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78-GT-74

an ASME publication

Ten Years' Engineering Development in Gas Turbine Driven Natural Gas Compressor Stations

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The first generation gas compressor stations for N.V. Nederlandse Gasunie was designed in 1967 and equipped with aircraft derivative gasturbines of 11 MW driving single wheel centrifugal compressors. In later stations, similar machines and also industrial type gas turbines of 11 and 26 MW have been installed. With the system now including eight stations, a considerable experience was gained. Special described items are: (a) air filtration, (b) noise abatement, (c) vibration due to pipe resonance, and (d) also the operational experience with aircraft derivative gas turbines compared with industrial type machines is described. During 1974, due to the increase in the energy prices, more emphasis was placed on energy savings. Studies revealed that for our case, the new generation high efficiency gas turbines showed more advantage than equipping existing machines with available recuperators or waste heat boilers and steam turbines. Replacement of some base load machines began in 1975. Later, a new recuperator was designed together with a well-known boiler firm, the first of which will come in operation by the end of 1978 to improve the efficiency of a 26-MW machine. Medio 1978 a total of 35 units with 502.2-MW output will have been installed, of which five are equipped with high efficiency drivers.

Contributed by the Gas Turbine Division of The American Society of Mechanical Engineers for presentation at the Gas Turbine Conference & Products Show, London, England, April 9-13, 1978. Manuscript received at ASME Headquarters December 16, 1977.

Copies will be available until January 1, 1979.

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS, UNITED ENGINEERING CENTER, 345 EAST 47th STREET, NEW YORK, N.Y. 10017

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INTRODUCTION

The N.V. Nederlandse Gasunie operated at the end of 1977 a main transmission grid of 67-bar pressure of 3480-km length divided in: 206-km 18-in. ϕ, 302-km 24-in. ϕ, 139-km 30-in. ϕ, 880-km 36-in. ϕ, 817-km 42-in. ϕ, 571-km 48-in. ϕ, and 565-km smaller diameter.

The maximum design capacity, Winter 1977 to 1978, is 19 x 10^6 m^3_o/hr; the transported amount in 1977, 95 x 10^9 m^3_o, leading to a usage of the maximum design rating of 5000 hr. The overall load factor is, thus, 5000/8760 = 0.57.

The transmission grid comprises, in 1977, two systems (Fig. 1).

The Groningen gas system serving:	Dutch industry and power stations
(gas of a calorific LHV of 35.17 MJ/m^3_o)[1]	Gas distribution companies
	Export to Germany, Belgium and France.
The high calorific gas system serving:	Export to Italy and Switzerland
(dotted lines)	Transport of Ekofisk gas to Belgium and France
(gas of a calorific LHV of 40.19 MJ/m^3_o)[1]	Some large industries.

In the Groningen system, the export customers have the highest load factor followed directly by industry and power stations. The gas distribution companies have the lowest load factor (summer - winter relation, 1:6).

The high calorific system has a high load factor of 7000 hr = 0.80.

In a system of this size, every capacity increase means an intensive study whether a new pipeline should be laid (high investment, low maintenance costs), or a recompression station should be built (medium investment, usage of gas for recompression and appreciable maintenance).

When judging the Dutch system for Groningen gas, it should be taken into account that the Groningen field was discovered in 1959, Gasunie founded in 1963, and the first 400-km 36-in. pipeline from North to South was ready in December 1964. The main decision years for building the main Groningen transmission grid and its compressor stations were 1965 to 1971. In these years, the cost of fossil fuel was low, its supply seemed abundant and gradual, but a moderate price increase for this fuel was expected.

GASUNIE'S COMPRESSOR STATIONS (DESIGNED 1966-1971)

From 1964 till 1974, the main Groningen gas transmission system was completed. In this system, the following compressor stations (Fig. 2) were built. On the North-South line serving the nation and export customers — Ommen, Zweekhorst, and Ravenstein.

On the East-West route serving the densely populated western part through the IJssellak: Oldeboorn, Wieringermeer, and Beverwijk. The stations are located at a distance of approximately 75 km from each other.

The units taken into service between 1969 and 1974 bear the same characteristics; i.e., the ISO full-load gas turbine efficiency is between 26 and 27 percent, the highest available at that time in the required output range. When Gasunie carried out its main design studies in 1965 and 1966, gas turbine driven units were chosen because of:

[1] (m^3_o:1 m^3 natural gas at 0 C, 1013-mbar pressure.)

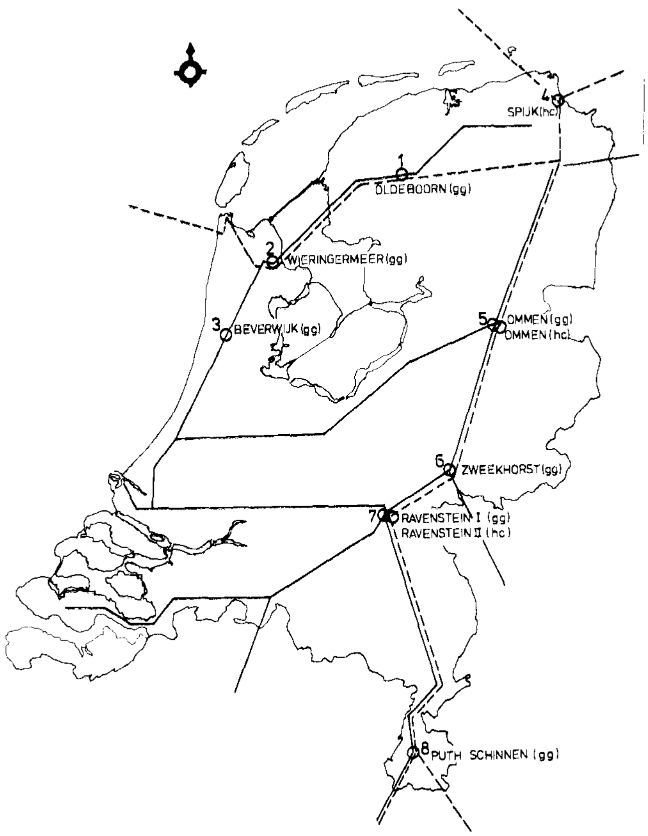

Fig. 1 Systematic layout of Groningen- and high calorific gas transmission systems with compressor stations

- Low investment
- Low maintenance costs compared with reciprocating machines
- High degree of automation, thus low personnel costs
- Low anticipated fuel costs.

This package ruled out alternatives as steam turbine driven units or electrically driven units.

The first series of machines consisted of three main designs, all in the 11-MW range and of split-shaft design:

- Aircraft derivative type (Rolls Royce Avon gas generators with Cooper Bessemer power turbines)
- Semi-industrial units (Stallaval GT 35 units)
- Full industrial units (G.E. frame size three machines).

