy efficiency can be improved by mixing refrigerant fluids in the cycle, with which isobaric phase change during the evaporation is continuously made, along a range of temperatures. This flexibility provides a significant potential to match between cooling profile and the process stream profile to be cooled, without the introduction of complexity into the cycle.

An approach for the optimal design of mixed refrigerant cycles has been proposed by current authors.[14] It considers multistage refrigerant compression with intercooling, full enforcement of the minimum temperature difference in the heat recovery, simultaneous optimization of design variables, systematic trade-off, and the application of genetic algorithm (GA) to overcome local optima. The effectiveness of the method was illustrated by revisiting previously published LNG case studies, for which better and feasible solutions were produced. The importance of considering multistage compression has been also illustrated, while the application of GAs provides computational confidence in the optimality of the results.

The base model had been extended to support generic mixed refrigerant cycles in cascade (Figure 8). Two MR cycles are coupled together at intermediate level and each cycle has two stages. The design variables are the refrigerant compositions and flowrates, the intermediate temperatures between heat exchangers, the suction pressures, discharge pressures, and stage compression ratios for compressors and the partition temperature. Flowsheet options such as the number of heat exchanger sections, or stages, for each cycle, whether the hot streams are precooled against the warmer cycle or not, subcooling or not subcooling the high pressure liquid refrigerant streams and the use of either liquid expanders or Joule-Thompson valves, can be set by the designer

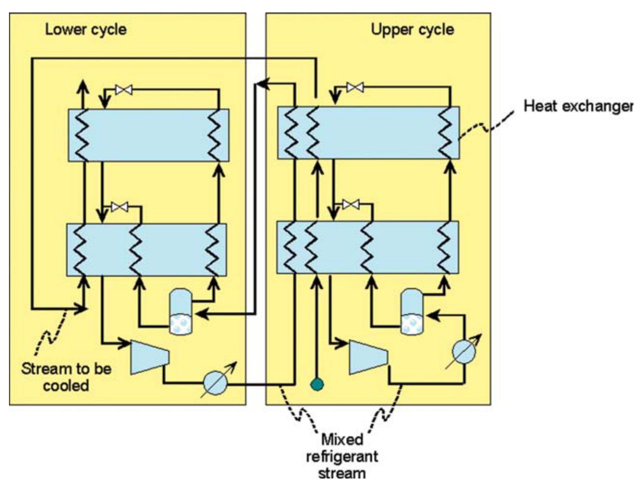

Figure 8. MR cycles in cascade.

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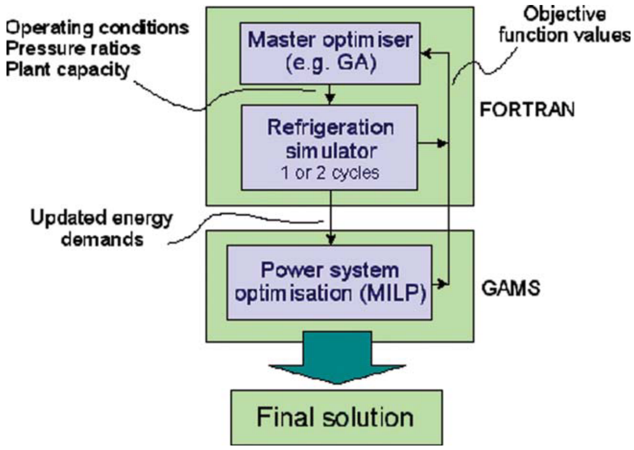

Figure 9. Integrated optimization strategy.

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before optimization. The details of mathematical formulation for modeling and optimization framework can be found in Del Nogal et al.[14]

The optimizer (GA in this case) proposes a series of candidate solutions and relies on the simulator for their assessment. How the optimization task evolves is decided by the GA based on the "fitness" of the candidate solutions as dictated by the objective function.[15,16] The interactions between the GA and the simulator result in a set of the best solutions found over a discretized solution space. Standard nonlinear programming (NLP) optimization(s) can be carried out afterwards having the best discretized solution(s) as initial guess to fine-tune and finally report the optimal solution on the basis of a continuous solution space.

