INDEN 5350 - CREATIVE COMPONENT
FEASIBILITY STUDY OF GAS ENGINE DRIVEN AIR COMPRESSORS

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ABSTRACT

The objective of this research is to conduct a feasibility study of Gas Engine Driven Air Compressors versus Electric Motor Driven Air Compressors. The methodology adopted in achieving this objective was to develop a linear equation, which was used to develop payback graphs. These graphs were used for the feasibility study. It was concluded that investing in engine driven air compressors is feasible only for a limited combinations of rate schedules and operating hours.

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CF - Conversion Factor.

EC - Electrical consumption in kWh.

EM - Capacity of electric motor driven air compressor, in HP.

E_{EM} - Electric motor driven air compressor efficiency (assumed 90%).

E_{GE} - Gas engine driven air compressor efficiency (refer section 2.3.1).

E_H - Heater efficiency (assumed 80%).

EMDC - Electric Motor Driven Air Compressor.

GC - Gas consumption in Btu/hr.

GE - Capacity of gas engine driven air compressor, in HP.

GEDC - Gas Engine Driven Air Compressor.

HF - Waste heat recovery factor (assumed 30%).

IIC - Incremental installation cost: This is the incremental cost to buy a GEDC instead of EMDC.

IICw - Incremental installation cost: This is the incremental cost to buy a GEDC instead of EMDC with the waste heat recovery system.

IICr - Incremental installation cost: This is the incremental cost to retrofit a EMDC with a GEDC (after making adjustments for the salvage value of the existing EMDC).

IICrw - Incremental installation cost: This is the incremental cost to retrofit a EMDC with a GEDC with waste heat recovery system (after making adjustments for the salvage value of the existing EMDC).

LF - As the compressors do not operate all the time at full load, a load factor is used, which is approximately equal to the average loading of the compressor.

MGE - Maintenance cost of gas engine.

n - Payback of the investment in years.

NS - Net savings.

SWHR - Savings from waste heat recovery.

t - Operating hours of the compressor per year.

X - Weighted average cost of electricity. This includes the demand ($/kW-yr) charge and energy ($/kWh) charge.

Y - Cost of natural gas per MCF.

CHAPTER 1

INTRODUCTION

No one knows exactly when air was first used as a power source. Today, air has been harnessed in almost every phase of industrial and commercial activity. One of the most important features of compressed air as an industrial tool is its flexibility. It can do many jobs well, and new uses are being constantly discovered [1].

The uses of compressed air can be broken down into two main categories: power service and process service. Power service may be defined as those applications in which air is used either to move something or to exert a force. Examples are operating cylinders, pneumatic tools, clamping devices, air lift and pneumatic conveyors. Process service is defined as those applications in which air or other gas enters into a process itself. Examples are combustion, liquefaction and separation of gas mixtures into components, hydrogenation of oils and other chemical reactions [1].

Compressed air generation, commonly referred to as the fourth utility of industry, can be one of the most energy consuming activities in an industrial operation. Studies show in a single year's operation, an electric compressor costs more to operate than it originally cost to buy. In fact, as much as 30% of all electricity used in industry is consumed producing compressed air ([8], [9]).

Many factors enter into the selection of a prime mover for air compressors. The price of the energy and its availability are the two most important factors. The other factors include first cost, maintenance cost, and hours of operation. Generally, one has a tendency to go in for a motor-driven installation, but it is best not to arrive at a hasty decision. Perhaps a better plan lies in first considering the other kinds of motive power. Other ways of providing drive to the air compressor are: 1) steam -either direct-connected steam engine or turbine, 2) oil -either diesel or gasoline engine, and 3) gas -either natural or manufactured-gas engine [3].

A traditional way of providing drive to the air compressor has been by an electric motor. But, now there may be a shift to gas engine drives. This is due to the fact that the operating cost of electric motor driven air compressor (EMDC) can be more than a gas engine driven air compressor (GEDC).

The concept of gas engines is not new. But, the extension of this concept to much smaller applications came about only in the recent past. Gas engine driven chillers are also popular. Vigorous research is being carried out by research institutes to come up with more economical and efficient gas engines. For example, a gas engine driving a 110 HP compressor is 30.67% efficient when the compressor is operating at full load. But, the efficiency increases to 31.87% as the compressor load drops to 70%.

CHAPTER 2

SYSTEMS PERFORMANCE

2.1 Compressor Performance

Air is a mixture of gases containing, by volume, about 78% nitrogen, 21% oxygen, and 1% other rare gases including argon, neon, crypton, xenon, etc. When air is placed under pressure, it is compressed, thereby developing a higher level of pressure than the normal atmospheric air. And, since it is characteristic of air to want to return to its normal state, this air under pressure, if controlled can be made to work. In this section some relevant definitions, types of compressors, operating characteristics, factors affecting performance of compressors and their industrial applications are briefly discussed [5].

2.1.1 <u>Definitions:</u> Some relevant definitions are as follows [5]:

<u>Actual capacity</u> of a compressor is the quantity of air or gas which the unit actually takes in, compresses, and delivers to the discharge line.

<u>Volumetric efficiency</u> is the ratio of the actual capacity of the compressor to displacement and is expressed in percent.

<u>Compression efficiency</u> is the ratio of the theoretical horsepower to the actual indicated horsepower required to compress a definite amount of gas and is expressed in percent.

<u>Compression ratio</u> for a compressor cylinder is defined as the

cylinder discharge pressure divided by the cylinder suction pressure with both pressures expressed in absolute units.

2.1.2 Types of compressors:

Air Compressors design can be divided into two major groups.

1) Positive displacement compressors:

Positive displacement compressors [1] are machines in which successive volumes of air or gas are confined within a closed space and elevated to a higher pressure. In one type, the pressure is increased as the volume of the closed space is decreased. This type includes reciprocating compressors, sliding-vane-type rotary compressors and rotary-liquid-piston-type compressors. In another type such as the rotary, two-impeller-lobe design, the trapped volume of air or gas is transformed from the suction pressure to the point of discharge without reduction in volume. The pressure increase in this type of machine, therefore takes place as air or gas

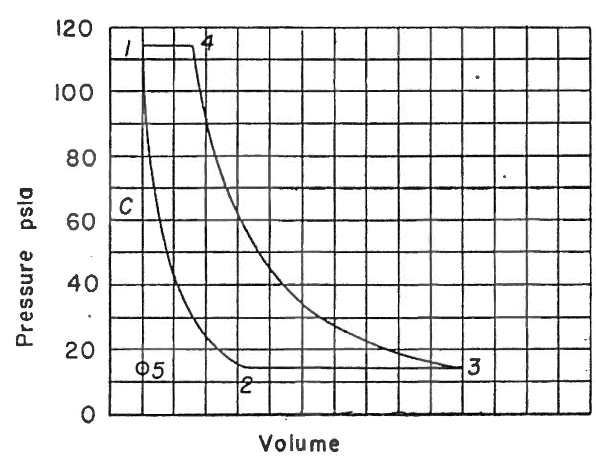


Fig. 2.1.1 Typical theoretical indicator card for a positive displacement compressor. Cylinder clearance, 10 per cent [1].

at discharge pressure is admitted to the trapped space. All positive displacement compressors in which the pressure is increased as the volume of the closed space is decreased have the same theoretical indicator card for equal design conditions. Figure 2.1.1 depicts the basic compression process and is a typical idealized card of a single-stage compressor.

2) Dynamic-type compressors:

Dynamic-type compressors [1] are machines in which air or gas is compressed by the mechanical action of rotating vanes or impellers imparting velocity and pressure to the flowing medium. By diffusing action, the velocity energy is converted into pressure. Compressors of this type include centrifugal, axial, and mixed flow. In an axial compressor, as the name implies, flow is in an axial direction; in a centrifugal compressor, flow is in a radial direction.

Dynamic-type compressors are inherently high speed machines which perform to the best advantage for relatively large capacity requirements. Centrifugal compressors can be driven by almost any prime mover be it motor, turbine, or engine.

2.1.3 Operating Characteristics:

When operated at constant speed, a centrifugal compressor will deliver a variable capacity from approximately 50 to 100 per cent of rated capacity at essentially constant discharge pressure. The shape of the pressure-capacity curve will vary slightly with the type of impeller blading used.

Figure 2.1.2 shows typical pressure-capacity curves of a centrifugal compressor with radial blades and with backward-curved blades, both at constant speed. These curves show that the compressor with radial blades has a smaller

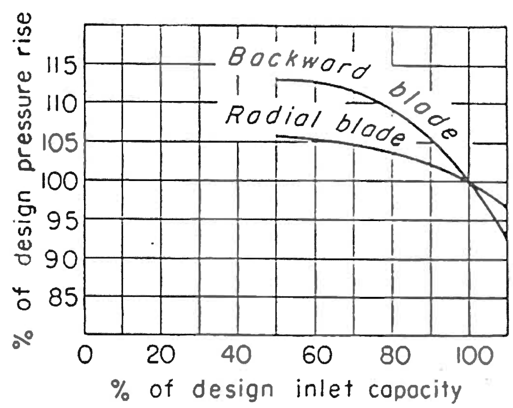


Fig. 2.1.2 Characteristic curves of a centrifugal compressor showing the effect of impeller blade angle [1].

pressure rise than the compressor with backward-curved blades; this difference in characteristics influences the selection and design of compressor control equipment. When operated at variable speed, the centrifugal compressor may be used to deliver a constant capacity at variable pressure, a variable capacity at constant pressure, or both a variable capacity and variable pressure. A typical set of pressure-capacity curves at variable speed are shown in the following figure 2.1.3. The solid lines to the right of the pumping limit, curve PP, show the normal stable operating range of the centrifugal compressor without surging. The

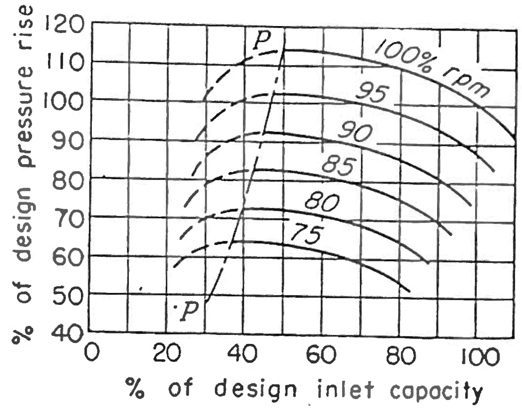


Fig. 2.1.3 Characteristics curves of a centrifugal compressor at variable speed, including adjustable inlet guide vanes [1].

dotted line to the left of this curve indicates how the stable operating range may be increased by the use of adjustable inlet guide vanes. The operating characteristics of a centrifugal compressor discussed here are based on constant inlet conditions [1].

2.1.4 <u>Factors affecting performance:</u>

One of the major and fundamental items in the performance guarantee is the volumetric efficiency. Supercompressibility affects three major items - the actual capacity, the volumetric efficiency and the brake horsepower requirement. Supercompressibility becomes important only when the pressures are approximately 1000 psig and higher [1].

2.1.5 <u>Industrial application of air compressors:</u>

Some of the industrial applications of air compressors and their capacities are listed in the following Table 2.1.1. The petroleum, natural gas, chemical, petrochemical and steel industries are currently the largest users of axial and centrifugal compressors, with typical applications as indicated in Table 2.1.1.

Table 2.1.1 Compressor applications and capacities [1]

Application	Compressor Capacity
1) Air for catalytic cracking process	110,000 icfm 5 psig
2) Gas recovery	3,000 to 50,000 icfm 125 to 300 psig
3) Alkylation refrigeration	3,000 to 33,000 icfm 85 to 115 psia
4) Nitric acid plants	10,000 to 30,000 icfm 40 psia

2.2 Electric motor driven air compressors

Electrical energy is economical and in many cases, readily available. The electric motor has attained a high degree of performance as a prime mover for air compressors. This section briefly examines the selection of motors, types of motors used for driving air compressors and finally the classification of EMDCs ([1], [2]).

2.2.1 Selection of a motor:

In selecting the physical characteristics of the motor drive, one must consider Horsepower, RPM and starting torque to accelerate the compressor from rest to full operating speed. Beside the above characteristics the efficiency of electrical input versus mechanical output and possibly the system power factor needs to be considered where this is a factor in the electric power billing [5].

2.2.2 Type of electric motors:

Two types of motors which are most commonly used for driving air compressors are the induction and synchronous motors. These motors are briefly discussed as follows.

