MECHANICAL VAPOUR RECOMPRESSION AT PONGOLA

By G. N. ALLAN
Sugar Milling Research Institute

M. R. KEDIAN
*Sugar Milling Research Institute**

D. E. TRATTLES
C. G. Smith Sugar Limited, Technical Services Division

Abstract

Four performance test runs were carried out on a 31 t h⁻¹ mechanical vapour recompression unit at Pongola, compressing vapour from 145 kPa abs to 196 kPa abs. The test results are examined and the overall effect on the steam balance is assessed. A saving of approximately 6 t h^{-1} of high pressure steam is achieved reducing extraneous fuel costs by about R100 000 per season. Capital cost of the installation is given.

Introduction

During the 1981/82 season a mechanical vapour recompression (MVR) system was installed at PG. A new evaporator vessel was installed specifically for this purpose and the compressor was set up to compress the Vapour One (V1) coming out of the vessel and to reintroduce it to the calandria of the vessel at steam conditions equivalent to that of exhaust steam.

The only difference between the vapour entering the calandria and the vapour leaving the vessel (e.g. exhaust and V1) is a small difference in pressure and temperature (from 196 kPa/120°C to 146 kPa/111,5°C). Mechanical vapour recompression takes the vapour from the vessel, compresses it to increase its pressure (and consequently the temperature) and sends it back to the calandria of the vessel where it can release not only the extra energy stored in it by virtue of being compressed, but it also condenses releasing latent heat. In other words, by performing a small amount of work on the vapour leaving the vessel, its latent heat can be made available to the evaporator.

Because PG operates a backend refinery (and diffuser) the factory uses extraneous fuel to make up the deficit between process demand and the steam provided by bagasse generation only. This quantity of fuel has been of the order of 600 tons of coal per week. Steam demand at a factory can best be regulated by vapour bleeding at the evaporator, but in an existing plant care has to be taken with the process steam/prime mover ratio so that there is at all times a minimum of flow through the let-down line and no surplus exhaust steam is blown off through safety valves.

It was considered that an extra 60 t h^{-1} of V1 could be bled from the evaporator, but initially, in order to prove the feasibility of the MVR concept, only 30 t h^{-1} would be bled.

In order to achieve 30 t h^{-1} evaporation from the first effect of the quintuple effect evaporator an equal amount of exhaust steam must be available and this steam could be provided in three ways:

(a) Direct let-down through a reducing valve to the vessel i.e. a ratio of 1 kg high pressure to provide 1 kg of exhaust steam (desuperheating excluded).

(b) Thermo-compression (TC), where 30 t h^{-1} of V1 would require 15 t h^{-1} of high pressure steam to provide 45 t h^{-1} of exhaust, an entrainment ratio of 1:2,0[1].

(c) Mechanical vapour recompression (MVR), where 9,4 t h^{-1} of high pressure steam would compress 30 t h^{-1} of V1 to exhaust condition, ratio of 1:3,2.

As the let-down margin at PG was not generous the MVR was selected as it would still leave a margin of safety to avoid excess exhaust steam. The overall high pressure steam demand from the boilers would be roughly the same for both TC and MVR. (In other applications the MVR has advantages in higher compression ratios, possible use of national electricity supply for electric drive etc).

The Sugar Milling Research Institute (SMRI) was invited to be involved in the commissioning of the compressor and to do performance tests on the machine once it was in stable operation.

Installation Details

The compressor installed was built by Bryan Donkin Ltd and is an RV 60 single inlet centrifugal compressor fitted with an inlet guide vane assembly. The setting of the guide vane angle is operated by a pneumatic cylinder and is used to control machine throughput.

The specifications of the compressor are given below (pressures are given in kPa abs).

Flow rate	31,13 t h^{-1}
Inlet flow	10,27 m^3 s^{-1}
Inlet pressure	146,2 kPa
Static outlet pressure after taper pipe	196,2 kPa
Total pressure rise	50,5 kPa
Total inlet temperature	110,6°C
Compressor speed	9 100 rpm
Power absorbed at shaft end	622,5 kW
Gas inlet density	0,8416 kg m^{-3}
Pressure losses allowed	0,5 kPa
Gas analysis	Dry saturated steam

Assuming a polytropic compression efficiency of 77% calculations show that the outlet temperature at 196 kPa should be 145,5°C.

A sectional drawing of the compressor is given in Figure 1.