In a later phase, more Rolls Royce units were added and on the main North-South trunk route, six additional General Electric frame 5 units driving 42-in. compressors were installed.

The number of operating hours of the various machines reflects their function. The machines in the Oldeboorn, Wieringermeer and Beverwijk stations on the East-West pipelines have accumulated 15,690 hr, thus 320 hr/year/machine/average year.

The machines on the North-South route Ommen, Zweekhorst and Ravenstein have accumulated 265,000 hr, thus 2010 hr/machine/year.

When judging these yearly operational hours, it should be borne in mind that 1969 to 1976 were very warm winters. Figs. 3 and 4 show layouts of two stations, Ommen and Ravenstein.

OPERATING EXPERIENCE WITH THE GAS TURBINE DRIVEN COMPRESSORS

Air Filtration

The first series of gas turbines we installed were equipped with inertia-type dust filters in the air inlets.

We soon found heavy fouling on the air compressor internals, necessitating water washing every 200 hr, later replaced by a soak wash and an unfired rinse using the starter motor.

This system kept the compressor in reasonable shape, but in the second year, corrosion became evident on the surface of the compressor blades as well as in the crevices at the blade root.

A research program was set up involving air sampling before and after the filters as well as a general air pollution study.

The results were the following:

1 The filters worked satisfactorily catching 90 percent of the dust of 10 μ and larger (Fig. 5).

2 The deposits in the compressor consisted of an oily layer with very fine dust.

3 The corrosion originated from sulfur compounds in the air and, to a minor extent, from chlorine. These compounds were in the form of fumes or vapors so we did consider it hopeless to try to catch them with any type of industrial filter. This corrosion only takes place in a wet environment, e.g., only on the first four rows of compressor blades. The most of Western Europe could be regarded as an industrial area (Fig. 6).

4 During stand-still of the machines, wind or draught deposited an appreciable amount of dust directly behind the filters. At the next start, this dust was transported into the engine and contributed to the fouling.

Year of application	station unit nr.	gas gen. made	Type	power turbine made	Type	ISO rating output MW	compr. made	Type	max. flow m^3_o/h	comp. ratio	Station control
1969	Ommen I P101	R.R.	56G	C.B.	R.T.48	11.3	C.B.	R.F.B. 36	2.10^6	1.22	Philips
	*Ommen I P102	R.R.	56G	C.B.	R.T.48	11.3	C.B.	R.F.B. 36	2.10^6	1.22	Philips
	Ommen I P104	R.R.	76G	C.B.	R.T.48	11.3	C.B.	R.F.B. 36	2.10^6	1.22	Philips
	Ommen I P105	R.R.	76G	C.B.	R.T.48	11.3	C.B.	R.F.B. 36	15.10^6	1.24	Philips
1970	Ommen II P201	G.E.	F.S.3			11.25	I.R.	C.D.P. 30	$1.6.10^6$	1.25	Philips
	Ommen II P202	G.E.	F.S.3			11.25	I.R.	C.D.P. 30	$1.6.10^6$	1.25	Philips
	Ommen II P203	G.E.	F.S.3			11.25	I.R.	C.D.P. 30	$1.6.10^6$	1.25	Philips
	Ommen II P204	G.E.	F.S.3			11.25	I.R.	C.D.P. 30	$1.6.10^6$	1.25	Philips
	Ravenstein 1-1	R.R.	56G	C.B.	R.T.48	11.3	C.B.	R.F.B. 36	$1.6.10^6$	1.25	G.B.-Entronic
	Ravenstein 1-2	R.R.	56G	C.B.	R.T.48	11.3	C.B.	R.F.B. 36	$1.6.10^6$	1.25	G.B.-Entronic
	Ravenstein 1-3	R.R.	56G	C.B.	R.T.48	11.3	C.B.	R.F.B. 36	$1.6.10^6$	1.25	G.B.-Entronic
	Wieringermeer 1-1	S.L.	G.T.35			9.9	D.L.	P.V.30	$1.6.10^6$	1.24	G.B.-Entronic
	Wieringermeer R	S.L.	G.T.35			9.9	D.L.	P.V.30	$1.6.10^6$	1.24	G.B.-Entronic
	Wieringermeer 1-2	S.L.	G.T.35			9.9	D.L.	P.V.30	$1.6.10^6$	1.24	G.B.-Entronic
1971	Ommen II P205	G.E.	F.S.3			11.25	I.R.	C.D.P. 30	$1.6.10^6$	1.25	Philips
	Ommen II P206	G.E.	F.S.3			11.25	I.R.	C.D.P. 30	$1.6.10^6$	1.25	Philips
	Ravenstein 1-4	R.R.	56G	C.B.	R.T.48	11.3	C.B.	R.F.B. 36	$1.6.10^6$	1.25	G.B.-Entronic
	Oldeboorn 1-1	R.R.	56G	C.B.	R.T.48	11.3	C.B.	R.F.B. 36	$1.4.10^6$	1.26	G.B.-Entronic
	Oldeboorn 1-2	R.R.	56G	C.B.	R.T.48	11.3	C.B.	R.F.B. 36	$1.4.10^6$	1.26	G.B.-Entronic
	Oldeboorn R	R.R.	56G	C.B.	R.T.48	11.3	C.B.	R.F.B. 36	$1.4.10^6$	1.26	G.B.-Entronic
1972	Ommen I P106	R.R.	76G	C.B.	R.F.48	11.3	C.B.	R.F.B. 36	$1.5.10^6$	1.24	Philips
1973	Beverwijk 1-1	R.R.	76G	C.B.	R.F.48	11.3	C.B.	R.F.B. 35	2.10^6	1.22	G.B.-Entronic
	Beverwijk 1-2	R.R.	76G	C.B.	R.T.48	11.3	C.B.	R.F.B. 35	$1.4.10^6$	1.25	G.B.-Entronic
	Zweekhorst 601	G.E.	F.S.5			23.9	C.B.	R.F.B. 42	$3.2.10^6$	1.26	General Electric
	Zweekhorst 602	G.E.	F.S.5			23.9	C.B.	R.F.B. 42	$3.2.10^6$	1.26	General Electric
	Zweekhorst 603	G.E.	F.S.5			23.9	C.B.	R.F.B. 42	$3.2.10^6$	1.26	General Electric