1) Induction Motor:

The most common and most widely used motor today is the induction type. It represents the least capital investment per driving horsepower. In addition, it requires minimal maintenance since the squirrel cage rotor normally is a cast set of brass with the basic iron core. The current carrying coils are all stationary and part of the motor stator. The current in the stator is induced rather than carried in from an external source on slip rings as in the case of the synchronous motor. The induction motor, for a given RPM range, is normally slightly less efficient and has a somewhat lower power factor than the synchronous motor within the same given range. The efficiency of the induction motor is normally within two to three percentage points, and the power factor which is about 20 per cent lower is readily correctable by use of capacitors which are normally less expensive than the price difference between synchronous and induction motors complete with starting equipment [5].

2) Synchronous Motor:

The synchronous motor has a more complex starting system and needs an external source of DC power to excite the rotating field coils. This normally comes from generator set or more likely from a static rectifier. The starting sequence of a synchronous motor requires that it accelerate as an induction motor for several seconds and approach approximately 97% of its full load RPM before the field current can be applied, locking the rotating field coils into the rotating magnetic field at full synchronous speed. The slip of approximately 2 to 3% and a slight variation in speed based on load on the induction motor are normally not a serious problem to the compressor it drives [5].

2.2.3 <u>Classification of motor driven compressors</u>

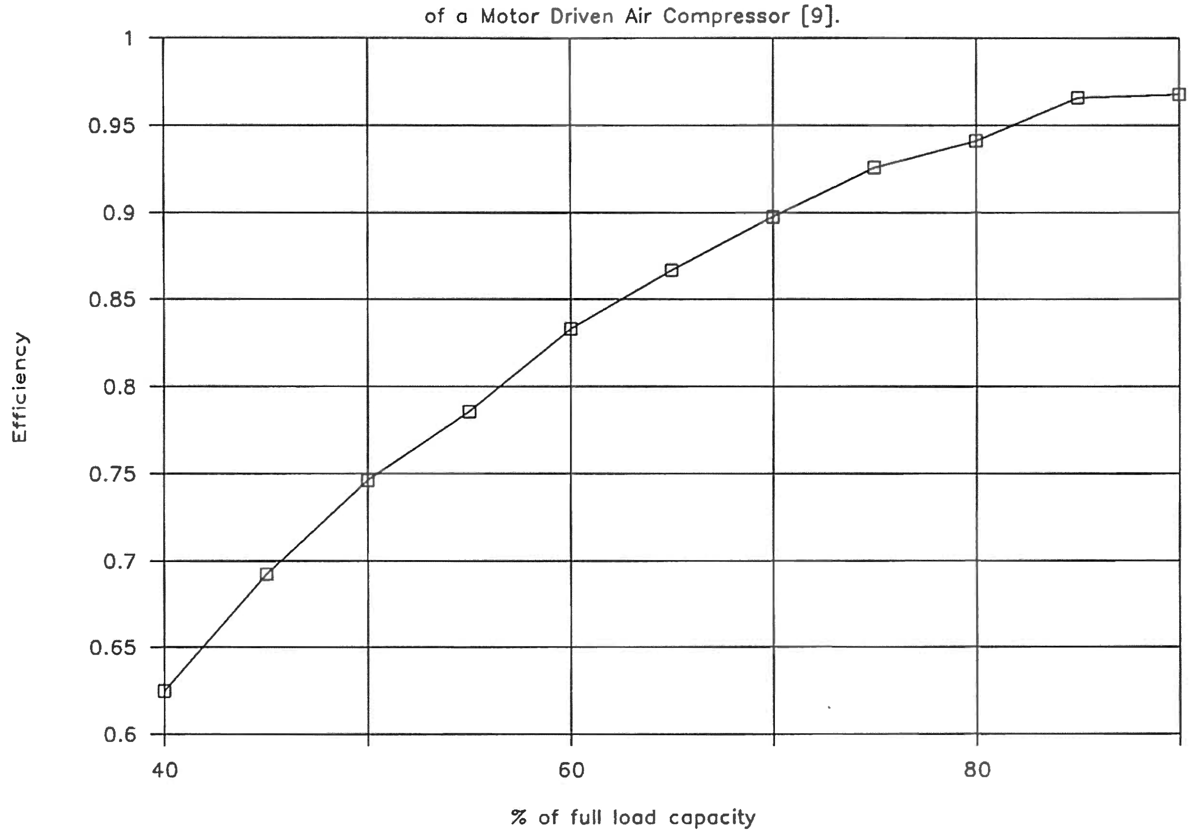
Motor driven compressors can be classified by the type of motor driving the compressor or the type of connection that is used between the motor and the compressor. A motor can be connected to a compressor in one of the following conventional ways ([1], [2]):

1) Belt (V-type or flat),
2) Flange mounting of motor,
3) Direct connected (shaft mounted) engine type synchronous motor,
4) Flexible couplings, and
5) Speed reducing gears.

Although electric motor drives are available for compressors of almost any capacity, certain types of machines are best driven by an induction motor, others by a synchronous motor [1].

Induction motors are most commonly available in the smaller horsepowers but are also readily available up to 400 HP. In the smaller horsepower range, the synchronous motors become increasingly more expensive when compared to induction motors and are generally not used in sizes below 100 to 150 HP [5]. The following figure 2.2.1 depicts the operating efficiency of one of the efficient motor driven air compressors available [9].

Fig. 2.2.1 Operating Efficiency
of a Motor Driven Air Compressor [9].



2.3 Gas engine driven air compressors

Today's industrial natural gas engines are reliable , rugged, fuel efficient prime movers that are ideally suited for various mechanical applications. The room space required is reasonable and the danger involved in operating a gas engine is not larger than any internal combustion engine. When an engine is equipped for variable speed operation, the part load requirements of mechanical equipment can be met, using far less energy. Compressing air with a rotary screw compressor can take advantage of an engine's variable speed operation ([8], [11]).

Gas engines also produce significant amounts of heat which can be recovered to preheat boiler feed water, for space heating, or for other processes requiring hot water. In addition, today's engines are designed and manufactured for very long life, and with proper maintenance will serve our needs for decades.

2.3.1 Performance of gas engines:

Rotary screw air compressors can take full advantage of variable speed operation. The rotary screw air compressor is a positive displacement machine, which, when operated at lower speeds, simply produces less air and consumes less horsepower. This characteristic makes the rotary screw air compressor coupled with a variable speed natural gas engine an ideal match. The net result is an energy efficient air compressor with excellent part-load capability. This avoids compressor inlet throttling which wastes expensive energy ([7], [12]).

Actual in-plant checks show that most compressors operate regularly in a 60-70% capacity range, and reach full loading only during peak demand periods. This results in disproportional energy expense because a typical throttled inlet compressor operating at 70% of rated capacity still requires 93% of full load horsepower, where as gas engine driven screw air compressors require much less than 93%. Table 2.3.1 shows the compressor load variation over a period of one shift and the Table 2.3.2 shows comparative part load energy requirements for ULTRA-AIR gas engine driven compressors and throttled inlet air compressors [8].

Table 2.3.1 Compressor load variation over a period of one shift ([8], [9])

Time of Day	% Loaded
9.00	90%
10.00	76%
11.00	60%
12.00	70%
13.00	60%
14.00	80%
16.00	70%
17.00	65%

Table 2.3.3 shows the efficiencies ([7], [8]) in per cent, of the prime movers of the gas engine driven air compressors (full load and 70% loaded) considered in this study.

Table 2.3.2 Comparison of % full load requirements at various part loads for a ULTRA-AIR and THROTTLED INLET compressor [8]

Per Cent of Full Load Capacity	Approx. % of Full Load Energy Required	
	ULTRA-AIR	THROTTLED INLET
90	89	98
85	84	96
80	78	95
75	73	94
70	67	93
65	65	91
60	62	90
55	61	88
50	61	87
45	60	86
40	59	84

As indicated by the Table 2.3.3 GEDC could be more economical as the compressor does not run at a constant load. GEDCs are also beneficial where expansion plans are limited due to transformer capacity, power availability, and where power is too costly to upgrade ([7], [8]).

Table 2.3.3 Efficiencies of engine driven air compressors at full and part load conditions ([7], [8])

Capacity in HP	% Full load	
	100%	70%
110	30.67	31.82
178	30.67	31.82
217	33.94	35.35
352	35.66	37.93

2.3.2 <u>Waste heat recovery and applications</u>

In addition to the above benefits GEDC offers a benefit of waste heat recovery. Various sources of waste heat and their applications are discussed below.

A considerable amount of heat is generated during compression and it is desirable in the interest of efficiency and reliability that some of this heat be carried away. Waste heat can also be recovered from jacket water coolant, the most easily recoverable source, leaving the engine at a temperature of about 190°. Approximately 30% of the heat generated from the combustion of natural gas is recoverable from the engine ([3], [8]).

Waste heat recovered can be used for: 1) preheating combustion air, 2) preheating boiler feed water, 3) space heating, 4) industrial wash operations, 5) absorption *refrigeration* or a variety of other plant applications [13].

The majority of industrial plants require dry air for normal operations. For plants requiring dew points of -40°F, the engine exhaust heated dryer will either reduce the purge air requirement (typically 14% of the dryer capacity) or eliminate the operation of an electric heater. Thus, the EGH (exhaust gas heater) Series Air Dryer, eliminates moisture as a problem, increases compressed air available for plant use, and costs next to nothing to operate (see Fig. 2.3.1 for an application) [8].

2.3.3 <u>Availability and cost of operation:</u>

Gas engine driven air compressors are commercially available and are beginning to make their presence felt. They are less expensive to operate than the electric units, averaging 4.5 cents/HP-hr to an electric motor's 10 cents/kWh [12]. They have a variable speed capacity which means they are just as efficient at any speed as they are at full speed. And they offer the benefits of optional waste heat recovery that can boost efficiencies and raise cost effectiveness. Units are now available from number of compressor manufacturers and range in sizes from 75 to 400 HP. Compressors smaller than 75 HP will be available in about a year from now. See Appendix A for information on engine driven air compressors, cost, and manufacturers [12].

2.3.4 <u>Various applications of gas engines:</u>

Stationary natural gas reciprocating engines have the exciting potential to replace small electric motors in *industrial* usage. They can provide the power and the drive for many processes. They can drive refrigerant and chiller systems, provide power for cogeneration, and create the energy needed to run compressors and a wide range of machinery. Because they produce peak efficiency at part-load conditions, they conserve energy and reduce peak electrical demands. Unlike the EMDC, GEDCs do not have the problem of lower efficiency or the power factor. Figure 2.3.1 depicts the GEDC with engine waste heat recovery [8].

Thus, many industries are already making extensive and profitable use of small gas-fueled engines. Cost-savings and increased efficiencies are the reason for the increased interest in the application of gas engine. Consequently,

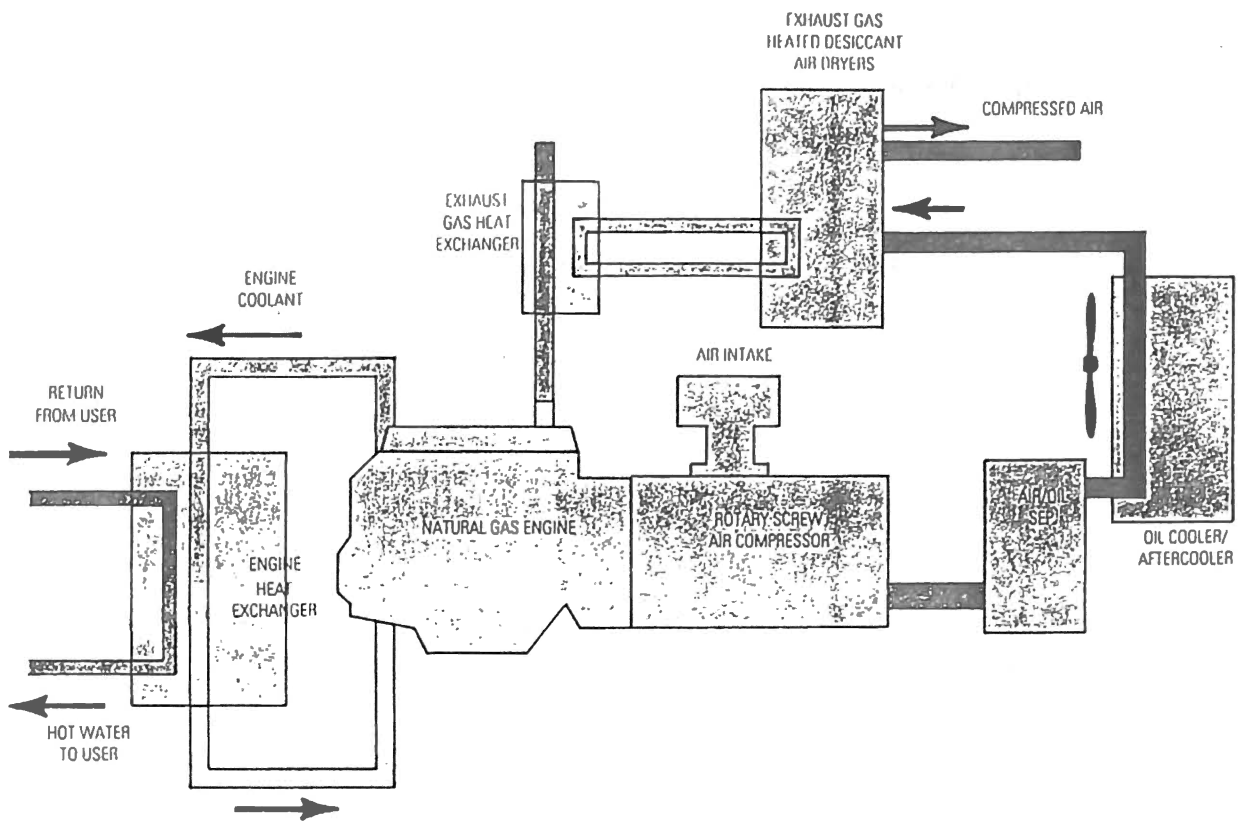


Fig. 2.3.1 Engine driven air compressor, with engine heat recovery [8].

this study intends to examine the requirements for a cost effective application of such technology to air compressors [12].