The compressor is driven by an Elliot Type CYR steam turbine which transmits 688 kW through a gearbox unit designed to increase turbine speed from 5 000 to 9 100 rpm. Steam conditions at the turbine were measured at a nominal 3,0 MPa and 400°C. Figure 2 shows a flow diagram for the MVR system of evaporator vessel and compressor.

Control Strategy and Instrumentation

The vapour compressor takes suction off the V1 main and discharges into the calandria of the dedicated vessel of 2 400 m^2 heating surface. As this vessel is capable of condensing 60 t h^{-1} of steam, an exhaust range make-up is provided to allow full utilisation of its area.

As the function of the vapour compressor is to prevent marginal coal burning it was decided to set its operating point by flow control and allow the exhaust make-up to fluctuate as required to maintain calandria pressure as described below.

An Annubar flow sensor is positioned in the compressor discharge line, and its signal is processed through a density

* Present Address: Mangosutho Technikon

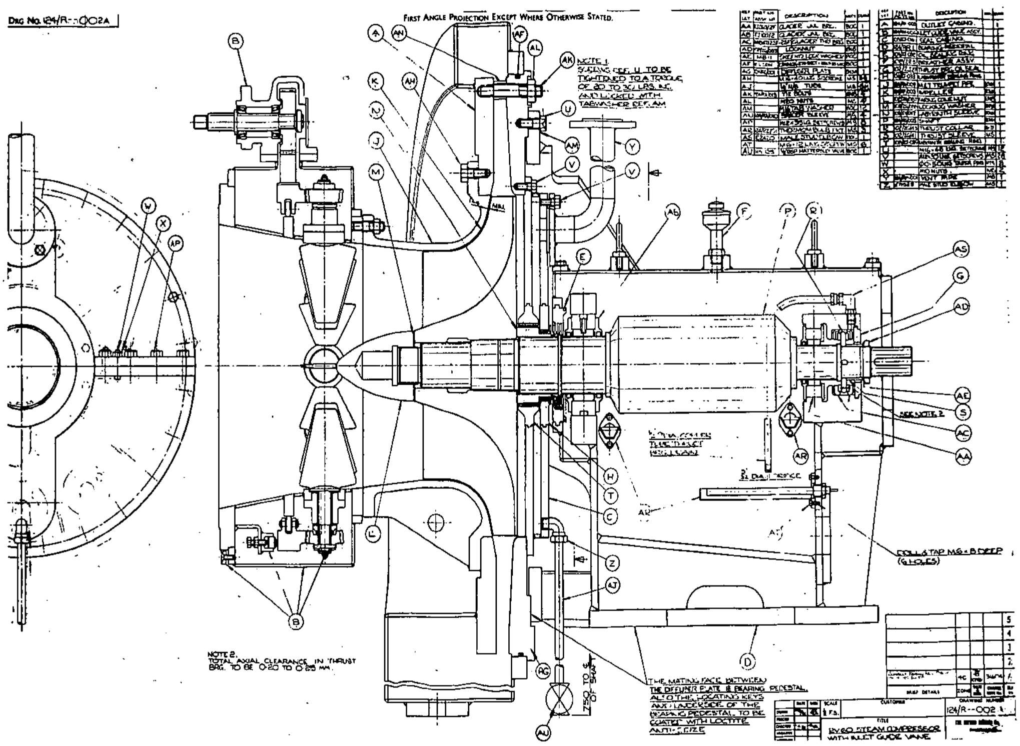

FIGURE 1

compensating multiplier unit and square root extractor. This is in turn fed to a two term controller which feeds back to a Kent power cylinder operating the suction guide vanes of the compressor.

The calandria pressure (and hence compressor back pressure) is regulated by a two term controller that feeds back to a 400 mm butterfly valve in the exhaust make-up line.

Surge control is effected very crudely by maintaining a preset minimum compressor throughput under all conditions. The signal to the compressor flow controller is paralleled to an auto/manual station that is set to operate in reverse mode and biased to suppress output above the desired set point chosen from the characteristic curves, in this case 25 t h^{-1}. The relief mechanism comprises a 150 mm Fisher quick opening valve to vent excess vapour, with its positioner ranged over a narrow input span to provide full actuation for a small drop in operating flow below the set point.