Simultaneous optimization

Synthesis of power systems and refrigeration systems has a very different mathematical nature. The refrigeration system design problem is a highly nonlinear one (mainly due to physical properties) with relatively few variables, all of which are continuous. Conventional NLP methods are likely to get trapped in local optima in these circumstances and, therefore, a stochastic optimizer has been proposed.[14] On the other hand, in the synthesis of power systems, the majority of the many variables subject to optimization are binary, due to discrete equipment choices. Although the formulation for power system design is in principle mixed integer nonlinear (MINLP), a mixed integer linear formulation (MILP) can be derived from it without a significant loss of accuracy, thus improving the robustness of the solver used and ensuring global optimality. If these two formulations (i.e., refrigeration and power systems) were merged in a single optimization problem, the resulting formulation would be unavoidably highly nonlinear and with many variables, most of them binary, which would seriously compromise the robustness of the solver and the quality of the solution.

Given the above considerations, Figure 9 illustrates the optimization framework adopted in this work. A GA-based approach is applied. In this case, for each candidate solution, the power system is optimized (in GAMS) after the refrigeration cycles are simulated (in FORTRAN). The results form both steps contribute to a unique objective function which is then evaluated and its value sent as feedback to the optimizer. Although not a simultaneous optimization, this strategy still permits the master optimizer to realize and exploit the interactions between both subsystems. Furthermore, it permits solving each subproblem with the most appropriate formulation and optimizer, as well as keeping them in their original software platform, provided the feasibility of a two-way communication. An optional initial guess can be provided to the Master Optimizer to speed up convergence, although this is not strictly necessary. This initial guess can be obtained from standalone optimization of the refrigeration system with an objective function set to either minimum power or minimum capital cost.

The manipulation of key degrees of freedom such as compression ratios will ultimately facilitate a feasible match between a set of mechanical and electricity demands and a particularly attractive set of drivers and/or power plants. At this moment, it is also useful to introduce a new optimization variable such as plant capacity. Allowing the plant capacity to vary, at least within a restricted space, would permit the master optimizer to ensure a full utilization of the equipment capacity for a given machinery selection. If the objective function is targeting costs, there is now the need to ensure these are defined on a specific basis (i.e., cost per unit of plant capacity). As absolute costs are expected to increase at higher capacities, there would be an implicit bias towards lower capacities if costs were expressed on an absolute basis.

Case Study 2: Integrated Design of Refrigeration and Power Systems

Process data

A natural gas stream is to be liquefied using two mixed refrigerant cycles in cascade to produce 6 MTPA of LNG. The temperature-enthalpy profile of the natural gas stream is given in Figure 10. MW (462.8) of combustion heat (LHV) are available from the end flash system. Any amount of fuel demand over this figure would be at the cost of 1 \$/MMBtu of LHV. The electricity demand is 39.9 MW excluding any helper motors. Power plants are subject to N + 1 sparing philosophy.

Nonintegrated design: Base case

An initial nonintegrated design was obtained by optimizing only the refrigeration system with minimum power as objective function and a ΔT_{min} of 3°C, followed by synthesis of the power system. It has to be said that targeting capital plus fuel costs in the refrigeration optimization step would not be appropriate as first, the actual fuel demand will not be known until the power system optimization step and second, assuming any kind of trend regarding fuel consumption would be considerably inaccurate as this is subject to the decisions on driver and power plant selection. Also, note that minimizing power will have, in practice, the same effect as minimizing capital cost as compressor costs are dominant in this subproblem.

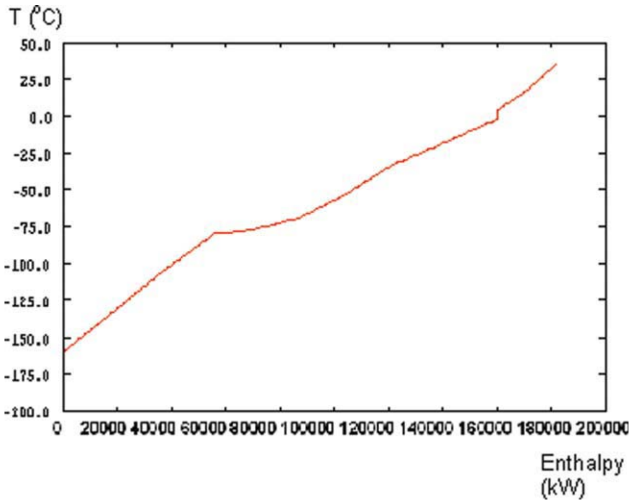

Figure 10. Temperature-enthalpy profile of the process stream.