2.3.5 Research in progress:

An extensive gas engine R&D program sponsored by GRI has been underway since the early 1980s. Current concerns with environmental protection and increasing industrial productivity and efficiency have given the program added impetus and focus.

Emissions, however, are not the only goals of current small engine research. To improve the competitive position of small gas-fueled engines in industry, research is also centering on improvements in durability, reliability, and efficiency.

In one project, six small engines were tested, ranging from 6 to 35 kW (approximately 8 HP to 47 HP), for reliability, durability, and performance during extended operation. The stated objective was to determine if currently available engines could meet the demands of small cogeneration systems. Results proved that existing engines, depending on specific design, could achieve 4,000 hour service intervals and 20,000 hour engine life, making them suitable for cogeneration applications ([6], [12]).

2.3.6 Summary:

The following Table 2.3.4 summarizes the advantages and disadvantages of EMDC and GEDC.

Table 2.3.4 Comparison of EMDC and GEDC

EMDC	GEDC
1) Low initial cost	1) High initial cost
2) Maintenance cost is negligible	2) Maintenance cost is \$0.01/HP-HR
3) High efficiency	3) Low efficiency
4) Efficiency at part load drops	4) Efficiency does not drop at part load
5) Has demand cost	5) No demand cost
6) High operating cost	6) Low operating cost
7) Does not occupy much space	7) Needs more space than EMDC
8) No engine waste heat to be recovered	8) Engine waste heat can be recovered

CHAPTER 3
METHODOLOGY

3.1 Methodology:

This chapter discusses the methodology adopted in developing the simple payback equations. These equations in turn are used to develop parametric payback graphs. Simple payback graphs have been developed for different compressor capacities with and with no waste heat recovery. After going through this chapter one will be in a position to find out the feasibility of buying a GEDC based on his/her own set of constraints.

The simple payback graphs developed can be used with any rate schedule. The load factor of the compressor has been considered as 70%. The amount of waste heat that can be recovered has been considered as 30% (conservative) of the input fuel. Efficiencies of the gas engine at 70% load are obtained from section 2.3.

Considering load factor, maintenance cost, efficiency, and other factors a linear equation has been developed to plot the payback graphs. The assumptions, terms used, and the derivation of this equation are as follows.

3.2 Assumptions:

The following assumptions are used to develop the parametric equations.

1) Compressor operates during the peak hours and hence contributes to the peak demand.

2) Compressor load factor - 70%

3) Waste heat that can be recovered from GEDC - 30% of the input fuel energy.

4) Engine to compressor coupling efficiency - 94%.

5) Heater efficiency - 80%.

6) Maintenance cost of gas engine driven air compressor is $0.01/HP-HR on actual load ([7], [8]).

7) Maintenance cost of electric motor driven air compressor is assumed to be negligible.

8) The installation cost of an EMDC and an equivalent GEDC are equal.

3.3 Notation:

The following are the terms used in the derivation of the equations.

CF - Conversion Factor.

EC - Electrical consumption in kWh.

EM - Capacity of electric motor driven air compressor, in HP.

E_{EM} - Electric motor driven air compressor efficiency (assumed 90%).

E_{GE} - Gas engine driven air compressor efficiency (refer to section 2.3.1).

E_H - Heater efficiency (assumed 80%).

GC - Gas consumption in Btu/hr.

GE - Capacity of gas engine driven air compressor, in HP.

HF - Waste heat recovery factor (assumed 30%).

IIC - Incremental installation cost: This is the incremental cost to buy a GEDC instead of EMDC.

IICw - Incremental installation cost: This is the incremental cost to buy a GEDC with the waste heat recovery system instead of EMDC.

IICr - Incremental installation cost: This is the incremental cost to retrofit a EMDC with a GEDC (after making adjustments for the salvage value of the existing EMDC).

IICrw - Incremental installation cost: This is the incremental cost to retrofit a EMDC with a GEDC with waste heat recovery system (after making adjustments for the salvage value of the existing EMDC).

LF - As the compressors do not operate all the time at full load, a load factor is used, which is approximately equal to the average loading of the compressor.

MGE - Maintenance cost of gas engine.

n - Payback of the investment in years.

NS - Net savings.

SWHR - Savings from waste heat recovery.

t - Operating hours of the compressor per year.

X - Weighted average cost of electricity. This includes the demand ($/kW-yr) charge and energy ($/kWh) charge.

Sample calculations of X:

Consider OG&E rate schedule PL-1 secondary service level 5.

Demand charge = \$100.09/kW-yr.

Energy Charge = \$0.03528/kWh (assuming total electrical consumption is less than 2,000,000 kWh/billing period).

Case I:

Let operating hours of air compressor per year = 2,000 hrs.

Weighted average cost of electricity (X)

= Energy charge + Demand charge/(operating hours of air compressor per year).

= \$0.03528/kWh + (\$100.09/kW-yr)/2000 hr/yr.

= \$0.08533/kWh.

Case II:

Let operating hours of air compressor per year = 4,500 hrs.

Weighted average cost of electricity (X)

= Energy charge + Demand charge/(operating hours of air compressor per year).

= \$0.03528/kWh + (\$100.09/kW-yr)/4500 hr/yr.

= \$0.05752/kWh.

Case III:

Let operating hours of air compressor per year = 7,500 hrs.

Weighted average cost of electricity (X)

= Energy charge + Demand charge/(operating hours of air compressor per year).

= \$0.03528/kWh + (\$100.09/kW-yr)/7500 hr/yr.

= \$0.04863/kWh.

The following Table summarizes the above calculations.

Table 3.1 Weighted average costs of electricity.

Case	Hours of operation	Energy Charge (1)	Demand Charge (2)	Weighted average cost (1 + 2)
I	2000 hrs/yr	$0.03528	$0.05005	$0.08533
II	4500 hrs/yr	$0.03528	$0.02224	$0.05752
III	7500 hrs/yr	$0.03528	$0.01335	$0.04863

Whenever there is a change in the rate schedule and/or hours of operation, the weighted average cost of electricity has to be recalculated.

Y - Cost of natural gas per MCF ($/MCF).

3.4 <u>Derivation of the equation with no waste heat recovery:</u>

Let the payback period be defined by,

n = Investment/Net Savings

= IIC/NS (3.11)

Where net savings are estimated as,

NS = (EC) (X) - (GC) (Y) - MGE (3.12)

IIC, EC, GC and MGE can be expressed as,

IIC = Cost of GEDC - Cost of an equivalent EMDC

EC = (EM) (0.746 kW/HP) (t) (X) (LF) ($1/E_{EM}$)

GC = (GE) (0.746 kW/HP) (t) (LF) (3412 Btu/kWh) ($1/E_{GE}$)

(1 MCF/1,000,000 Btu) (Y)

MGE = ($0.01/HP-hr) (GE) (t) (LF)]

Substituting these values in equation 3.11, we have

n = IIC/((EC) (X) - (GC) (Y) - MGE)

= IIC/[(EM) (0.746 kW/HP) (t) (X) (LF) ($1/E_{EM}$)

$$- (GE)\ (0.746\ kW/HP)\ (t)\ (LF)\ (3412\ Btu/kWh)\ (1/E_{GE})$$

$$(1\ MCF/1{,}000{,}000\ Btu)\ (Y) - (\$0.01/HP\text{-}hr)\ (GE)\ (t)$$

$$(LF)] \tag{3.13}$$

Let CF = (0.746 kW/HP) (3412 Btu/kWh) (1 MCF/1,000,000 Btu)

Simplifying equation 3.13, we have

$$n = IIC/[(EM)\ (0.746\ kW/HP)\ (t)\ (X)\ (LF)\ (1/E_{EM})$$

$$- (GE)\ (t)\ (LF)\ (CF)\ (1/E_{GE})\ (Y)$$

$$- (\$0.01/HP\text{-}hr)\ (GE)\ (t)\ (LF)] \tag{3.14a}$$

Equation 3.14a can be rewritten as,

$$[(EM)\ (0.746\ kW/HP)\ (t)\ (X)\ (LF)\ (1/E_{EM}) - (GE)\ (t)\ (LF)\ (CF)$$

$$(1/E_{GE})\ (Y) - (\$0.01/HP\text{-}hr)\ (GE)\ (t)\ (LF)]$$

$$= IIC/n \tag{3.14b}$$

Rearranging the terms in equation 3.14b and multiplying both sides by -1, we have

$$(GE)\ (t)\ (LF)\ (CF)\ (1/E_{GE})\ (Y)$$

$$= -IIC/n + (EM)\ (0.746\ kW/HP)\ (t)\ (X)\ (LF)\ (1/E_{EM})$$

$$- (\$0.01/HP\text{-}hr)\ (GE)\ (t)\ (LF)] \tag{3.14c}$$

Rewriting the equation 3.14c in the form of Y = b + aX, where Y is the gas cost and X is the weighted average cost of electricity, we have

$$Y = - IIC/[(n)\ (GE)\ (t)\ (LF)\ (CF)\ (1/E_{GE})]$$

$$+ [(EM)\ (0.746\ kW/HP)\ (X)]/[(GE)\ (CF)\ (1/E_{GE})\ (E_{EM})]$$

$$- [(\$0.01)\ (E_{GE})]/CF \tag{3.15}$$

3.5 <u>Derivation of the equation with waste heat recovery:</u>

The payback period can be redefined as,

$$n = Investment/Net\ Savings$$

$$= IICw/NS \tag{3.16}$$

In this case net savings are estimated as,

$$NS = (EC)\ (X) - (GC)\ (Y) - MGE + SWHR$$

IICw and SWHR can be expressed as,

$$IICw = IIC + \text{Cost of waste heat recovery system}$$

$$SWHR = (GE)\ (0.746\ kW/HP)\ (t)\ (LF)\ (3412\ Btu/kWh)\ (1/E_{GE})$$
$$(1\ MCF/1{,}000{,}000\ Btu)\ (Y)\ (HF)\ (1/E_H)]$$

Substituting these values in equation 3.16, we have

$$n = IICw/((EC)\ (X) - (GC)\ (Y) - MGE + SWHR)$$
$$= IICw/[(EM)\ (0.746\ kW/HP)\ (t)\ (X)\ (LF)\ (1/E_{EM})$$
$$- (GE)\ (0.746\ kW/HP)\ (t)\ (LF)\ (3412\ Btu/kWh)\ (1/E_{GE})$$
$$(1\ MCF/1{,}000{,}000\ Btu)\ (Y)$$
$$- (\$0.01/HP\text{-}hr)\ (GE)\ (t)\ (LF)$$
$$+ (GE)\ (0.746\ kW/HP)\ (t)\ (LF)\ (3412\ Btu/kWh)\ (1/E_{GE})$$
$$(1\ MCF/1{,}000{,}000\ Btu)\ (Y)\ (HF)\ (1/E_H)] \qquad (3.17a)$$

Let $CF = (0.746\ kW/HP)\ (3412\ Btu/kWh)\ (1\ MCF/1{,}000{,}000\ Btu)$

Simplifying equation 3.17a by assuming $HF = 0.30$ and $E_H = 0.80$, we have

$$n = IICw/[(EM)\ (0.746\ kW/HP)\ (t)\ (X)\ (LF)\ (1/E_{EM})$$
$$- (GE)\ (t)\ (LF)\ (CF)\ (1/E_{GE})\ (Y)$$
$$- (\$0.01/HP\text{-}hr)\ (GE)\ (t)\ (LF)$$
$$+ (GE)\ (t)\ (LF)\ (CF)\ (1/E_{GE})\ (Y)\ (0.3)\ (1/0.8)] \qquad (3.17b)$$