Year of application	station unit nr.	gas gen. made	Type	power turbine made	Type	ISO rating output MW	compr. made	Type	max. flow m^3_o/h	comp. ratio	Station control
1974	Ommen III P301	G.E.	F.S.5			23.9	C.B.	R.F.B. 42	$3.6.10^6$	1.25	Philips
	Ommen IV P401	G.E.	F.S.5			23.9	C.B.	R.F.B. 42	$3.6.10^6$	1.25	Philips
	Ommen IV P402	G.E.	F.S.5			23.9	C.B.	R.F.B. 42	$3.6.10^6$	1.25	Philips
1975	Ommen I P103	R.R.	R.B.211	C.B.	R.T.48	19.4	C.B.	R.F.B. 36	$3.2.10^6$	1.22	Philips
1976	Schinnen 1-1	Solar	C.E.R.			2.6	Solar	C 304	$0.14\text{-}0.24 . 10^6$	1.08	G.B.-Entronic
	Schinnen 1-2	Solar	C.E.			2.8	Solar	C 304	$0.14\text{-}0.24 . 10^6$	1.08	C.B.-Entronic
	Schinnen 1-3	Solar	S.A.			0.8	Solar	C 165	$0.08 . 10^6$	1.06	C.B.-Entronic
	Spijk R 1-1	D.L.S.	H.V.A.16	reciprocating engines		4.4	D.L.S.	M.C. 4	310.10^3	1.48	Honeywell
	Spijk R 1-2	D.L.S.	H.V.A.16	"		4.4	D.L.S.	M.C. 4	310.10^3	1.48	Honeywell
	Spijk R 1-3	D.L.S.	H.V.A.16	"		4.4	D.L.S.	M.C. 4	310.10^3	1.48	Honeywell
	Ravenstein II 2-1	D.L.S.	H.V.A.16	"		4.4	D.L.S.	M.C. 4	400.10^3	1.38	C.B.-Entronic
	Ravenstein II 2-2	D.L.S.	H.V.A.16	"		4.4	D.L.S.	M.C. 4	400.10^3	1.38	C.B.-Entronic
	Ravenstein II 2-3	D.L.S.	H.V.A.16	"		4.4	D.L.S.	M.C. 4	400.10^3	1.38	C.B.-Entronic
	Ravenstein II 2-4	D.L.S.	H.V.A.16	"		4.4	D.L.S.	M.C. 4	400.10^3	1.38	C.B.-Entronic
	Ravenstein II 2-5	D.L.S.	H.V.A.16	"		4.4	D.L.S.	M.C. 4	400.10^3	1.38	C.B.-Entronic
1977	Ommen I P102	R.R.	R.B.211	C.B.	R.T.56	19.4	C.B.	R.F.B. 42	$3.2 . 10^6$	1.22	Philips
	Ommen III P302	R.R.	R.B.211	C.B.	R.T.56	19.4	C.B.	R.F.B. 42	$3.2 . 10^6$	1.22	Philips
	Oldeboorn 4	R.R.	56G	C.B.	R.T.48	11.3	C.B.	R.F.B. 36	$2 . 10^6$	1.22	G.B.-Entronic
	* Reinstalled at Oldeboorn in 1977										
Planned 1978	Beverwijk 1-3 P103	R.R.	76G	C.B.	R.T.48	11.3	C.B.	R.F.B. 36	2.10^6	1.22	C.B.-Entronic
	Ravenstein 1-5	R.R.	Spey	C.B.	162	12.1	C.B.	R.F.B. 36	$2.3.10^6$	1.25	C.B.-Entronic
	Spijk 1-4	D.L.S.	H.V.A.16	reciprocating		4.4	D.L.S.	M.C. 4	310.10^3	1.48	Honeywell
	Spijk 1-5										
	Ommen III P301 (installation recuperator										

Legende:
- R.R. = Rolls-Royce
- G.E. = General-Electric
- S.L. = Stal-Laval
- D.L.S. = DeLaval
- C.B. = Cooper Bessemer
- I.R. = Ingersoll Rand
- C.E. = Centaur
- S.A. = Saturn
- C.E.R. = Centaur with recuperator

Fig. 2 Table with all data of Gasunie's compressor stations

Fig. 2 (Cont'd)