[Color figure can be viewed in the online issue, which is available at wileyonlinelibrary.com.]

The features of the refrigeration system with minimum compression power are summarized in Table 8. The total mechanical demand from the refrigerant compressors is 199.44 MW, or 33.24 MW per metric ton/yr of LNG, with eight compression stages in total, and the partition temperature is −56.3°C. The power system, once optimized for this input data, makes use of four gas turbine drivers; three LM6000, and one Frame 7 units, with a Trent gas turbine as simple cycle generator plus the respective spare unit. For this ''sequential'' or nonintegrated design of refrigeration and power system, the total cost is 299.93 MM$, or 49.99 $ per metric ton of annual capacity, which includes the installed cost of major equipment (e.g., compressors and drivers, power plants, main heat exchangers, at 295.37 MM$ or 49.23 $/t.yr LNG) and the present value of the fuel costs along the plant lifetime (4.56 MM$ or 0.76 $/t.yr LNG).

Table 8. Minimum Power Refrigeration System

	Lower Cycle	Upper Cycle
Refrigerant flowrate (kmol/s)	11.90	18.70
Refrigerant composition (mol %)		
N_2	11.09	0.08
CH_4	40.08	9.78
C_2H_6	35.33	41.06
C_3H_8	13.03	11.80
n-C_4H_{10}	0.47	37.28
Compressor inlet pressure (bar)	1.99	4.00
Compressor outlet pressure (bar)	42.21	37.29
Compression ratios		
Stage 1	3.00	2.91
Stage 2	3.00	1.43
Stage 3	1.56	1.51
Stage 4	1.51	1.48
Compression power (kW)		
Stage 1	31,204	58,804
Stage 2	39,850	15,842
Stage 3	16,398	13,514
Stage 4	14,435	9308

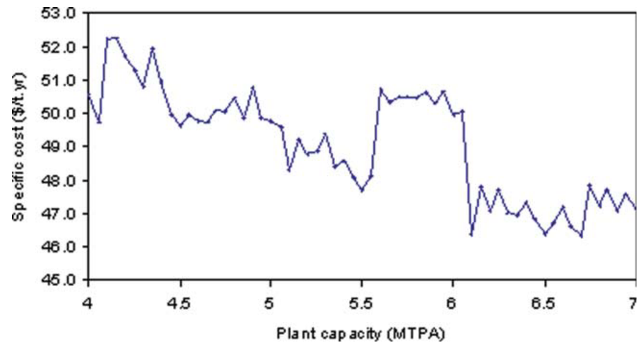

Figure 11. Sensitivity analysis results.

[Color figure can be viewed in the online issue, which is available at wileyonlinelibrary.com.]

To illustrate the influence of plant capacity and the discrete equipment options over the overall system design, a sensitivity analysis was carried out. All the variables in the refrigeration system were kept the same as in the previous solution and the plant capacity was varied between 4 and 7 MTPA. For each capacity, the optimal power system was obtained. The results are summarized in Figure 11. As a result of the economy of scale, the overall specific cost trend is to decrease as the plant capacity increases. However, there are numerous and sudden upsets in the trend due to the discrete decisions in equipment selection and configuration. These sudden changes sometimes pull in the same direction of the economy of scale and help in reducing the specific cost, but sometimes go in the other direction and are big enough to overcome the effect of the economy of scale at least locally. Compression ratios have a similar effect as plant capacity on the total specific cost. This is why these variables are key in achieving minimum total cost through a fine match between power demand and supply.

Optimal design with an integrated approach

The integrated optimization strategy was applied in this case study, aiming for minimum specific cost. The plant capacity, within a range of ±10%, and the compression ratios were subject to optimization. The other variables were kept the same as in the minimum power solution (i.e., refrigerant compositions, suction, and discharge pressures and partition temperature), as they provide an acceptable match between the composite curves, except refrigerant flowrates, which were pro-rated according to the plant capacity. The resulting integrated solution has a capacity of 5.92 MTPA LNG, which is not too far from the original 6 MTPA.