A non-return valve is to be fitted to prevent reverse flow in the event of surge occurring. This will be a 500 mm fail open butterfly, with its positioner ranged to the same input span as the surge vent valve and connected to the same controller output signal to provide simultaneous, but opposite, operation. This additional valve will also allow independent start-up and shutdown of the vapour compressor while the evaporator train is in operation.

For additional protection when the evaporator train is starved of juice, with the consequent upset in the vapour ranges, a low V1 (inlet) pressure trip has been installed with a setting of 120 kPa.

Other panel mounted instrumentation includes pressure indicators for compressor suction and discharge, a triple pen recorder for compressor flow, clear juice feed rate to the evaporator train and turbine nozzle box pressure, a digital indicator for juice and steam temperatures, and a digital indicator for compressor speed.

The compressor itself has a local panel with oil supply pressure gauges and temperature indicators fitted with mercury switches for connection to the alarm circuit.

There are three differential oil pressure switches mounted on

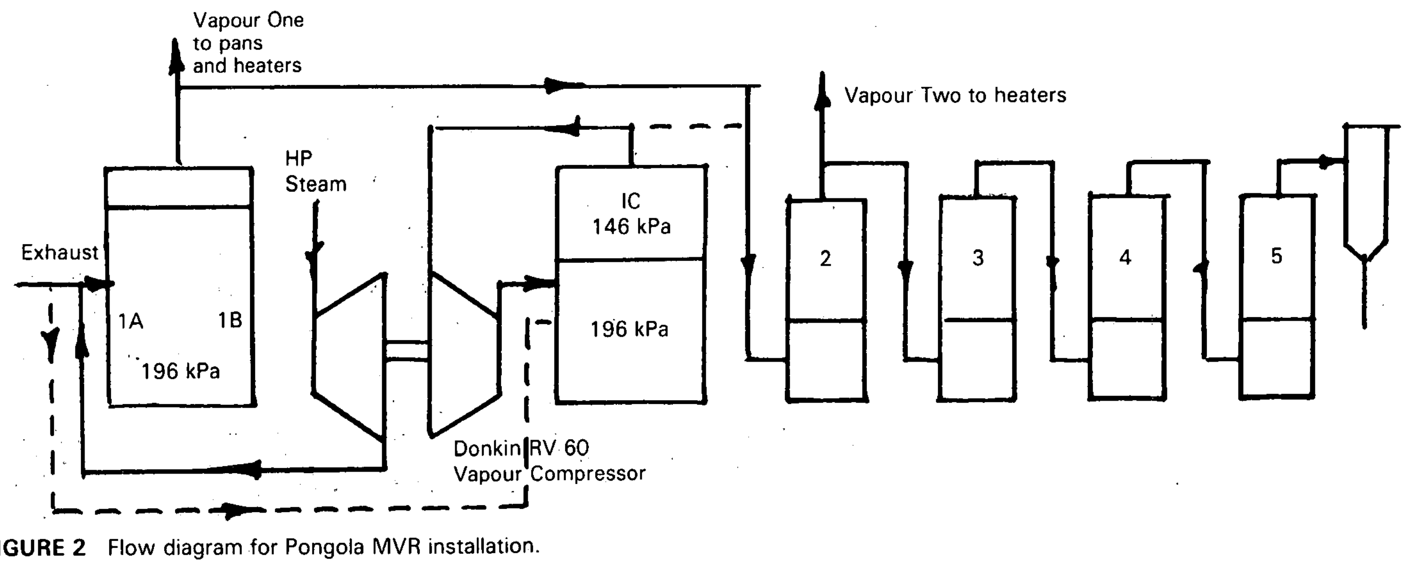

FIGURE 2 Flow diagram for Pongola MVR installation.