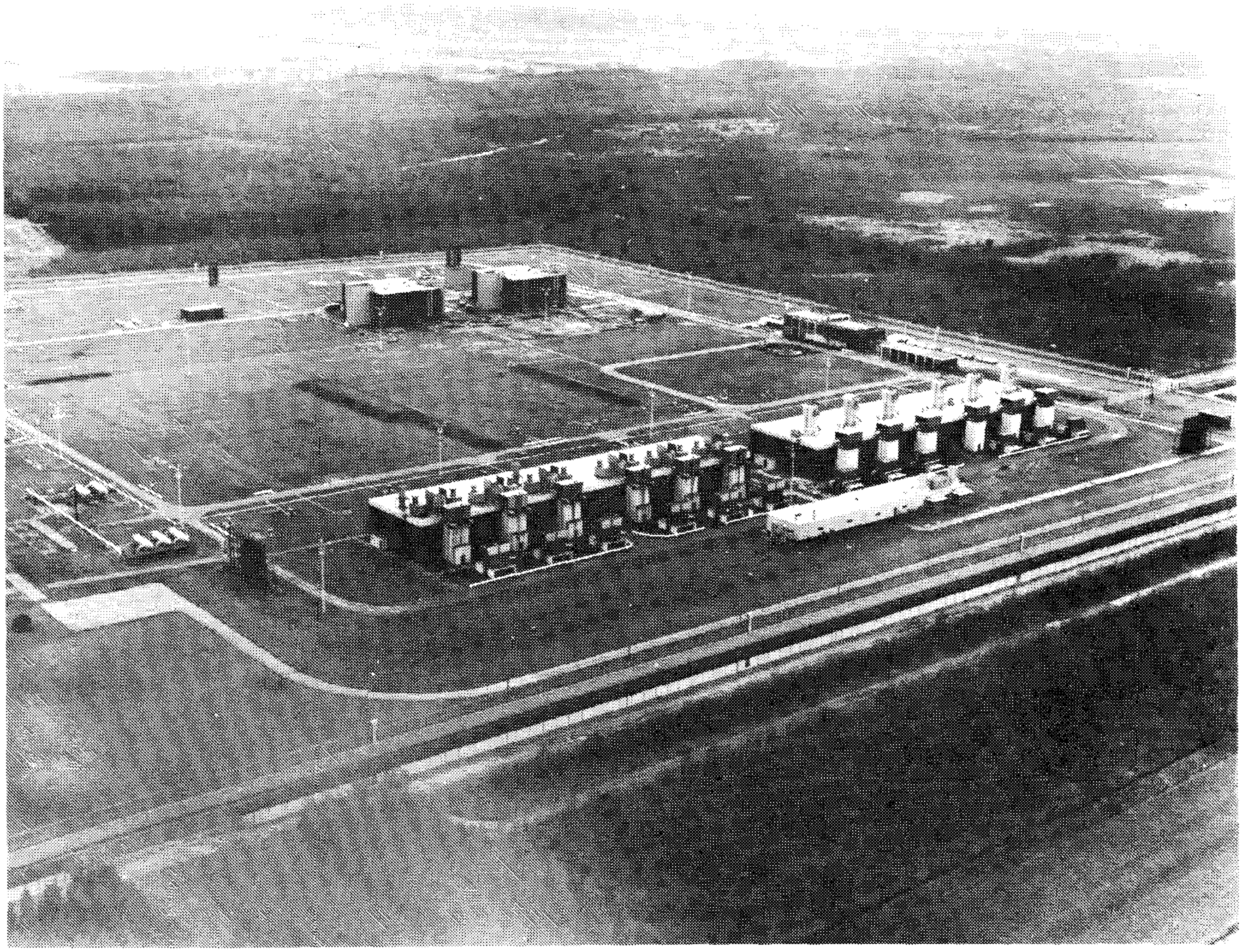

Fig. 3 Aerial picture of Ommen compressor station taken 1977. Total ISO output, 242 MW

The following measures were taken:

1 Aluminium blades were replaced by titanium blades; blade retaining pins were made of better corrosion-resisting material (first stage). Mild steel (low alloy) blades were coated with sermetel coating (four stages).

2 Motorized closing louvers were installed in front of the air inlet filters and operated by the start/stop logic.

3 Electric heating was installed under and around the front end of the compressor; this heating is switched on during standby in an effort to combat morning dew.

We have had no corrosion problems for three years since the aforementioned alterations have been made. Compressor cleaning is now done every 500 hr as liquid wash, sometimes as fired wash, and sometimes as crank soak wash.

Noise Abatement

Ten years ago, when designing low noise compressor stations for the first time, we aimed at silencing the high-frequency noise, so well known from bare gas turbines. The art of silencing was sufficiently advanced to design satisfactory inlet and outlet silencers. Some tests were necessary to get an exact picture of casing noise so that a good unit enclosure could be designed.

During operation, two noise problems became evident:

1 Gas compressor piping noise
2 Low frequency turbine outlet noise.

Gas Compressor Piping Noise. Gas piping noise was of high frequency and could be sufficiently dampened by insulating above-ground manifolds or by designing manifolds underground (Fig. 7).

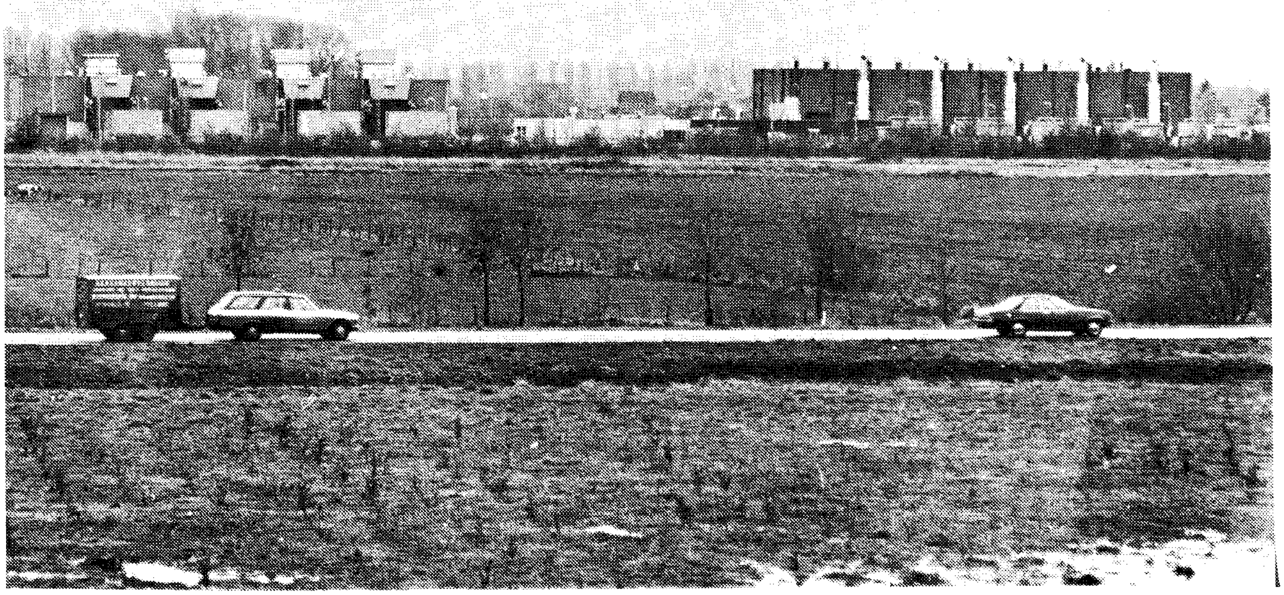

Fig. 4 Ravenstein compressor stations: Ravenstein I for GG-gas built in 1970, 4 x 11.3 MW; Ravenstein II for HC-gas built in 1976, 5 x 4.4 MW

Inside the compressor buildings, insulating of the main piping was successful (Fig. 8).

In both cases, 10 cm of glasswool was sufficient. We can now design for minimum investment by locating the manifold partly underground and partly above-ground with insulation.

Low Frequency Turbine Outlet Noise. The low frequency turbine outlet noise traveled quite far, through the air and also through the soil. It could be felt more than heard inside the buildings as far as 500 m away. As this concerned six gas turbine compressor units in our station, we had to be careful not to overdesign the silencing so as to keep investment at a reasonable level.

Resonance of the soil and also of roofs and window panes of buildings occurs in the range of 20 to 100 Hz, and we concentrated on dampening this frequency range.

Laboratory tests were set up using sound generators and microphones on a variety of constructions, and we came to a silencer design with a special low frequency section having a small number of very heavy splitters. The result is shown in Figs. 9 and 10.

Vibration Due to Pipe Resonance

On two of our stations, we experienced heavy vibration of the manifold at certain modes of operation. Apart from the frightening noise, it damages field-mounted instrumentation. Also on two aircraft derivative units, the vibration monitors gave a trip signal although these units were not running at all. These stations handle very large quantities of gas, and the manifolds were built with the largest pipe diameters available at the time (36 and 42 in.).

Extensions have been made to these stations, and, although a conservative gas velocity has been used during the design, detailed computer studies now revealed that, at specific tees, high velocities were responsible for the problems.