Equation 3.17b can be rewritten as,

$$[(EM)\ (0.746\ kW/HP)\ (t)\ (X)\ (LF)\ (1/E_{EM}) - (GE)\ (t)\ (LF)\ (CF)$$
$$(1/E_{GE})\ (Y) - (\$0.01/HP\text{-}hr)\ (GE)\ (t)\ (LF)$$
$$+ (GE)\ (t)\ (LF)\ (CF)\ (1/E_{GE})\ (Y)\ (0.3)\ (1/0.8)]$$
$$= IICw/n \qquad (3.17c)$$

Rearranging the terms in equation 3.17c and multiplying both sides by -1, we have

$$Y(1 - 0.3/0.8)[(GE)\ (t)\ (LF)\ (CF)\ (1/E_{GE})]$$

$$= -\ IICw/n + (EM)\ (0.746\ kW/HP)\ (t)\ (X)\ (LF)\ (1/E_{EM})$$

$$-\ (\$0.01/HP\text{-}hr)\ (GE)\ (t)\ (LF) \qquad (3.18)$$

Rewriting equation 3.18 in the form of Y = b + aX, where Y is the gas cost and X is the weighted average cost of electricity, we have

$$Y = -IICw/[(n)\ (GE)\ (t)\ (LF)\ (CF)\ (1/E_{GE})\ (0.625)]$$

$$-\ \$0.01/[(CF)\ (1/E_{GE})\ (0.625)]$$

$$+\ (EM)\ (0.746\ kW/HP)\ (X)/[(GE)\ (CF)\ (1/E_{GE})\ (E_{EM})$$

$$(0.625)] \qquad (3.19)$$

The equations 3.15 (w/o waste heat recovery) and 3.19 (with waste heat recovery) will be used to develop parametric plots and also in the sensitivity analysis in the following chapters.

CHAPTER 4

ANALYSIS OF RESULTS

Within the U.S. the utility cost of electricity varies from 2.70 cts/kWh to 15.45 cts/kWh, and that of the gas varies from $2.61/MCF to $13.34/MCF [14]. Hence, the upper limits of the X-axis, and Y-axis have been chosen to be 20 cts/kWh, and $20/MCF respectively. Math CAD has been used to develop the Payback graphs used in this research [15]. Refer to appendix B for a list of utility rates within the U.S. In this section the case studies considered are briefly discussed and also steps to use the payback graphs are listed.

4.1 <u>Case studies considered:</u>

The following are the case studies considered in this research. As an equivalent GEDC was not available for the EMDC in study, the next higher capacity of GEDC available has been chosen.

Case I - A 110 HP GEDC as an alternate for a 100 HP EMDC.

Case II - A 217 HP GEDC as an alternate for a 200 HP EMDC.

Case III - A 352 HP GEDC as an alternate for a 350 HP EMDC.

Economics of retrofiting the EMDC with GEDC can also be studied. But, this paper limits to the above three case studies mentioned.

4.2 <u>Discussion of results:</u>

Figures 4.1 to 4.6 illustrate the fact that keeping the gas cost constant with an increasing electricity cost results in payback getting more and more attractive. Where as, if the electricity cost were to be constant with an increasing gas cost, the payback gets less attractive. These figures also illustrate the fact that irrespective of the compressor capacity, if the waste heat (generated by burning the fuel) could be recovered the payback would be even more attractive. Chapter 5 lists the controlling parameters and discusses the effect of these parameters on the payback.

With low gas cost, irrespective of hours of operation or capacity the payback will not be effected always. In Case I, with fewer hours of operation and a low gas cost, it takes longer to payback with waste heat recovery. This is due to the additional investment incurred for the waste heat recovery system. Whereas in the case of larger capacities this is not true, as the payback will not be affected by either change in capacity or hours of operation. However, with a larger gas cost, payback will be faster with waste heat recovery irrespective of hours of operation or capacities.

Figures 4.7 to 4.12 are payback graphs as function of hours of operation and gas cost. These graphs have been developed for OG&E rate schedule PL-1 secondary service level 5 with varying gas cost and varying hours of operation. From these figures it is very clear that, lesser the gas cost and more the hours of operation faster is the payback. The investment becomes less attractive as the gas cost increases and at a stage the payback will be no longer attractive.

Hence, it can be concluded that for every set of constraints there is a break even point (gas cost) beyond which the investment will longer be attractive.

4.3 How to use the payback graphs:

The following is the procedure to use the payback graphs.

Step 1 - Compute the weighted average cost of electricity X (refer to pg. #23).

Step 2 - Draw a line perpendicular to the X-axis from a point equal to the electricity computed in step 1.

Step 3 - Draw a line perpendicular to Y-axis from a point equal to the gas cost.

Step 4 - The point of intersection of the lines drawn in steps 1 and 3 specifies the payback region.

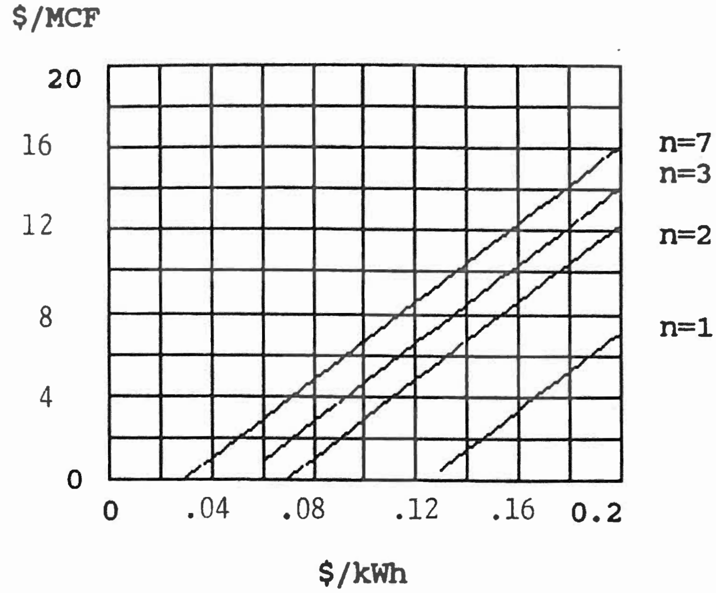


Fig. 4.1. Payback graph for case I,[1] operating 4500 hrs/yr, with a load factor of 0.70, and with no waste heat recovery.

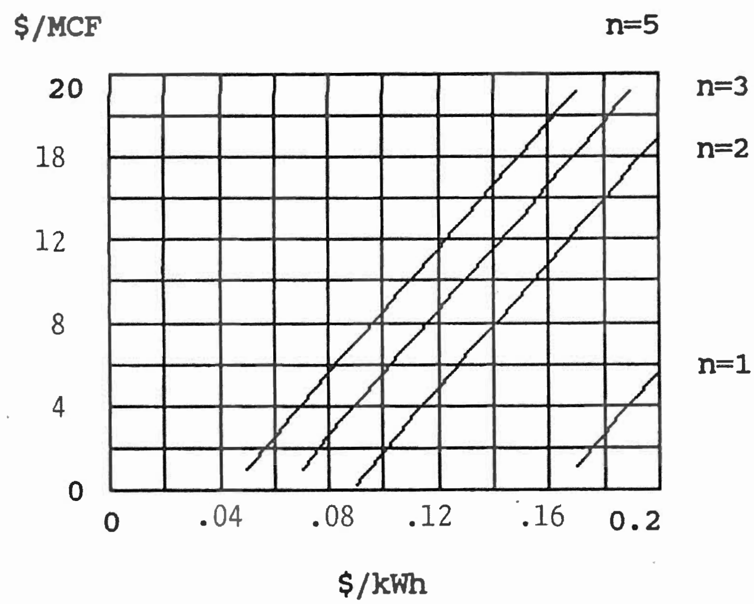


Fig. 4.2. Payback graph for case I,[1] operating 4500 hrs/yr, with a load factor of 0.70, and with waste heat recovery.

1 - A 110 HP GEDC as an alternate for a 100 HP EMDC

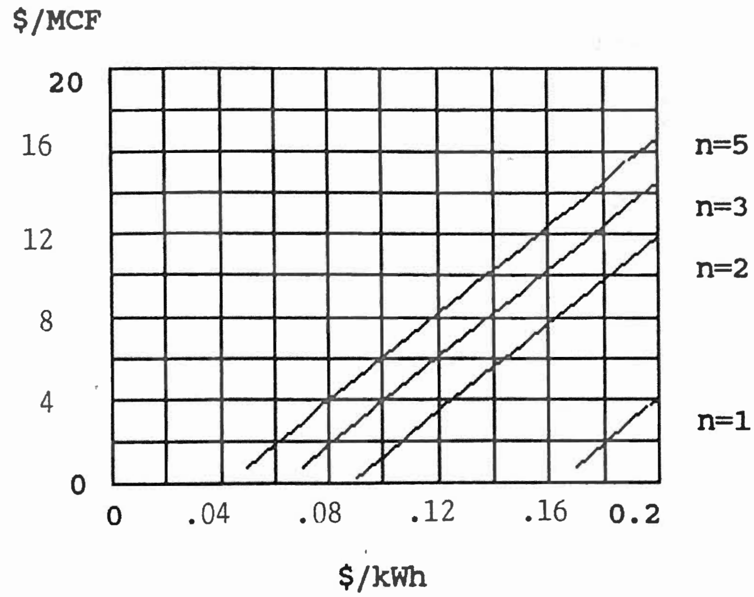


Fig. 4.3. Payback graph for case II[2], operating 4500 hrs/yr, with a load factor of 0.70, and with no waste heat recovery.

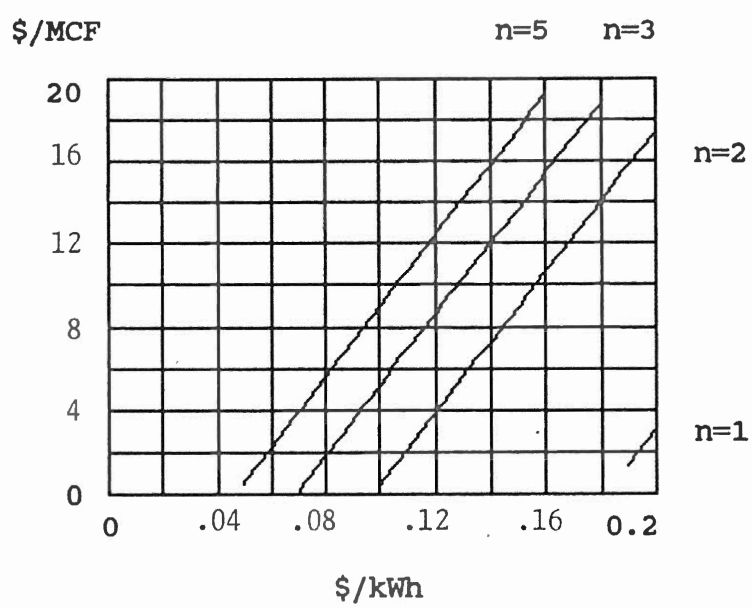


Fig. 4.4. Payback graph for case II[2], operating 4500 hrs/yr, with a load factor of 0.70, and with waste heat recovery.

2 - A 217 HP GEDC as an alternate for a 200 HP EMDC

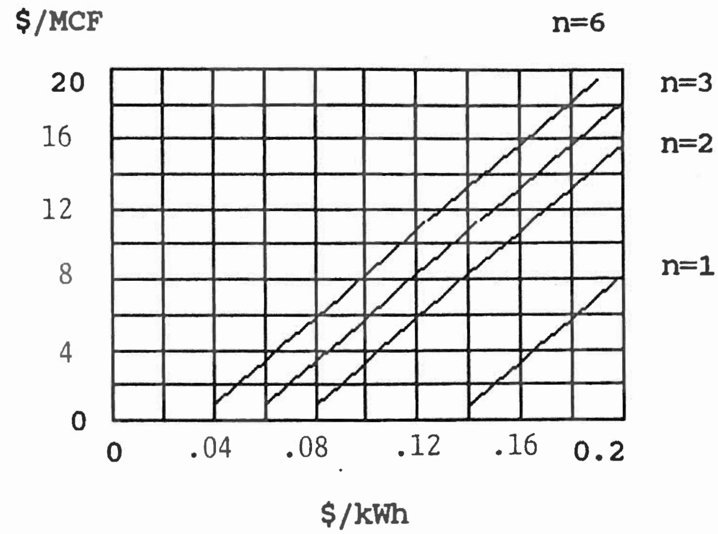


Fig. 4.5. Payback graph for case III[3], operating 4500 hrs/yr, with a load factor of 0.70, and with no waste heat recovery.