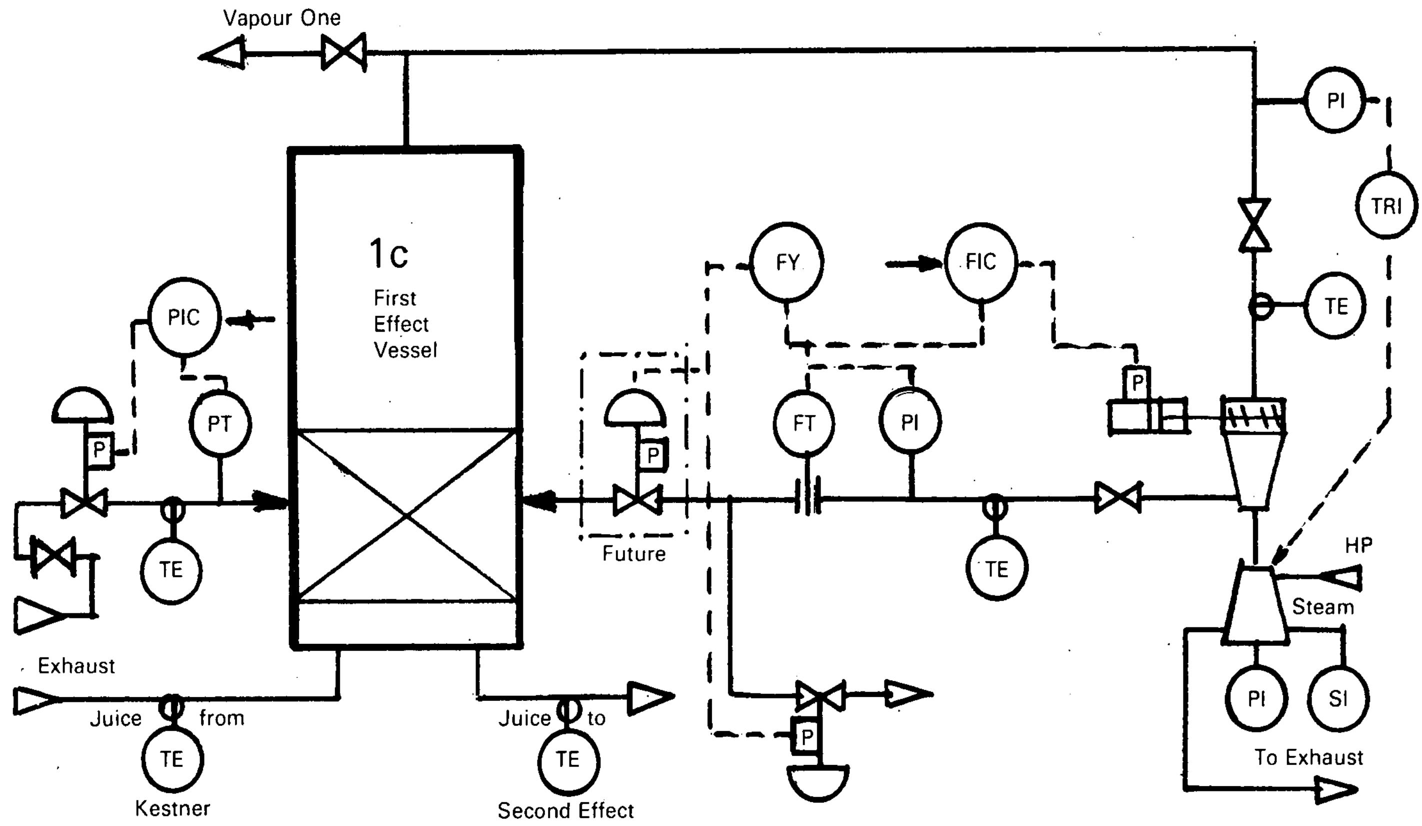

FIGURE 3 Pongola MVR instrumentation diagram.

the base frame to control the auxiliary oil pump and provide a trip signal in the event of complete loss of pressure.

All process instrumentation is Fisher AC[2] with the exception of the Eurotherm digital indicators.

The control system is shown in Figure 3.

Results

Four series of test runs were performed on the MVR. The machine was run at two speed settings viz. 9 000 and 8 080 rpm and at each speed the vane setting was reduced from 0° to 60° in either four or five steps and then restored to 0° in