The troublesome tees had the flow going straight through the run and the branch had a

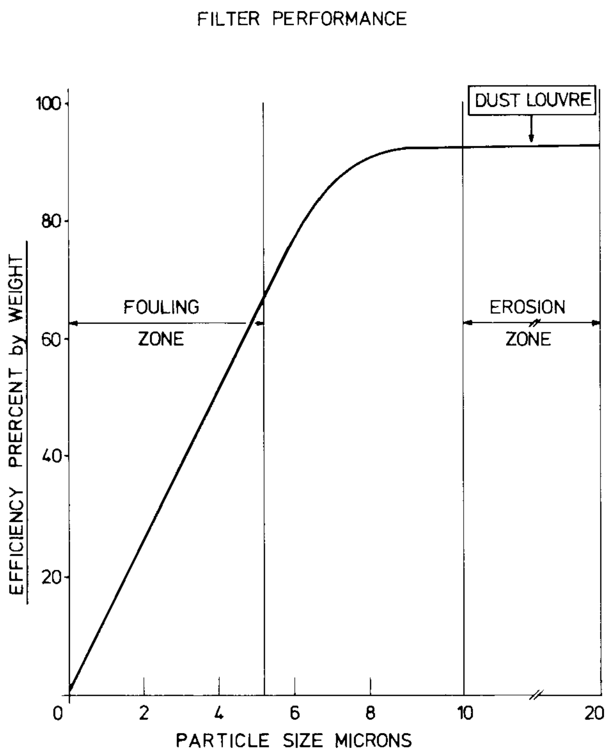

Fig. 5 Filter performance

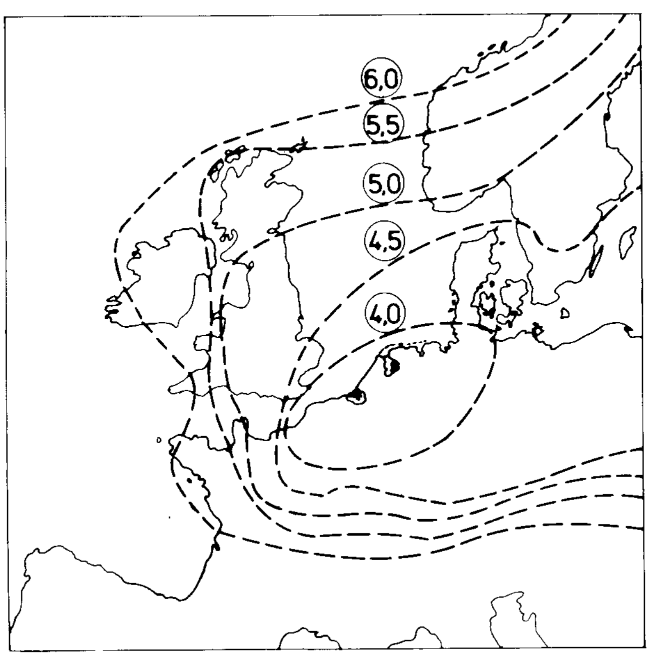

Fig. 6 Average ph of rain water in Europe

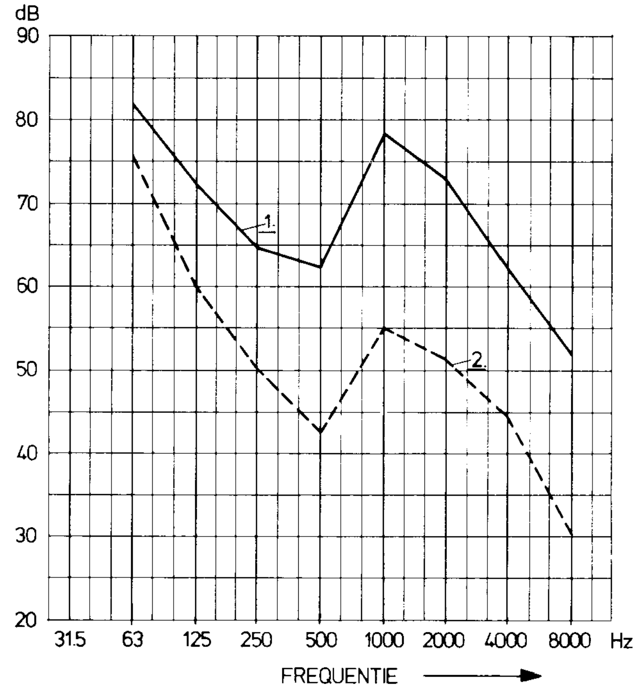

Fig. 7 Graph showing manifold noise

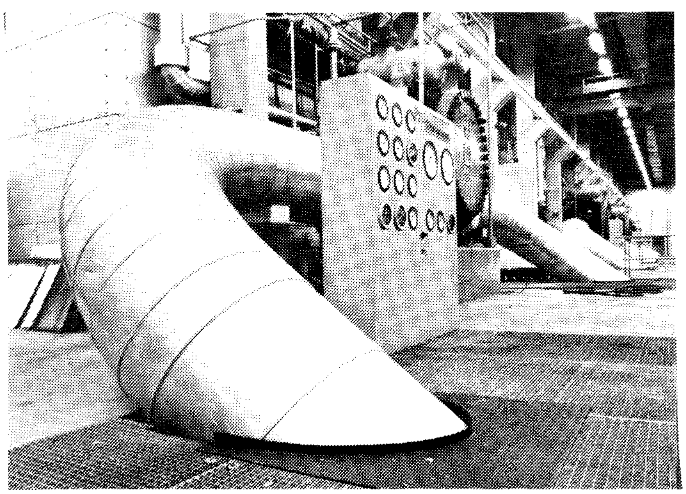

Fig. 8 Insulated piping inside building

closed valve 10- to 20-m distance from the run. As this large size pipe (and valves) is very bulky and expensive, a series of model tests and computer simulations were set up in order to find out how to use very high velocities without problems.

The first rule we found was: Avoid dead ends. Where we cannot avoid these, such as on suction and discharge of standby units on the branch-off from the discharge to the recycle valve of all units, the best position is to put the valve directly against the tee.