The optimization framework allowed a reduction of the specific cost by 8.7% to $45.66 per metric ton of annual capacity. This integrated solution features four Trent gas turbine drivers and two Frame 5 generators in simple cycle plus one spare unit. As expected, the surplus power is nearly zero, which indicates that the capacity of this power system configuration is being fully utilized.

It should be noted that annual capacity (6.1 MTPA) at lowest specific cost ($46.5 per metric ton of annual capacity) obtained from Figure 11 is different to the optimal capacity

Table 9. Integrated Solution

	Lower Cycle	Upper Cycle
Compression ratios		
Stage 1	2.99	2.47
Stage 2	2.96	1.68
Stage 3	1.59	1.53
Stage 4	1.50	1.47
Compression power (kW)		
Stage 1	30,738	48,884
Stage 2	38,750	26,366
Stage 3	16,962	13,281
Stage 4	14,007	8548

(5.92 MTPA with \$45.66 per metric ton of annual capacity) identified from an integrated approach. In Figure 11, the plant capacity is only changed, whereas the structure of refrigeration system and their interactions with power systems are not fully optimized. This explains why the lowest specific cost from Figure 11 is higher than optimal result.

Table 9 shows the compression ratios and compression powers of the integrated solution. The total compression power in this case is 197.62 MW. Note that, although the specific power has increased slightly to 33.41 MW per metric ton/yr of LNG, the savings resulting from a better match between energy demands and supply are large enough to overcome this. From Table 9, the main difference between the integrated and the nonintegrated solutions is in the compression ratios of Stages 1 and 2 of the upper (i.e., pre-cooling) MR cycle. With the integrated solution compression ratios, and the consequent stage powers, it is now possible to accommodate the first stage of the upper cycle compressor in one aeroderivative gas turbine driver, more efficient than the previous Frame 7. Furthermore, all the compression stages can now be grouped in such a way that they are all feasibly driven by Trent gas turbine drivers, slightly more efficient than the LM6000 units in the previous solution. The specific fuel cost was reduced by 59.9% to 0.30 \$/t.yr LNG and the specific capital cost by 7.9% to 45.36 \$/t.yr LNG. This was possible thanks to the adaptation of the refrigeration systems, through the rearrangement of the energy demands, to exploit a more cost-effective driver and power plant configuration with minimum impact on the main process features.

This case study has centered in minimizing costs. A further assessment would be required to ensure that the reduction in profit resulting from the decreased production capacity do not overcome the cost savings. Also, different solutions could be obtained depending on the driver and power plant options and the relative size of helper motors allowed by the designer.

Conclusions

The problem formulation was applied to an LNG case study, proving to be appropriate, robust, and flexible enough to produce solutions ranging from no steam at all to pure-steam ones. It was demonstrated how the optimal solutions evolve and use more thermally efficient components such as aeroderivative gas turbines, steam turbine drivers with waste heat recovery, combined cycle power plants, and motor drivers when faced with increasing fuel costs. The use of parallel compression was explored and found to have a significant positive impact on the economic performance of the project. Furthermore, accounting for scheduled and unscheduled downtime directly in the objective function allowed a dramatic improvement in the total cost of the optimal series compression system. Incorporating availability in the objective function introduces a new dimension to the problem and allows considering capital/energy/availability trade-offs instead of the classic capital/energy approach. It was also illustrated how sparing philosophy can also be considered to tackle the availability of electricity generation. The new approach for the synthesis of power systems allowed the exploration of an extended solution space and the exploitation of new trade-offs in the synthesis of energy systems, a distinctive advantage over the existing design methodologies without systematic availability considerations and without the flexibility for allowing nonsteam systems.

Although there is clearly a good potential for producing integrated refrigeration and power systems in an LNG context, doing so proves difficult without a systematic approach. This article also presented a framework for the integrated design of refrigeration and power systems. The framework is based on a combination of stochastic and MILP optimization methods and relies on recently developed approaches for the design of refrigeration and power systems individually. The robustness of the integration approach and the magnitude of the impact were proven in a case study. The systematic optimization of key degrees of freedom, such as compression ratios and plant capacity, allowed a better exploitation of the interactions between both systems, achieving an 8.7% of reduction on the specific cost of fuel and major capital equipment.

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