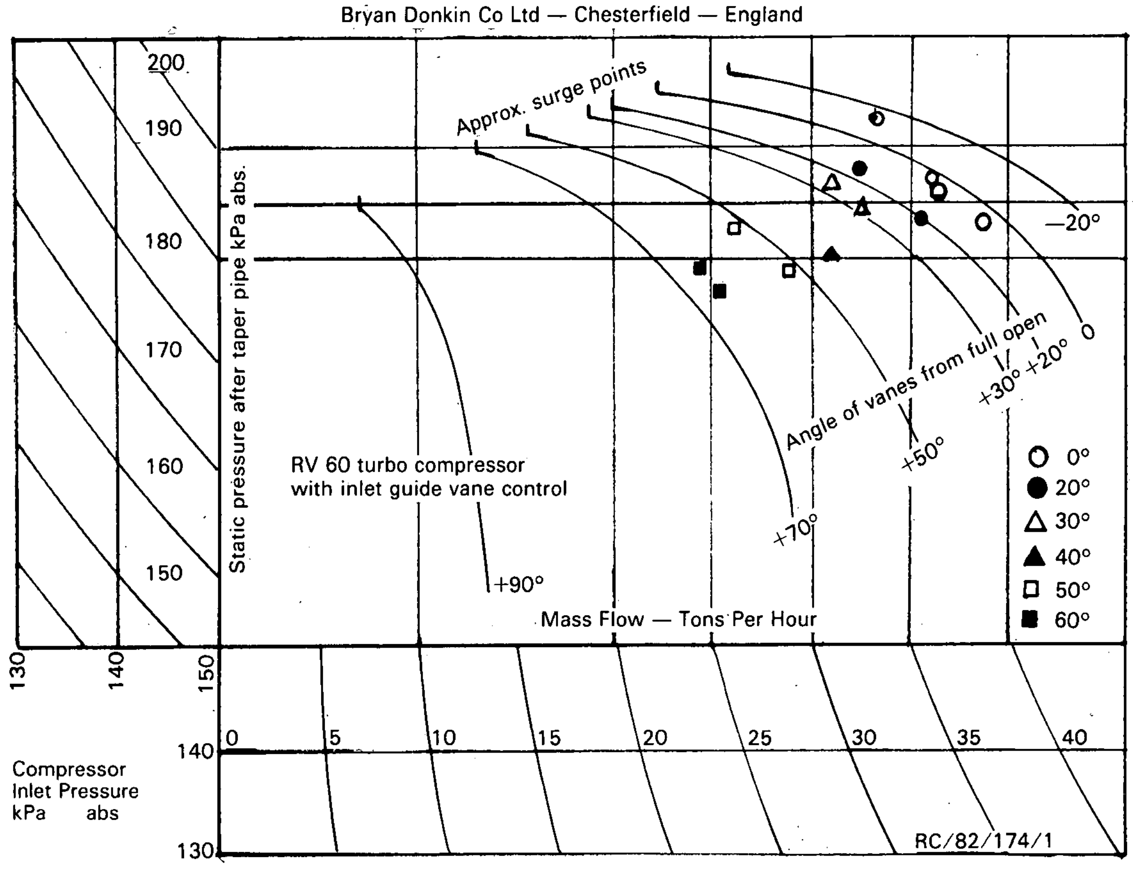

FIGURE 4 Test results on manufacturers performance curves.

similar fashion. Altogether some 59 test runs were made. Overall performance of the unit has been trouble free with a full capacity test average throughput of 30 250 kg h^{-1}. The pressure levels are below specification, mainly because the level of V1 pressure at PG has been nearer 137 kPa rather than the 146 kPa specified.

The vapour outlet flow was measured by means of the Annubar flow recorder previously described.

In order to ensure that these readings were accurate, a proportional head weir tank was installed to measure the condensate flow from the evaporator vessel calandria supplied by the vapour compressor. The weir tank was to be used as a check on malfunctioning of the Annubar and indeed proved very useful in Test No. 4 when flow measuring problems did indeed arise. Corrections were made for water density decrease as the tank was calculated for, and calibrated at, ambient conditions and also for the flash of condensate from 120/116° to ambient condition.

Full Capacity Tests

Thirteen vapour flow measurements were taken from four test series at 0° vane setting and 9 000 rpm to obtain an average full capacity rating (at various pressure levels). The average rating was 30 250 kg h^{-1} with a range of 25 200 to 35 000 kg h^{-1}.

A typical test series at 9 000 rpm with various vane settings is illustrated in Figure 4 showing that the MVR unit operated closely to its performance curves.

Throughput Measurements with Vane and Speed Control

The tests were run with the machine at two different speeds viz. 9 000 rpm and 8 080 rpm with approximate vane settings of 0°, 20°, 30°, 50° and 60° (See Figure 5).