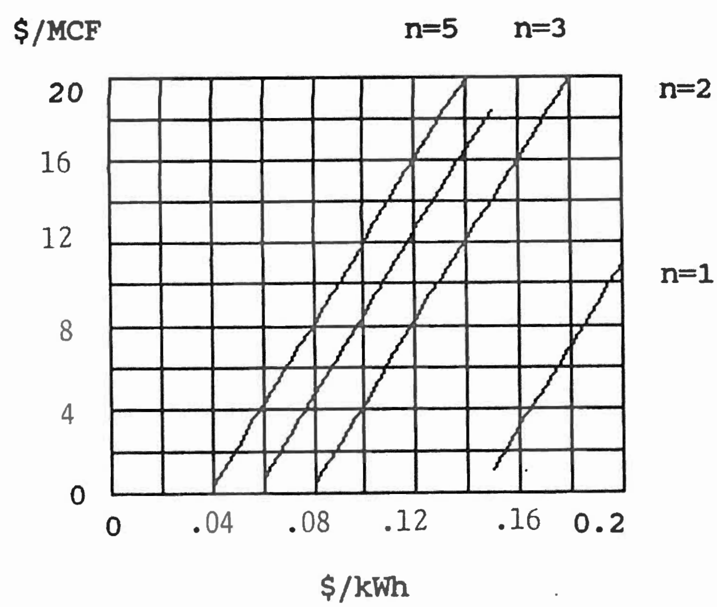


Fig. 4.6. Payback graph for case III[3], operating 4500 hrs/yr, with a load factor of 0.70, and with waste heat recovery.

3 - A 352 HP GEDC as an alternate for a 350 HP EMDC

Fig 4.7 Payback as f(gas cost, hrs/yr)

for Case I, with no waste heat recovery

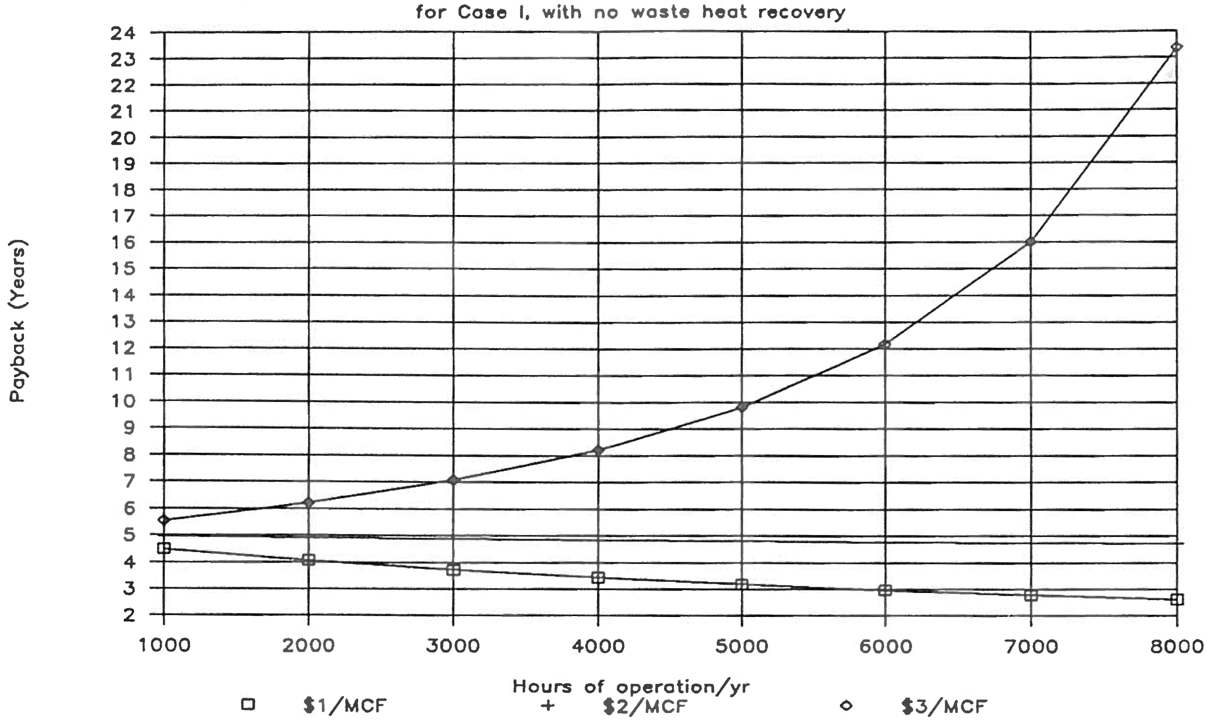
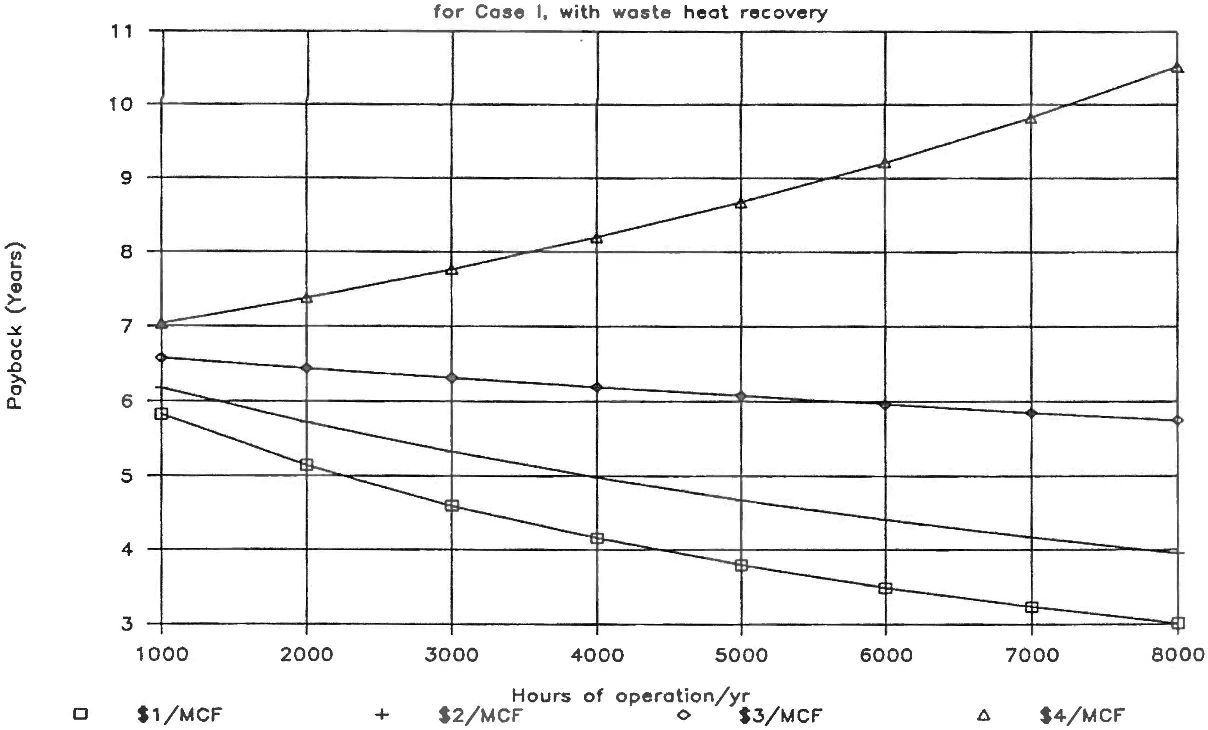


Fig 4.8 Payback as f(gas cost, hrs/yr)

for Case I, with waste heat recovery



Case I - A 110 HP GEDC as an alternate for a 100 HP EMDC, operating at 70% load, OG&E PL-1 Secondary Service Level 5

Fig 4.9 Payback as f(gas cost, hrs/yr)

for Case II, with no waste heat recov.

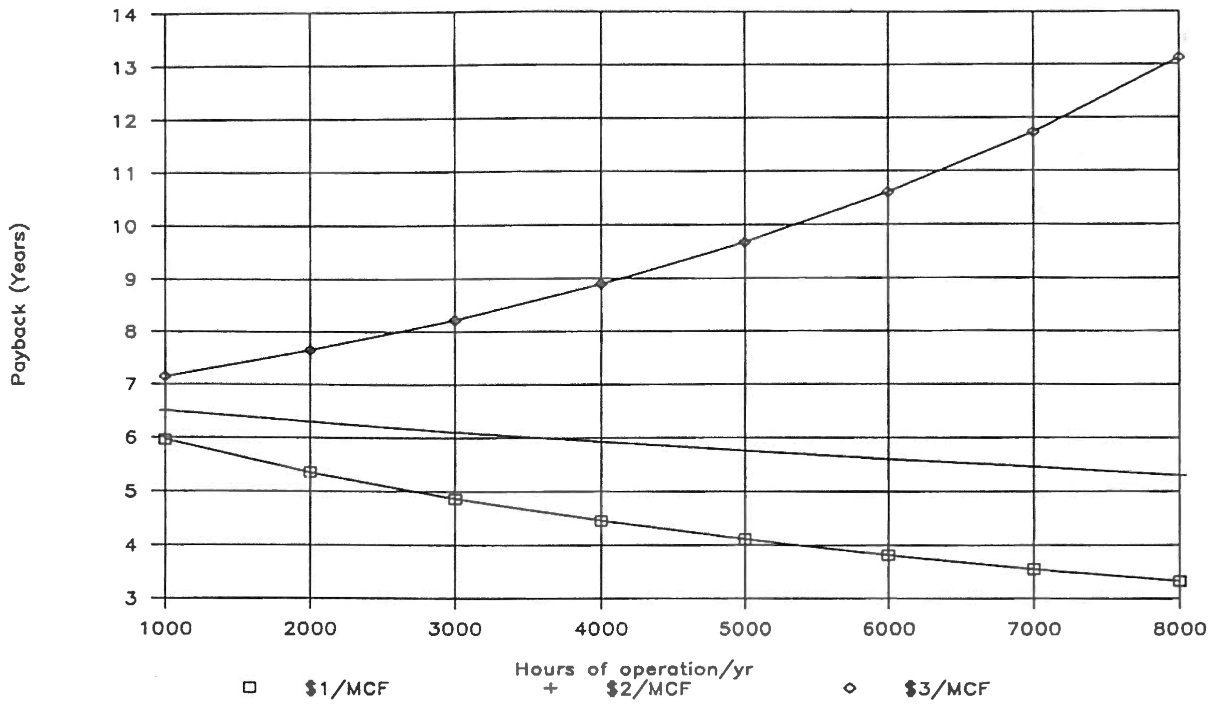
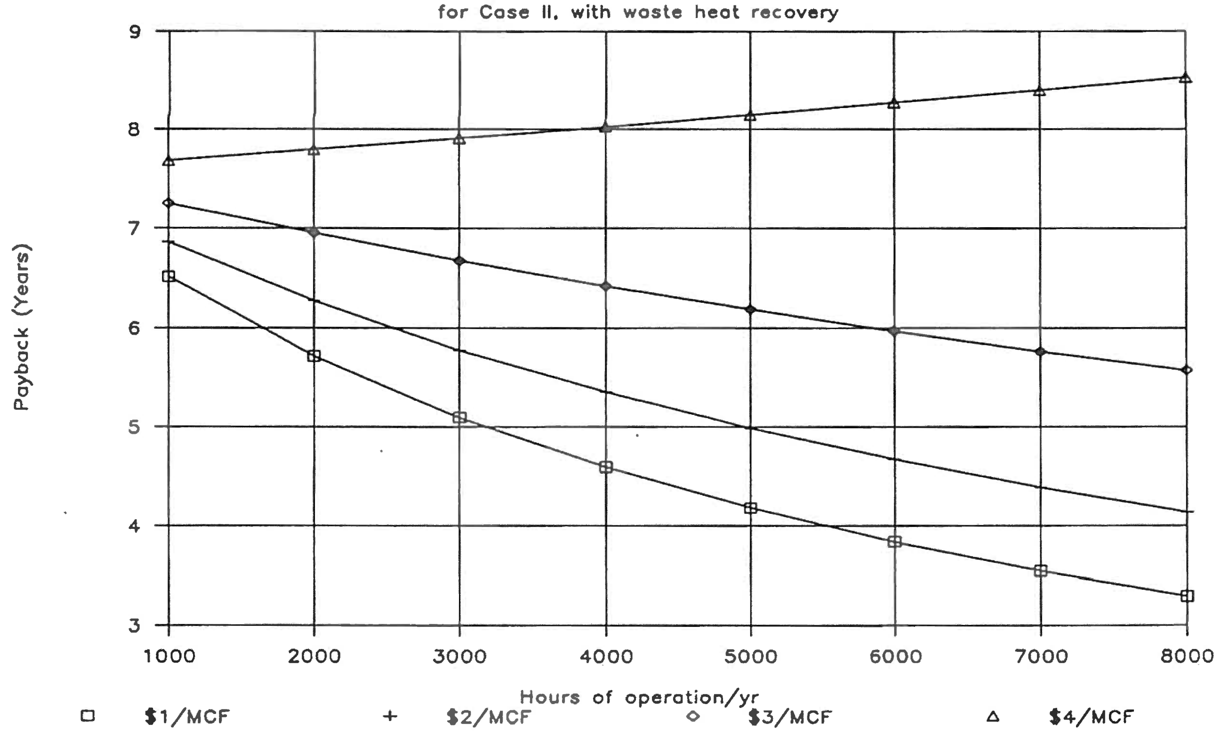


Fig 4.10 Payback as f(gas cost,hrs/yr)

for Case II, with waste heat recovery



Case II - A 217 HP GEDC as an alternate for a 200 HP EMDC, operating at 70% load, OG&E PL-1 Secondary Service Level 5

Fig 4.11 Payback as f(gas cost,hrs/yr)

for Case III, with no waste heat recov.