By carefully checking the conditions at

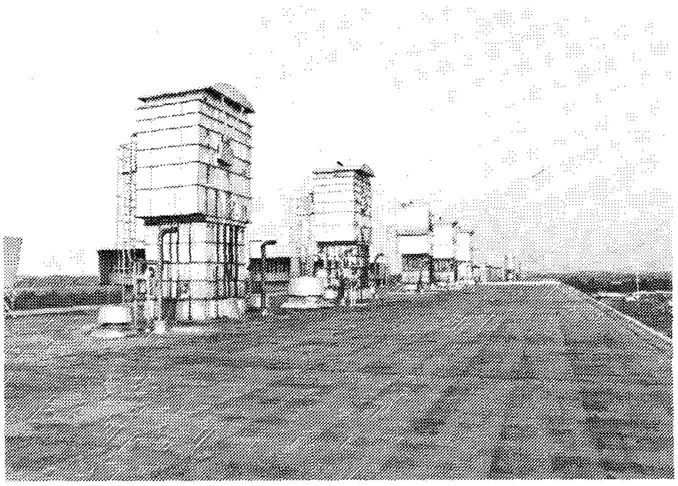

Fig. 9 Outlet silencers with low frequency section

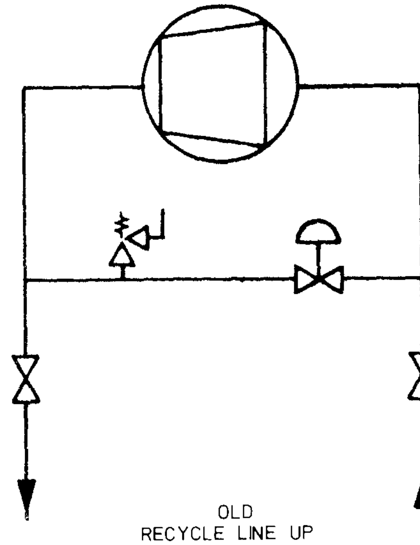

Fig. 11 Schematic of compressor recycle

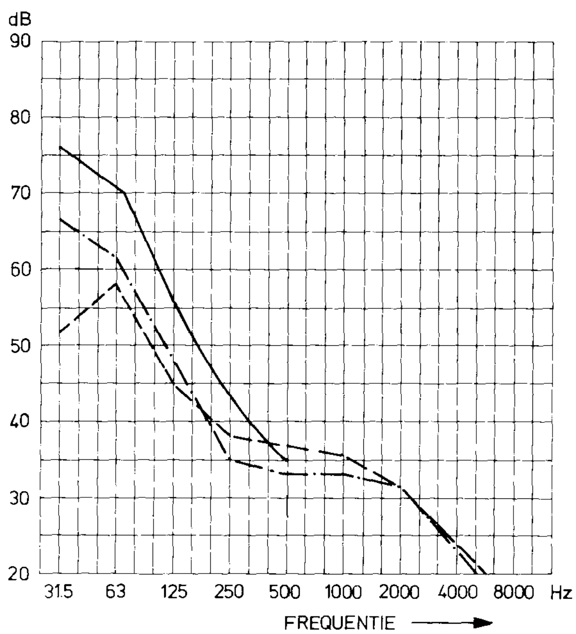

FIG. 10 :
NOISE AT 300 m DISTANCE FROM COMPRESSOR BUILDING
—— ORIGINAL OUTLET SILENCERS
—·— MODIFIED OUTLET SILENCERS
-—- BACKGROUND NOISE

Fig. 10 Graph showing noise of turbine outlet

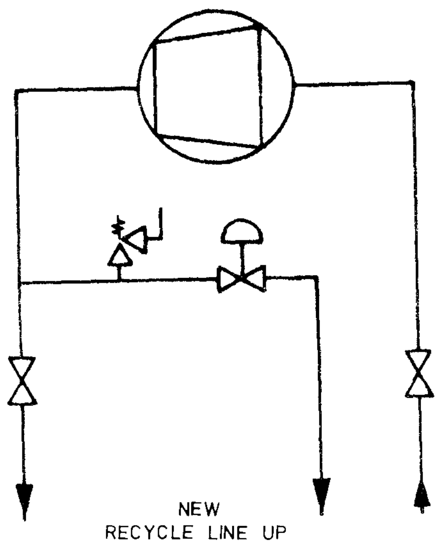

Fig. 12 Schematic of new recycle layout

each tee for all possible modes of operation, we can now design complicated manifolds allowing much higher velocities than before (Figs. 11, 12, and 13).

Industrial Versus Aircraft Derivative Units

We now have in operation 12 heavy industrial gas turbines and 20 aircraft derivative units.

The only difference between the two types is the construction of the gas generator, but this difference is so important that many parameters are affected as shown in the following list:

	Industrial	Aircraft Derivative
Investment	High	Lower
Life (TBO)	80,000 hr	20,000 hr
Cost of overhead	High	Lower
Weight/bulk	High	Lower
Possible modernization	Rare	Frequent
Maintenance cost	Equal	Equal
Efficiency	22-27 percent	25-34 percent
Available size	Up to very large	Up to medium size

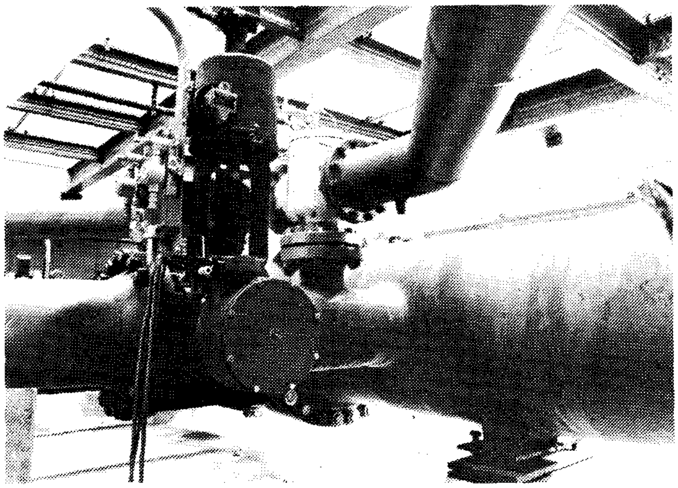

Fig. 13 Good position of recycle valve

Regarding Machine Life. Much depends on the estimated running hours per year. If the estimate is 500 to 2000 hr a year, the industrial could last 50 years or more, and we surely would want to get rid of it in favor of more modern machinery long before that. In this respect, the industrial is a first choice for duties asking 6000 to 8500 hr a year.

In the gap between 2000 and 6000 hr a year, a lot of further thinking and calculating must be done to find the most suitable machine. Should a life between 5 and 10 years be aimed at, an aircraft GG could fill most of this gap.

Should you finally expect to possess a fair number of units, say more than eight, the lower cost of the aircraft derivative could pay for a spare GG and you would still have the advantage of lower weight and bulk, meaning less foundation costs, smaller building, and lighter lifting tools.

Assuming all manufacturers concerned will supply first-class modern equipment, downtime will mainly originate from accidents, such as ingestion of ice or stray metal, liquid slug in gas fuel, sour fuel, or extraordinary air pollution. This is where the spare GG can help to bring you on line again in approximately 8 hr. Should you find that your equipment has to make many more hours a year than you expected, the spare GG can also be used there as a scheduled change-out during overhaul of a worn-out GG.