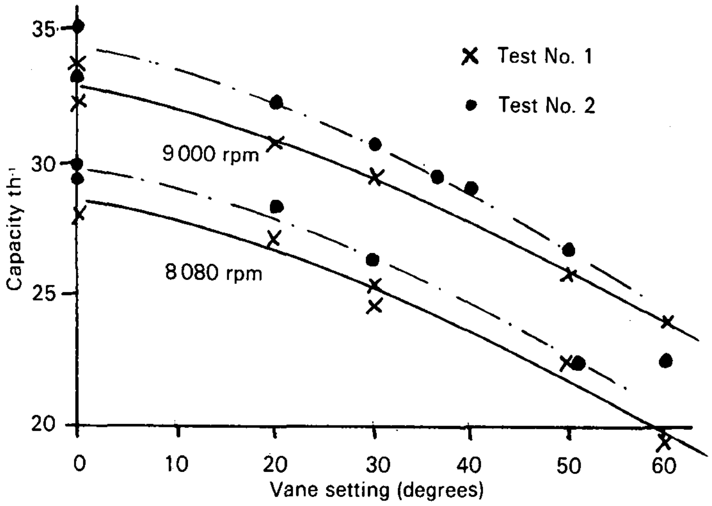

FIGURE 5 Compressor capacity versus speed and vane setting.

However it is unlikely that both control modes would be used simultaneously to determine output.

Vane control is the obvious means of control for an MVR unit if the steam balance position or the low cost of electricity indicates an electric motor drive. Vane control is regarded as an efficient throughput controller and is much more efficient than a butterfly valve type suction control as it does not lose energy. An illustration of the vane control system is shown on the machine detail drawing in Figure 1.

Average Pressure and Temperature Differentials

A summary of the average values of inlet and outlet pressures with corresponding saturation temperature increases is shown in Table 1. These values were obtained with the machine at 9 000 rpm and 0° vane setting during Tests 1 to 4.

The temperature differential for Test 3 was higher than previously observed and subsequent inspection of the vessel indicated that considerable scaling had taken place due to the large heating surface (2 400 m^2).

During Test 4 the V1 pressure setting was increased to enable the specified 146 kPa suction pressure to be reached, but the actual pressure remained higher than the machine specification throughout the test period.

The temperature differential (saturated) varied between 7,8 and 9,6°C with an overall average of 8,4°C.

Turbine Input Power

Test No. 4 was arranged to measure both turbine input power and compressor output.

Steam flow was measured to the turbine by a standard orifice plate designed to BS 1042 and temperature and pressure of inlet and exhaust steam were measured. On this test the pressure gauge on the steam chest failed but in the previous test series this pressure was observed to be about 2,1 MPa.

	°C	kPa	kJ kg^{-1}
Turbine Inlet Steam	377	2 600	3185,7
Turbine Outlet Steam	225	212	2920,7
			265,0

S = 8 417 kg h^{-1} (= 2,338 kg s^{-1})
Power = 265 × 2,338 = 619,57 kW (say 620 kW)

On the turbine side the manufacturers ran a computer check on the CYR type Elliot steam turbine taking the data supplied by us i.e. 620 kW inlet steam conditions at 2 600 kPa, 377°C and running at 9 000 rpm. The corresponding steam flow and exhaust condition calculation matched our measured figures reasonably well.

	Measured	Elliot Computed Values (CYR)
Steam flow kg h^{-1}	8 417	8 526
Exhaust °C	225	227

Compressor Energy Output

The Annubar flow reading was confirmed by the weir condensate flow for this power measurement test. The machine was on 0° vane setting and the inlet vapour was assumed to be at dry saturated condition.

Vane setting	0°	
Condensate	27 000	kg h^{-1}
Inlet temperature	112	°C
Outlet temperature	144	°C
Inlet pressure	159	kPa
Total heat at inlet (dry sat.)	2 696,2	kJ kg^{-1}
Outlet pressure	214	kPa
Total heat at outlet	2 755,2	kJ kg^{-1}
Heat increase	59	kJ kg^{-1}

H = 2 755,2 − 2 269,2 = 59 kJ kg^{-1}
Vapour flow = 27 000 kg h^{-1} (= 7,5 kg s^{-1})
Output energy = 59 × 7,5 = 443 kW

TABLE 1

Average Values of Tests Run at 9 000 rpm and 0° Vane Setting

Test No.	Run No.	Av. Inlet Pressure	Av. Outlet Pressure	Pressure Increase	Av. Outlet Temperature	Sat. Inlet Temperature	Sat. Outlet Temperature	Temperature Increase
1	1, 2, 7, 23, 28	134	176,0	42	138,5	108,0	116,2	8,2
2	1, 8	138	179,0	41	139,0	108,9	116,7	7,8
3	1, 2, 3, 4, 8	139	190,4	51,4	138,6	109,1	118,7	9,6
4	1, 5, 8	157	212,7	55,7	143,0	112,7	121,4	8,7

Comparing the measured turbine input power value of 620 kW with the above value of 443 kW there is a wide discrepancy to be explained.