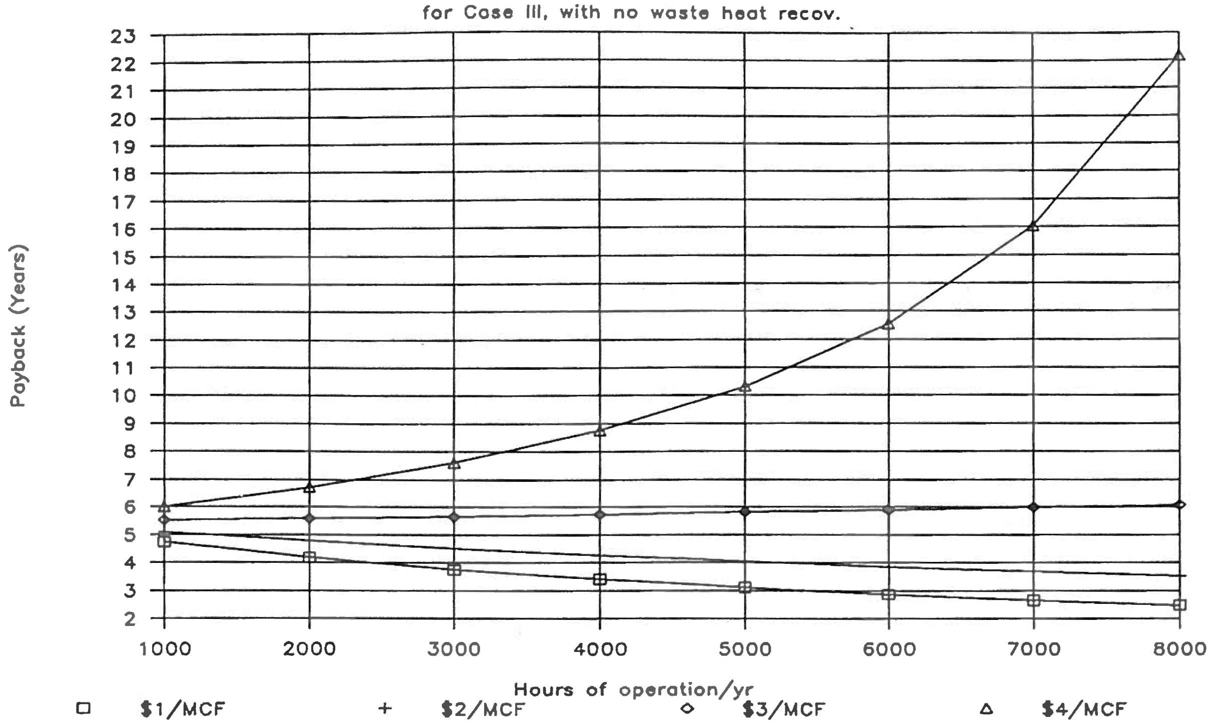
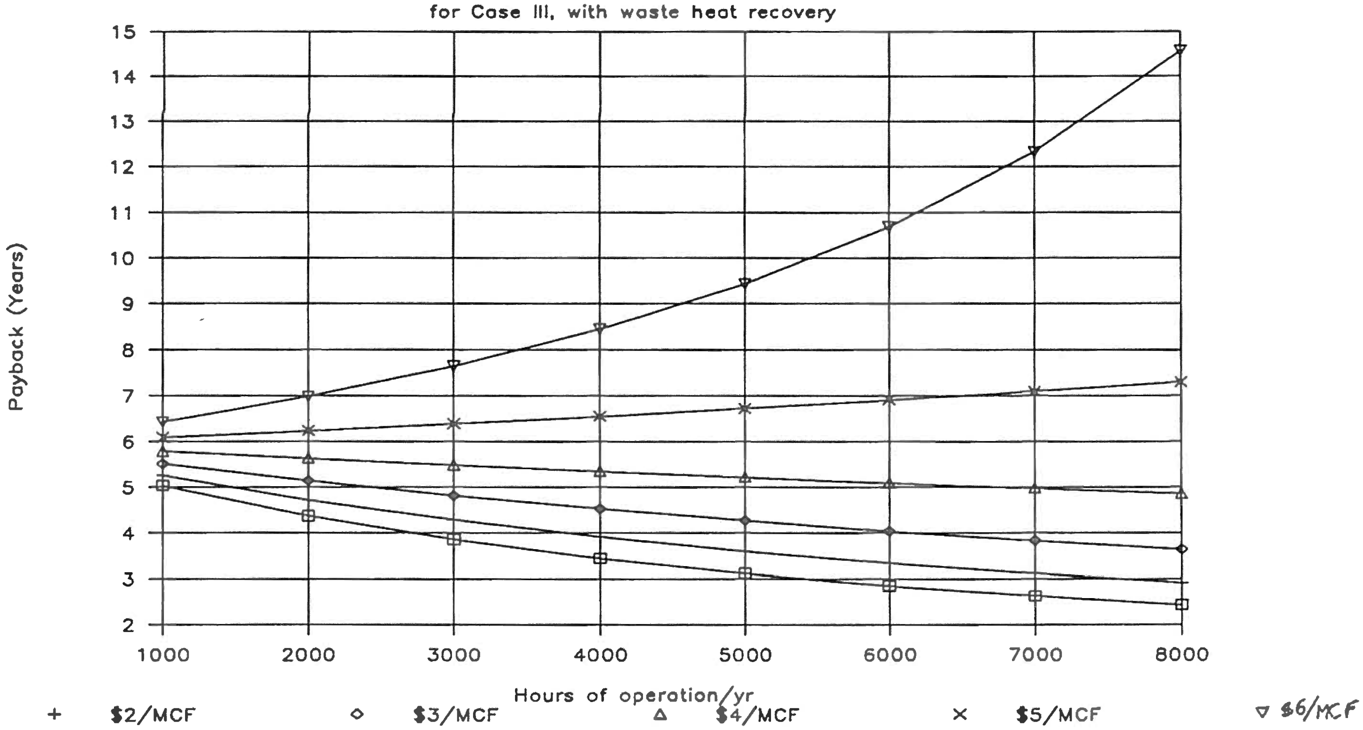


Fig 4.12 Payback as f(gas cost,hrs/yr)

for Case III, with waste heat recovery



Case III - A 352 HP GEDC as an alternate for a 350 HP EMDC, operating at 70% load, OG&E PL-1 Secondary Service Level 5

CHAPTER 5
SENSITIVITY ANALYSIS

The measure of evaluation used in this study is simple payback. The payback is dependent on the following parameters: 1) initial cost, 2) hours of operation, 3) compressor capacity, 4) load factor, 5) GEDC efficiency, 6) EMDC efficiency, 7) maintenance cost of engine driven compressor, 8) electricity cost, 9) gas cost, and 10) waste heat recovery. To reduce the complexity of the analysis, variables 4, 5, 6 and 7 will be considered constant. The effect of changing the above variables on the payback is examined in this chapter.

5.1 <u>Change in capital/initial cost:</u>

As the payback is directly proportional to the initial investment, payback is a linear function of the initial cost.

5.2 <u>Change in hours of operation:</u>

From the payback graphs, it can be concluded that the hours of operation have an effect on the payback. Refer to Fig. C1 (pg. #53) with an electricity cost of 12 cts/kWh and gas cost of $2/MCF, the payback for the investment is 3 years. Now consider Fig. C3 (pg. #54) for the same utility rates, the payback for the investment is less than 1 year. However, from Fig. 4.7 to Fig. 4.12 it is obvious that increasing hours of operation do not always lead to more attractive payback. Keeping the electricity cost constant, increasing the hours of operation with an increase in the gas cost, the payback tends to become less attractive.

5.3 <u>Change in compressor capacity:</u>

Changing compressor capacity, keeping the other variables constant does not have much effect on the payback. Considering the Fig. C3 (pg. #54), Fig. C7 (pg. #56), and Fig. C11 (pg. #58) the payback for the investment with an electricity cost of 10 cts/kWh and gas cost of $6/MCF. Figures C3 and C7 shows a payback of 2 years, where as Fig. C11 shows a payback of a little less than 2 years.

5.4 <u>Change in utility rates:</u>

It is very clear from the payback graphs that the cost of electricity and gas can affect the payback. For a constant gas cost and increasing electricity cost, the payback becomes more attractive, where as for a constant electricity cost and increasing gas cost the payback gets less attractive. This fact is true irrespective of the capacity of the compressor, hours of operation, initial cost, maintenance cost of the engine driven compressor and with or with out waste heat recovery. However a change in load factor and/or efficiency might affect the payback.

5.5 <u>Choice of waste heat recovery:</u>

The choice of waste heat recovery has an effect on the payback. Consider the figures C11 (pg. #58) and C12 (pg. #58) for an electricity cost of 8 cts/kWh and gas cost of $4/MCF. The payback from figure C11 is 2 years, where as the payback from figure C12 is less than 2 years. This example is illustrated as follows, using the original equations 3.13 and 3.17a.

We have,

IIC = Cost of 352 HP GEDC - Cost of 350 HP EMDC

= \$173,900 - \$63,000

= \$110,900

IICw = IIC + Cost of heat recovery system

= \$110,900 + \$10,000 (max.)

= \$120,900

EM = 350 HP

t = 7500 hr/yr

LF = 0.70

E_{EM} = 0.90 (refer to Fig. 2.2.1)

GE = 352 HP

E_{GE} = 0.3793 (refer to Table 2.3.3)

HF = 30% of input fuel to GEDC (assumed)

E_H = 80% Heater efficiency (assumed)

For payback with no waste heat recovery we have the equation 3.13.

$$n = IIC/((EC)(X) - (GC)(Y) - MGE)$$

$$= IIC/[(EM)(0.746\ kW/HP)(t)(X)(LF)(1/E_{EM}) - (GE)(0.746\ kW/HP)(t)(LF)(3412\ Btu/kWh)(1/E_{GE})(1\ MCF/1{,}000{,}000\ Btu)(Y) - (\$0.01/HP\text{-}hr)(GE)(t)(LF)] \quad (3.13)$$

Substituting the above values in equation 3.13, we have

$$n = (\$173{,}900 - \$63{,}000)/[(350\ HP)(0.746\ kW/HP)(7500\ hr/yr)(\$0.08/kWh)(0.70)(1/0.90) - (352\ HP)(0.746\ kW/HP)$$

(7500 hr/yr) (0.70) (3412 Btu/kWh) (1/0.3793)

(1 MCF/1,000,000 Btu) ($4/MCF) - ($0.01/HP-hr) (352 HP)

(7500 hr/yr) (0.70)]

= ($110,900)/[$121,846.67 - $49,605.17 - $18,480]

= ($110,900)/($53,761.50)

= 2.06 yrs.