Maintenance Cost. Excluding serious accidents but including all preventive maintenance and the accompanying repairs on the machine proper, the auxiliaries, and the instrumentation, we found the same figure per HP for industrial as well as for aircraft units. This means a mean figure for 35 gas turbines with between 4000

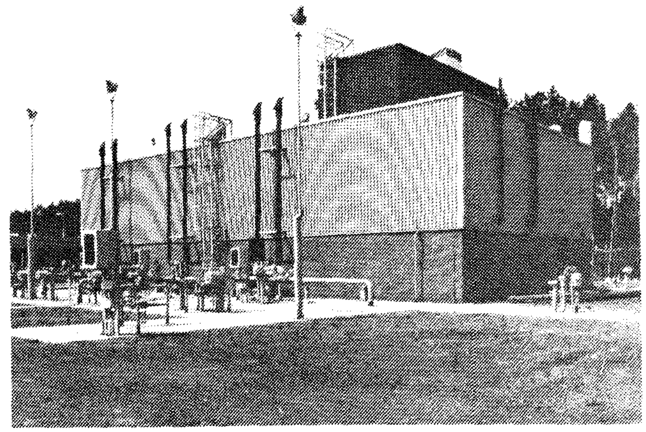

Fig. 14 Schinnen mixing/compressor station: 2- x 2.8-MW, 1- x 0.8-MW solar units

and 25,000 hr each.

GASUNIE COMPRESSOR STATIONS DESIGN FROM 1974-1977

Stations for the High Calorific System

In 1973, as is well known, quite distinctive changes in the future outlook on fossil fuels took place. Already in 1972, for all future compression units, the industrial market price for natural gas in the Dutch market was used. The HC North-South network needed three new stations:

- One at Spijk to compress the Ekofisk gas from 45 to 67 bar
- One at Ravenstein to recompress the HC gas stream consisting of Norwegian and Dutch HC gas
- One at Schinnen to recompress Groningen gas into the HC line to Italy in order to ensure the correct calorific value (mixing station).

The philosophy at these stations, possibly as a reaction to Zweekhorst station, where in retrospect the machines had been chosen too large, was to take comparatively small machines in order to have greater flexibility in operation.

These considerations led, for the first time, to the acquisition of reciprocating units of 4.4-MW output, which with their full-load engine efficiency of 37 percent and good partial load characteristic formed the economic answer to that kind of output. In Spijk, three such units were installed totaling 13.2 MW; in Ravenstein II, five units totaling 22.0 MW.

For the Schinnen mixing station which would run around 4000 hr, the most economical solution was found in acquiring Solar Centaur gas turbine driven units. The main running machine is equipped with a Harrison recuperator, thus giving the gas turbine a full-load efficiency of 32 percent.

The backup machine comprises a standard Centaur set.

Due to the long delay in the start-up of the Ekofisk gas supply, the Schinnen machines were far too large for their mixing duties. Calculations showed that fast installation of a 1-MW Solar Saturn machine in the first year of operation would pay itself off within two years, thanks to the fuel savings to be gained in comparison with running at part load with the 2.8-MW machines (Fig. 14). It was installed within three months after ordering in autumn 1976.

On September 17, 1977, the Ekofisk gas supply started; thus, the Spijk station started its duties. Ravenstein II and Schinnen stations had already taken up their duties in December 1976.

In 1977, it was decided to extend Spijk station with one more 4.4-MW reciprocating engine to be operationable in 1978. The reciprocating machines will be used for the main load. Gas turbine backup units will be installed in 1978 for backup use only. The choice was not finalized at the time of writing this article.

Upgrading Groningen Gas Compressor Station Ommen

Installation of Rolls Royce RB 211 Coberra Units. In 1974, investigations were made to find out whether replacement of one RR-Avon unit by a new RB 211 Coberra high efficiency unit could be economically justified.

A 36-in. Cooper Bessemer compressor could be modified in view of absorbing the additional 8.5 MW.

The efficiency increase from 26 to 33 percent, giving an energy savings of 27 percent, could justify such replacement provided that the machine could run at ~3500 full-load hours.

In Ommen, with the 6- x 11.25-MW + 6- x 11.3-MW + 3- x 23.9-MW units, this proved possible. So in 1974, it was decided to replace machine P103 by a RB 211 gas generator and Cooper Bessemer RT 56 power turbine.

The machine was installed in 1975 and worked well during the 1975 to 1976 and 1976 to 1977 seasons. The measured gas turbine efficiency, including the power turbine, being 33.7 percent!

Further investigations showed that a second RR-Avon could be replaced by a RB 211 set on the same basis. This coincided with the need of installation of a fourth unit in Oldeboorn. In 1977, the complete P102 gas turbine compressor set was transferred to Oldeboorn, and a new RB 211 unit with 42-in. compressor was installed in Ommen in place of P102.

For the transport of HC gas in the winter of 1977 to 1978 and thereafter, the transfer of 3- x 11.25-MW units from the Groningen gas to the HC-system was made in 1976. In order to make up for the loss in output, a third RB 211 unit with 42-in. Cooper Bessemer compressor was ordered in March 1977 for installation in the Ommen III station on the last empty berth. The unit is operational from December 1977.

At the end of 1977, 3- x 19.4-MW RB 211 units with 33.7 percent gas turbine efficiency will be in operation at Ommen. These units will run as first choice so that they make the highest number of hours.

Installation of Regenerator at G.E. frame 5 Gas Turbine. During 1975, Gasunie was approached by Backer & Rueb of Breda, a member of the Rijn-Schelde-Verolme (RSV) group, who had developed a completely new type of regenerator of a size suitable for a frame 5 G.E. gas turbine. (General Electric's manufacturing associate Thomassen at De Steeg being in the same RSV group as Breda.)

The design of this regenerator was based on the best steam boiler design traditions of the Breda firm and aimed to produce a regenerator which can withstand the heat shocks associated with starting and stopping of gas turbines.

The stress calculations were performed in conjunction with the Eindhoven Technical University. Fig. 15 shows the design which is described as follows: The basic design resembles, in fact, the convection bank of a two-drum watertube steam boiler. The compressed air flows from the top to the bottom in the pipes (contrary to the standard waterflow in the boiler) and the gas turbine exhaust gas flows around the air tubes from the bottom to the top. (In the boiler it would be a cross flow pattern.)

The heart of this design, which retains the good flexibility of the two-drum boiler design, is the tube assembly (Fig. 16). Each heat transfer unit placed in the longitudinal section consists of three double-wall identical tubes. The compressed air enters the assembly via a single tube on the top and is then divided in three parts. Each part flows through the inner tube and ends in an assembly chamber where it is united and leaves the chamber via one tube. The hot gas flows in the opposite direction.