On the turbine side a mechanical loss of about 20 kW can be allowed leaving 600 kW in gas compression at the gearbox input shaft. It will be seen that the turbine power still exceeds that of the compressor by approximately 160 kW and consideration must be given to the reasons for this on the compressor side.

Effect of Moisture in Inlet Vapour

We suspect that the initial dry saturated steam conditions assumed for the inlet vapour are not in fact correct and an estimate of the effect of moisture on these measured conditions will be calculated.

Taking the value of the above test series, (Test No 4 Run 1) the inlet and outlet conditions indicated by gauges are as follows:

	Press. kPa	Temp (°C)
Inlet	159	112
Outlet	214	144

Dry saturated temperatures at these pressures are given as 113,1 and 122,4°C respectively. The former gauge error is due to a digital readout (in whole degrees only) which is not serious if the pressure is known. The latter value will indicate the degree of superheat in the compressed vapour (144-122,4 = 21,6°C).

Two considerations are required:

(1) Dry saturated vapour compressed to 214 kPa gives a final steam temperature of 148,4°C. (Compare the measured value of 144°C).

(2) Assuming an inlet dryness fraction of 0,966 compressed to 214 kPa we get a final steam temperature of 143,8°C. The total heat at inlet condition now drops to 2 686,2 kJ kg^{-1} and the polytropic heat increase goes to 68,6 kJ kg^{-1} compared with the original estimate of 59 kJ kg^{-1}. The theoretical power required becomes 515 kW (for 27 000 kg h^{-1}).

Taking into account a mechanical efficiency figure of 97% for both the compressor and gearbox this gives a shaft input power of:

$$\frac{515}{0{,}97 \times 0{,}97} = 547 \text{ kW}$$

These test results indicate turbine power of 600 kW against a compressor output of 550 kW, a discrepancy of about 8%.

It can therefore be seen that an assumed moisture content of inlet vapour of only 0,4% makes a difference of 2 695,1 - 2 686,2 = 9 kJ kg^{-1} in the inlet total heat value. This in itself is negligible but when compared with an isentropic heat increase of the order of 60 kJ kg^{-1} it becomes nearly 15% of the total.

There is some corroborative evidence to support this theory. The expansion joint in the suction line has been removed this off-crop to enable limited inspection of the MVR compressor. This has revealed some corrosion pitting in the taper section holding the regulating vanes and in the inlet section, some circumferential corrosion has taken place in line with the leading edge of the impeller. The impeller blades (at inlet) are rough to the touch and some impingement has occurred here. It is apparent that moisture droplets are present in the incoming vapour. This limited access does not enable the blade tips to be examined and it is hoped that no erosion is present as a balancing problem could arise at the high rotational speed used.

The by-passing of some of the compressed vapour into the inlet stream, or direct heating, is recommended by manufacturers as desirable to avoid droplet impingement on the rotor.[2] Under typical operating conditions of say 135 kPa (108°) inlet at 0,996 dryness fraction and 180 kPa (137,5°C) outlet, the effect of adding heat to the inlet stream may be examined.

To heat the inlet vapour to dry saturation condition a fraction F of the outlet steam must be by-passed.

$$(F \times 2\,745{,}5) + ((1 - F)\,2\,679{,}2) = 1 \times 2\,688$$
$$F = 0{,}13$$

i.e. 13% of the stream must be by-passed.

However as the inlet stream becomes dry saturated the outlet condition moves to 142°C and 2 755 kJ kg^{-1}. The by-passed outlet stream will be reduced as there will be a greater heat drop to saturation conditions:

$$(F \times 2\,755) + ((1 - F)\,2\,679{,}2) = 1 \times 2\,688$$
$$F = 0{,}116$$

and only 11,6% will be required. If 1°C superheat is required in the inlet stream then the quantity increases to 14%, 3°C has been mentioned as a desirable margin but in this case would not be acceptable.