For payback with waste heat recovery we have the equation 3.16.

$$n = IIC/((EC)\ (X) - (GC)\ (Y) - MGE + SWHR)$$

$$= IIC/[(EM)\ (0.746)\ (t)\ (X)\ (LF)\ (1/E_{EM}) - (GE)\ (0.746)\ (t)\ (LF)\ (3412\ Btu/kWh)\ (1/E_{GE})\ (1\ MCF/1{,}000{,}000\ Btu)\ (Y) - (\$0.01/HP\text{-}hr)\ (GE)\ (t)\ (LF) + (GE)\ (0.746)\ (t)\ (LF)\ (3412\ Btu/kWh)\ (1/E_{GE})\ (1\ MCF/1{,}000{,}000\ Btu)\ (Y)\ (HF)\ (1/E_H)] \quad (3.16)$$

Substituting the values of EM, GE, E_{EM}, E_{GM}, t, LF, HF and E_H, we have

n = ($183,900 - $63,000)/[(350 HP) (0.746 kW/HP) (7500 hr/yr)

($0.08/kWh) (0.70) (1/0.90) - (352 HP) (0.746 kW/HP)

(7500 hr/yr) (0.70) (3412 Btu/kWh) (1/0.3793)

(1 MCF/1,000,000 Btu) ($4/MCF) - ($0.01/HP-hr) (352 HP)

(7500 hr/yr) (0.70) + (352 HP) (0.746 kW/HP)

(7500 hr/yr) (0.70) (3412 Btu/kWh) (1/0.3793)

(1 MCF/1,000,000 Btu) ($4/MCF) (0.30) (1/0.80)]

= ($120,900)/[($121846.67 - $49605.17 - $18480 + $18601.94]

= ($120,900)/($72,363.44)

= 1.67 yrs.

However, for a compressor having fewer hours of operation, the choice of waste heat recovery is not economical as an additional cost is incurred for the waste heat recovery system. For the above utility cost, consider Fig. C10 (pg. #57), and Fig. C12 (pg. #58). The payback from Fig. C10 is about 6 years, where as the payback from Fig. C12 is less than 2 years.

It can be concluded from this chapter that after the initial cost, the payback is most sensitive to utility cost.

CHAPTER 6

CONCLUSIONS AND RECOMMENDATIONS

6.1 Conclusions

Payback graphs have been developed for different compressor capacities, different operating hours, and different utility rates. It has been found that GEDC is economical when compared to EMDC under certain conditions.

The following conclusions can be made from this research:

* With increasing number of hours of operation, the payback for the investment will be faster, irrespective of capacity of the compressor. This is not always true. In certain cases, keeping the electricity cost constant, increasing the gas cost with an increase in the hours of operation, the payback tends to become less attractive.
* The option of GEDC is not economical for a compressor operating single shift with an electricity cost of less than $0.10/kWh.
* If natural gas can be purchased at a cheaper price (by negotiating with the utility, self help gas, etc.) than the current price, the payback will be faster.
* With the more economical and more efficient gas engines to come, engine driven compressors will become more and more attractive.

6.2 Recommendations for further research

Following is the scope of further research:

* The payback graphs developed in this research assume

constant dollars. As time value of money has an affect on the payback, payback graphs with time value of money could be developed.

* The payback graphs could be developed considering variable load and efficiency.
* A computer model could be developed where in the user inputs the variables: 1) compressor capacities, 2) operating hours, 3) load factor, 4) efficiency, 5) incremental cost, 6) maintenance costs, 7) option of heat recovery, and 8) rate of return. The output will be a payback graph with multiple payback's, with electrical cost on the X-axis and gas cost on Y-axis.
* The methodology used in this research could be extended to engine driven refrigeration compressors and chillers.

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APPENDICES

APPENDIX A

Engine Driven Air Compressors Cost Data

APPENDIX A

DEARING

Contact #: (216) 783-2258

GRS 110 HP SLIA	$46,000
GRS 178 HP LA	$65,000
GRS 217 HP LA	$110,000
Accessories:	
Enclosure	$4,000
Custom heat recovery fitting	$10,000 (Max.)

SULLAIR

Contact #:(219) 879-5451

W Cat 3408, 1600 CFM (with enclosure)	$175,000

INGERSOLL-RAND

(with enclosure)

Contact #: (800) 375-5678/(405) 947-0931

		Engine (Delivered)	
1600L-NG	1600 CFM	375 BHP (352 HP)	$173,900
1400H-NG	1400 CFM	371 BHP (349 HP)	$173,900
300HH-NG	1300 CFM	384 BHP (361 HP)	$173,900

ALTURDYNE

Contact #: (619) 565-2131

ALTURDYNE is in the process of developing gas engine driven air compressors. They are currently developing 40 HP, and 80 HP compressors. These compressors will be available in the market approximately one year from now.

Electric Motor Driven Air Compressor and Costs

Gardner - Denver Electro Saver Rotary Screw Air Compressors with enclosures.

Capacity	Cost
100 HP	$21,000
150 HP	$31,500
200 HP	$36,000
250 HP	$42,700
300 HP	$47,024
350 HP	$63,000
375 HP	$74,000

APPENDIX B

Utility Rates Within The U.S.

RANKING OF ELECTRICITY PRICES — COMMERCIAL

Rank	Cts/kwh	State	UTILITY	Cum. kwh %
1.	15.45	NY	Long Island Lighting	0.6%
2.	15.14	NY	Orange/Rockland Utils.	0.7%
3	13.13	Pa	Philadelphia Elec.	1.5%
4	13.11	HI	Hawaii Elec. Light	1.5%
5	12.52	NY	Consolidated Edison	5.1%
6.	12.26	HI	Maui Electric	5.1%
7	12.06	Me	Bangor Hydroelectric	5.1%
8.	12.04	Ct	United Illum.	5.5%
9.	11.86	Oh	Toledo Edison	5.8%
10.	11.72	Ca	Pacific G&E	11.3%
11.	11.66	Ma	Commonwealth Elec.	11.4%
12.	11.24	Az	Tucson Elec	11.6%
13	11.24	NH	P.S. New Hampshire	11.8%
14	11.17	RI	Blackstone Valley Elec.	11.8%
15	11.07	Ca	S. Calif. Edison	15.7%
16.	10.72	MI	Detroit Edison	17.1%
17	10.61	NY	Rochester G&E	17.4%
18.	10.58	Oh	Cleveland Elec.	18.2%
19.	10.49	NJ	Jersey Central P&L	19.1%
20.	10.37	NY	Niagara Mohawk Power	20.8%
21.	10.19	Md	Baltimore Gas & Elec.	21.2%
22.	10.06	Ma	W. Massachusetts Elec.	21.3%
23.	10.05	Oh	Ohio Edison	22.2%
24.	9.88	NY	New York State E&G	22.7%
25	9.81	NJ	Atlantic Electric	23.1%
26.	9.70	NJ	Public Service E&G	25.8%
27.	9.61	IL	Illinois Power	26.5%
28.	9.57	Ca	San Diego G&E	27.3%
29	9.55	Az	Arizona PS	[illegible]
30	9.54	Ct	Connecticut L&P	29.6%
31	9.52	Ak	Golden Valley Elec.	29.6%
32	9.51	Ma	Eastern Utilities	[illegible]
33	9.47	NM	PS New Mexico	[illegible]
34	9.33	SD	Northwestern P.S.	30.1%
35.	9.33	RI	Narragansett Elec.	30.4%
36.	9.30	Ca	Los Angeles Mun.	31.6%
37.	9.05	CA	Sacramento MUD	[illegible]
38.	8.97	[illegible]	Commonwealth Edison	[illegible]
39.	8.94	[illegible]	Texas-N.M. Power	[illegible]
40.	8.91	[illegible]	Massachusetts Elec.	[illegible]
41	8.87	[illegible]	Central Hudson G&E	[illegible]
42	8.86	[illegible]	Northern Indiana P.S.	[illegible]
43	8.78	[illegible]	Modesto Irrign. Distr.	[illegible]
44	8.76	[illegible]	Central Power & Light	[illegible]
45	8.55	[illegible]	Mississippi P&L	[illegible]
46.	8.49	[illegible]	Boston Edison	[illegible]
47.	8.43	[illegible]	Arkansas P&L	[illegible]
48	8.41	[illegible]	Salt River District	[illegible]
49	8.35	[illegible]	Hawaiian Electric	[illegible]
50	8.30	[illegible]	El Paso Electric	[illegible]
51	8.28	[illegible]	Potomac Electric	[illegible]
52	8.26	[illegible]	Iowa Elec. L&P	[illegible]
53	8.16	[illegible]	Delmarva P&L	[illegible]
54	8.14	[illegible]	Central Maine Power	[illegible]
55	8.10	[illegible]	Anchorage Mun. L&P	[illegible]
56	7.95	[illegible]	Oklahoma G&E	[illegible]
57	7.94	[illegible]	Pennsylvania P&L	[illegible]
58	7.91	[illegible]	New Orleans P.S.	[illegible]
59	7.82	[illegible]	Delmarva P&L	[illegible]
60	7.80	[illegible]	C. Illinois PS	[illegible]
61	7.65	[illegible]	Metropolitan Edison	[illegible]
62	7.55	[illegible]	Houston L&P	[illegible]
63	7.49	[illegible]	Gulf States Utilities	[illegible]
64	7.41	[illegible]	Georgia Power	[illegible]
65	7.36	[illegible]	Union Electric	[illegible]
66	7.29	[illegible]	Kansas City P&L	[illegible]
67	7.25	[illegible]	Kansas City P&L	[illegible]
68	7.23	[illegible]	Pacific P&L	[illegible]
69	7.15	[illegible]	Pennsylvania Elec.	[illegible]
70	7.14	[illegible]	Cambridge Elec.	[illegible]
71	7.12	[illegible]	Green River Electric	[illegible]
72	7.11	[illegible]	Otter Tail Power	[illegible]
73	7.10	[illegible]	Carolina P&L	[illegible]
74	7.09	[illegible]	Iowa-Illinois G&E	[illegible]
75	7.06	[illegible]	Carolina P&L	[illegible]
76	7.01	[illegible]	Omaha PPD	[illegible]
77	7.01	[illegible]	"Union L,H&P"	[illegible]
78	7.00	[illegible]	Black Hills Corp.	[illegible]
79	6.99	[illegible]	Louisiana P&L	[illegible]
80	6.88	[illegible]	Chugach Elec. Assn.	[illegible]
81	6.87	[illegible]	Alabama Power	[illegible]
82.	6.86	[illegible]	Iowa Power & Light	[illegible]
83.	6.83	[illegible]	UtiliCorp United	[illegible]
84	6.83	[illegible]	Kansas City agency	[illegible]
85.	6.82	[illegible]	Gulf States Utilities	[illegible]
86.	6.82	Tn	Knoxville agency	58.2%
87.	6.81	Vt	Green Mountain Power	58.2%
88.	6.76	ND	Montana-Dakota Utils.	58.3%
89.	6.73	Ga	Savannah Electric	58.5%
90.	6.70	NC	Virginia E&P	58.6%
91.	6.65	Fl	Jacksonville agency	59.0%
92.	6.60	Tn	Nashville agency	59.1%
93.	6.59	Fl	Tampa Elec.	59.8%
94.	6.58	Tn	Chattanooga agency	59.8%
95.	6.57	Oh	Cincinnati G&E	60.5%
96.	6.56	Ia	Interstate Power	60.6%
97.	6.55	Mi	Consumers Power	62.0%
98.	6.51	Tx	Austin agency	62.5%
99.	6.49	Fl	Florida P&L	66.9%
100.	6.49	Ne	Nebraska PPD	67.0%
101.	6.42	SC	Duke Power Co.	67.5%
102.	6.36	Nv	Nevada Power	67.8%
103.	6.34	ND	Northern States Power	67.9%
104.	6.31	Va	Virginia E&P	70.5%
105.	6.28	Oh	Dayton P&L	71.0%
106.	6.25	Nv	Sierra Pacific Power	71.3%
107.	6.21	Tx	Texas Utilies Elec.	74.9%
108.	6.21	Wi	Northern States Power	75.0%
109.	6.21	Wi	Madison G&E	75.2%
110	6.16	In	Indiana/Michigan Elec.	75.7%
111	6.11	Tx	San Antonio agency	76.1%
112.	6.03	Wi	Wisconsin Elec.	77.0%
113.	5.99	Ky	Louisville G&E	[illegible]
114.	5.98	In	Indianapolis P&L	[illegible]
115.	5.94	WV	Monongahela Power	[illegible]
116	5.94	NC	Duke Power	79.5%
117	5.94	Co	PS Colorado	[illegible]
118.	5.93	[illegible]	Potomac Edison	[illegible]
119.	5.91	[illegible]	Northern States Power	82.0%
120.	5.89	Wi	Wisconsin P&L	[illegible]
121.	5.88	Ut	Pacific P&L	82.8%
122.	5.76	[illegible]	Mississippi Power	83.2%
123.	5.73	[illegible]	C. Louisiana Elec.	83.4%
124.	5.70	[illegible]	Memphis agency	[illegible]
125	5.68	[illegible]	Minnesota P&L	[illegible]
126.	5.67	[illegible]	South Carolina E&G	[illegible]
127.	5.67	[illegible]	Southwestern PS	[illegible]
128.	5.65	[illegible]	Iowa Southern Utils.	[illegible]
129.	5.65	[illegible]	Southwestern Elec.	[illegible]
130.	5.65	[illegible]	Southwestern Elec.	[illegible]
131.	5.64	[illegible]	Wisconsin PS	[illegible]
132.	5.63	[illegible]	Huntsville agency	[illegible]
133.	5.63	[illegible]	Ohio Power	[illegible]
134	5.54	[illegible]	Tri-County Elec.	[illegible]
135	5.51	[illegible]	Southern Indiana G&E	[illegible]
136.	5.51	[illegible]	Kentucky Power	[illegible]
137	5.47	[illegible]	PS Oklahoma	[illegible]
138	5.46	[illegible]	Florida Power	[illegible]
139	5.40	[illegible]	Washington Water Power	[illegible]
140	5.38	[illegible]	Appalachian Power	[illegible]
141	5.36	[illegible]	Columbus Southern Power	[illegible]
142	5.32	[illegible]	Empire District Elec.	[illegible]
143	5.30	[illegible]	West Penn Power	[illegible]
144	5.27	[illegible]	Puget Sound P&L	[illegible]
145	5.24	[illegible]	Washington Water P.	[illegible]
146	5.23	[illegible]	Gulf Power	[illegible]
147	5.13	[illegible]	Iowa Public Service	[illegible]
148	5.05	[illegible]	South Carolina PS	[illegible]
149	4.99	[illegible]	"Cheyenne L,F&P"	[illegible]
150	4.96	[illegible]	Lansing Bd. Water & Light	[illegible]
151	4.94	[illegible]	Pacific P&L	[illegible]
152	4.87	[illegible]	Appalachian Power	[illegible]
153	4.83	[illegible]	Portland GE	[illegible]
154	4.75	[illegible]	Pacific P&L	[illegible]
155	4.69	[illegible]	PS Indiana	[illegible]
156	4.61	[illegible]	Lincoln Elec. Mun.	[illegible]
157	4.57	[illegible]	Pacific P&L	[illegible]
158	4.52	[illegible]	Montana Power	[illegible]
159	4.27	[illegible]	Kentucky Utilities	[illegible]
160.	4.09	[illegible]	Tacoma DPU	[illegible]
161	3.95	[illegible]	Eugene Mun.	[illegible]
162	3.65	[illegible]	Idaho Power	[illegible]
163	3.34	[illegible]	Snohomish Util. Dist.	[illegible]
164	2.70	[illegible]	Seattle City Light	[illegible]