The heart of each of the three tube units, which forms a heat transfer unit, is formed by a

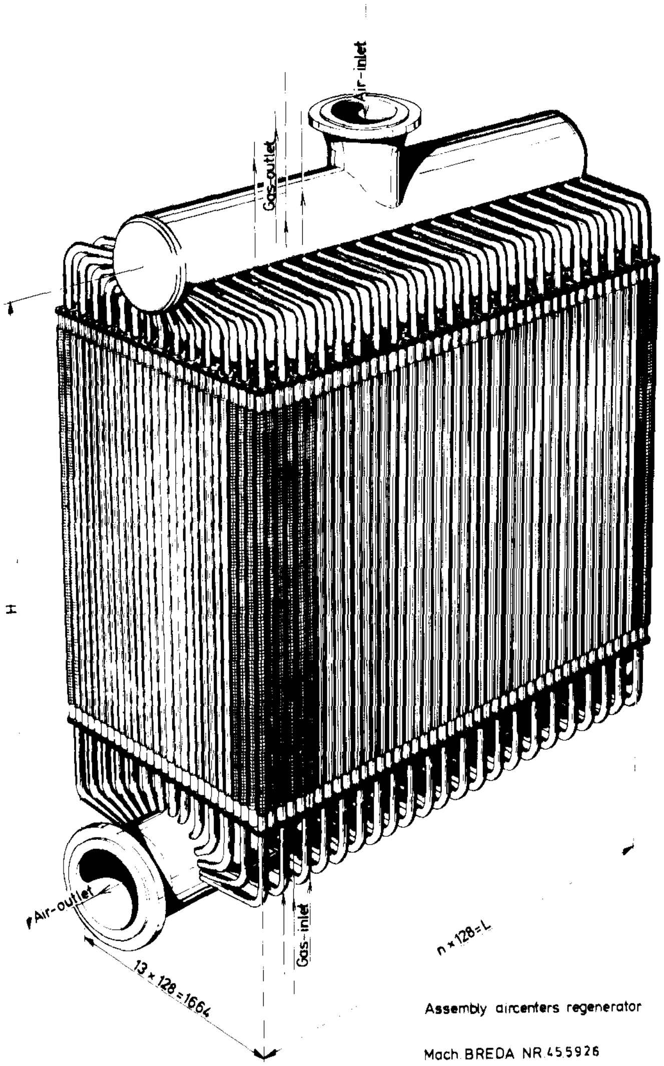

Fig. 15 Breda regenerator as ordered behind a General Electric Frame 5 machine

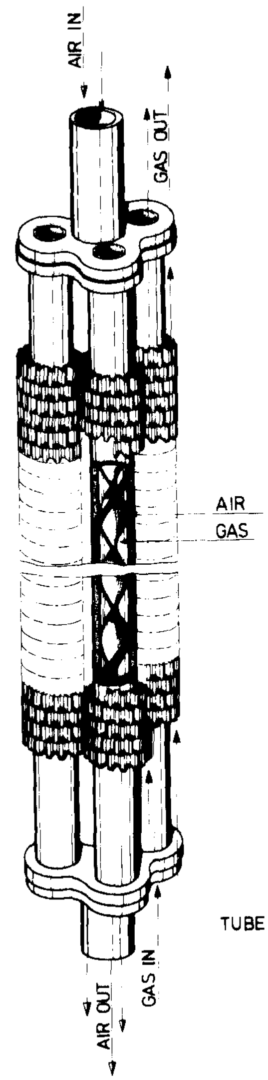

Fig. 16 Breda regenerator heat transfer unit

retarder which fits into the inner tube. The hot gas flows through the inner tube with the retarder. By the pitch of this retarder, the gas speed and thus the heat transfer from hot gas to the wall can be determined to fix the heat absorption rate of the air flow which surrounds the inner tube. To be sure that the outer tube which contains the air flow fits correctly around the inner (gas) tube, a spirally wound wire surrounds the inner tube. The pitch of this wire determines the effective speed of the air, controlling the correct heat transfer. At the outer side of the air tube, fins are placed in the hot gas stream and thus transfer their heat to the tube. The heat transfer from gas to air is thus achieved in both directions in such a way that the gas temperatures at the end of the heat transfer section are equal, and the same is achieved for the air temperatures at the other end.

Thus, seen from top to bottom, every precaution has been taken to prevent unduly material stresses resulting from unwanted temperature differences. Extensive tests of the heat transfer units at the Technical University have supported this thesis. We expect that this design is capable of withstanding the fast temperature changes that will be imposed on it in such a way that no high maintenance costs will arise.

After careful examination, it was found that if the frame 5 machine could be operated at approximately full load during 3500 hr per year at a gas price of 20.5 cents/$m^3{}_0$, it would be economically justified to install such a regenerator. It was ordered in 1976 and will be installed during spring 1978.

The output of the G.E. frame 5 machine is reduced from 23.9 to 22.8 MW.

Its full-load efficiency increased from 27.1 to 33.5 percent.

Resuming: Ommen station will, from spring 1978, onward, consist of:

• Groningen system	1 RR Avon	11.3 MW
	3 RB 211	58.2 MW
	6 GE frame 3	67.5 MW
	3 GE frame 5	71.7 MW
	Total	208.7 MW
• HC-system	3 RR Avons	33.9 MW
		242.6 MW

of which 3 RB 211 and 1 GE frame 5 with regenerator totaling 81 MW will form the heart of the operation, running at 3500 full load hours/year, thus saving $24 \times 10^6 m^3{}_o$ per year, representing at 20.5 cents per m^3, 5×10^6 Dfl per year.

Extension of Ravenstein I station

In the winter of 1978 to 1979, further power increase of Ravenstein I station is necessary. Here the same study pattern was followed. It was carefully examined whether either the new 12-MW RR Spey unit or two Solar Mars units would not be an economical solution. The higher price per kilowatt installed output compared with Avon units should then be compensated by the lower fuel consumption. It proved possible to engage the Spey engine in such a way that the highest energy savings are obtainable, thus justifying its acquisition.

The first Spey with Cooper Bessemer RT 45 power turbine to be installed on the continent was ordered for delivery in autumn 1978 together with a 36-in. Cooper Bessemer compressor.

RESUME

Since 1973, Gasunie was able to follow a program of installation of high efficiency compression equipment leading to the following situation in the winter of 1978 to 1979 of:

• Nine reciprocating compressors	39.6 MW
• Three RB 211 units	58.2 MW
• One G.E. frame size 5 unit with recuperator	22.8 MW
• One Solar Centaur unit with recuperator	2.6 MW
• One RR Spey unit	12.1 MW
	135.3 MW

Out of a total of 541.8 MW installed output.

BIBLIOGRAPHY

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