Effect of MVR on PG Steam Balance

Two evaporator balances were calculated based on typical PG clear juice flow and a refinery melt of more than 90% of its raw sugars. One, using a slightly optimistic MVR output of 32 000 kg h^{-1} is shown in Table 2. Using the MVR and the new evaporator vessel, the exhaust steam supply to the first effect is reduced by 126 487 - 120 087 = 6 400 kg h^{-1} and the effect this has on high pressure steam production is a nett saving of 5,5 t h^{-1} steam or 0,71 t h^{-1} coal.

Working on a 40 week season and an overall time efficiency of 0,75, 3 580 tons of coal could be saved. At a landed cost at PG of R27 per ton a saving of R96 000 per season could be made in the cost of extraneous fuel.

MVR Costs

Capital Cost

The PG installation was originally designed to work with two MVR's of 30 t h^{-1} capacity working around a dedicated evaporator vessel. This vessel is therefore twice the size required for the single unit described. It is of course also understood that the compressor could readily be used in conjunction with an existing installation and in fact could be used between two pressure ranges without a dedicated vessel, as is proposed for Sezela.

An MVR installation does not necessarily require more heating surface as less evaporation would be required at the tail of the evaporator to obtain the required output brix and a reallocation of existing heating surface could be arranged.

Total installed cost was R600 000. Some of the major capital cost items are given below:

Compressor	R126 000
Turbine	R 40 000
Instrumentation	R 22 000
Vessel	R104 000
Tubes	R 89 00
Demister	R 9 000

The individual items are listed to give some idea of the main component costs, the balance being piping, erection, contract fees etc.

Maintenance Cost

The first year of operation of the MVR unit involved some operational problems of surging, reverse rotation, oil cooling difficulties etc., and the cost of repairs should not be included as a fair reflection of machine maintenance costs.

The second season has been trouble free and maintenance costs which are not to hand should be no more than is usually involved with a typical mill drive turbine and gearbox of 600 kW capacity. A problem associated with shaft sealing leakage

TABLE 2

PG Evaporator Steam Distribution

Pans, Heater, MVR V(1) = 122534 kg h^{-1} Heater V(2) = 20488 kg h^{-1}
Vapour compression = 32000 kg h^{-1} V1 to exhaust

Evaporator effects	First	Second	Third	Fourth	Fifth	Condenser
Calandria press kPa	180,0	140,0	109,0	78,0	47,0	16,0
Calandria temperature °C	117,1	109,6	102,5	93,3	80,3	55,5
Boiling point elevation °C *	0,0	0,5	1,5	2,0	3,5	
Juice temperature °C	109,6	103,0	94,8	82,3	59,0	
Juice flow kg h^{-1}	251734,8	99647,0	70093,7	61028,5	51693,2	42898,0
Brix percent juice	11,928	30,134	42,840	49,204	57,788	70,000
Evaporation kg h^{-1}	152087,8	29553,2	9065,2	9065,2	9065,2	
H.T.C. kW m^{-2} °C^{-1} *	2,400	2,000	1,700	1,300	0,650	
Estimated Δ T °C	7,4	6,6	7,6	11,0	21,3	
Heating surface m^2	5232,7	1381,9	437,1	400,3	419,1	
Spec. Evap. kg m^{-2}	29,0	21,3	30,7	22,6	21,6	

Overall (ΔT − BPE)°C	=	54,04°C	Exhaust steam to first vessel	=	120087 kg h^{-1}
Total evaporation	=	208836,8 kg	Exhaust steam to C. J. heater	=	3815 kg h^{-1}
Total atea	=	7871,3 m^2	**Total exhaust steam** (factor 1.1)	=	136293 kg h^{-1}
Specific evaporation	=	26,5 kg m^{-2}	Exhaust steam kg per kg brix in mixed juice	=	4,53
Area/TCH	=	40,1 m^2	Exhaust steam percent cane	=	59,5

* Assumed values.

causing some condensate contamination of the oil was solved by putting positive air pressure on to the labyrinth seal.

Conclusions

Now that these tests have been completed it has become apparent that the measuring techniques for future tests will have to be far more accurate than those used to date. Additional tests, e.g. dryness fraction of inlet vapour will have to be carried out to obtain a better assessment of the energy balance across the system.

Overall, the MVR measured output matches well with the detailed performance sheets supplied by Bryan Donkin Ltd.

C.G. Smith Sugar Limited Technical Services Division (TSD) is to be commended for a novel solution to a problem of extraneous fuel usage. As far as is known, this MVR unit is the second one installed in South Africa in any industry.

Acknowledgements

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