These prices represent what electric utilities charged commercial customers in June 1992, based on dividing total revenue by total kwh sold to these customers. These prices may not reflect what individual customers were charged. Prices include demand, power, and fuel adjustment charges. Where possible, special rate programs for large customers have been eliminated to more closely represent actual per kwh costs to most customers. Some utilities make arbitrary one-month adjustments in revenue tallies that do not correspond to what customers were billed. The cumulative kwh percentile tells what percent of the surveyed utilities' total kwh was sold by the individual utility and those above it ranking as more costly. Source: DOE form EIA-826 and EUN survey.

RANKING OF GAS UTILITY PRICES BY STATE — COMMERCIAL

Rank	State	Rank	State	Rank	State
1.	Hawaii	18	Massachusetts	35	Arkansas
2.	Connecticut	19	Maryland	36	Wisconsin
3	New Hampshire	20.	Tennessee	37	Missouri
4	West Virginia	21.	Oregon	38	Iowa
5.	Pennsylvania	22.	Indiana	39	Kentucky
6.	Vermont	23	Montana	40	South Dakota
7.	Arizona	24	Michigan	41.	Wyoming
8	Rhode Island	25	Idaho	42	Virginia
9	New York	26	North Dakota	43	Colorado
10.	Maine	27	North Carolina	44	Minnesota
11	Alabama	28	Louisiana	45	Texas
12	New Jersey	29	Ohio	46	Mississippi
13.	Georgia	30.	Nevada	47	California
14.	District of Columbia	31.	Washington	48	Nebraska
15.	Delaware	32.	Oklahoma	49	New Mexico
16	South Carolina	33.	Florida	50	Kansas
17.	Illinois	34.	Utah	51	Alaska

Prices are from May 1992. National Average: 4.59. Source: DOE Natural Gas Monthly. Numbers are subject to revision.

APPENDIX C

Payback Graphs For 2000, and 7500 hrs/yr of operation

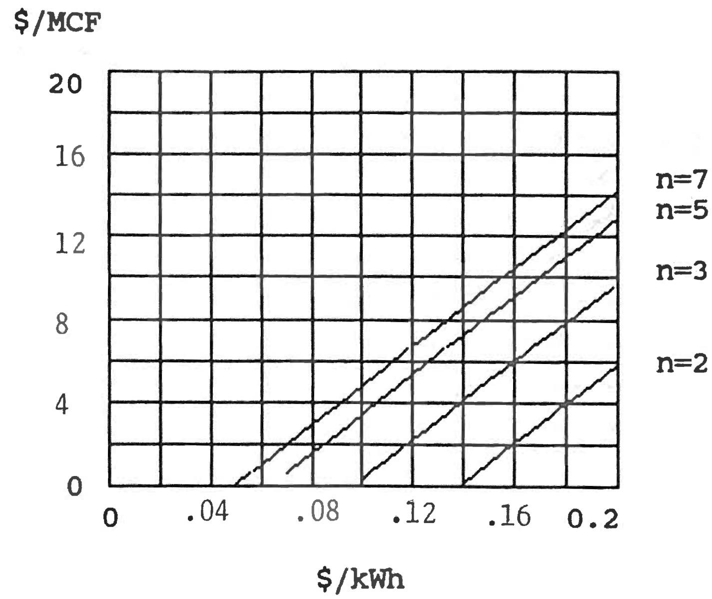

Fig. C1. Payback graph for case I[1], operating 2000 hrs/yr, with a load factor of 0.70, and with no waste heat recovery.

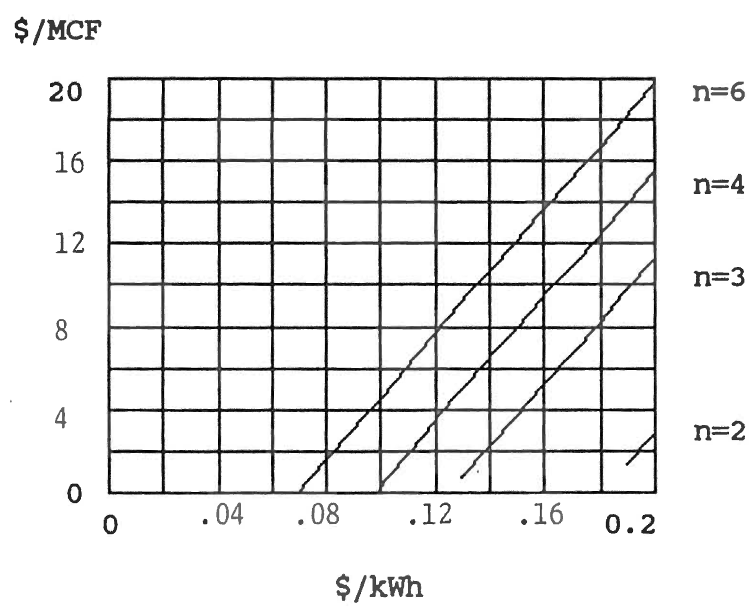

Fig. C2. Payback graph for case I[1], operating 2000 hrs/yr, with a load factor of 0.70, and with waste heat recovery.

1 - A 110 HP GEDC as an alternate for a 100 HP EMDC

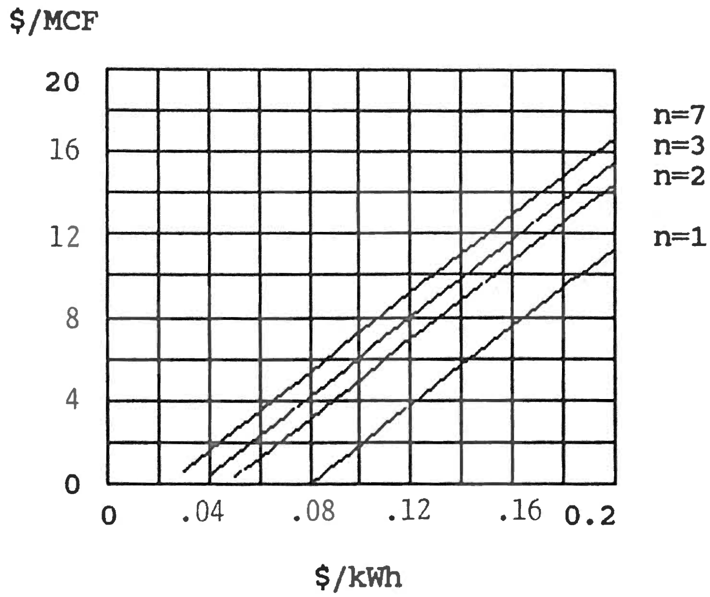

Fig. C3. Payback graph for case I[1], operating 7500 hrs/yr, with a load factor of 0.70, and with no waste heat recovery.

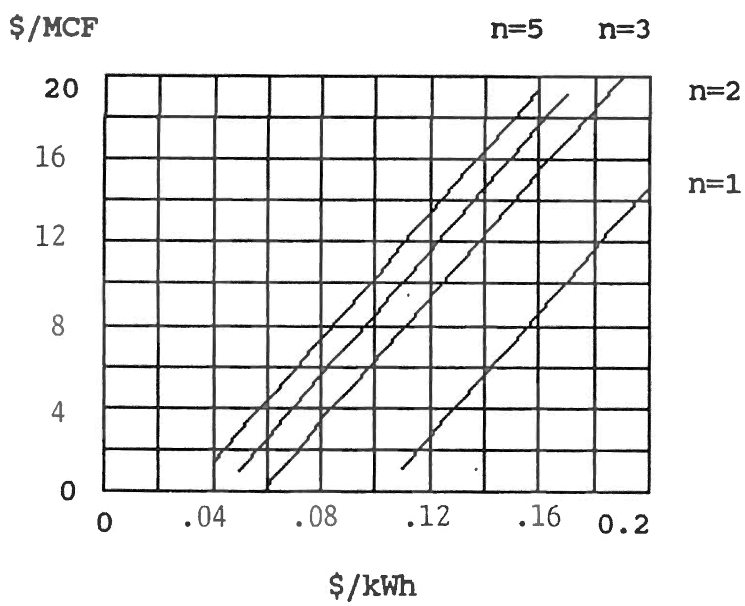

Fig. C4. Payback graph for case I[1], operating 7500 hrs/yr, with a load factor of 0.70, and with waste heat recovery.

1 - A 110 HP GEDC as an alternate for a 100 HP EMDC

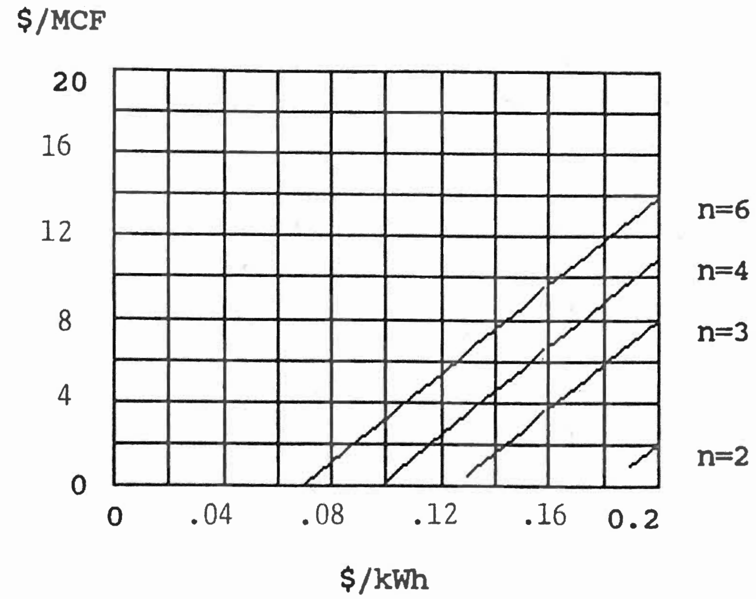

Fig. C5. Payback graph for case II[2], operating 2000 hrs/yr, with a load factor of 0.70, and with no waste heat recovery.

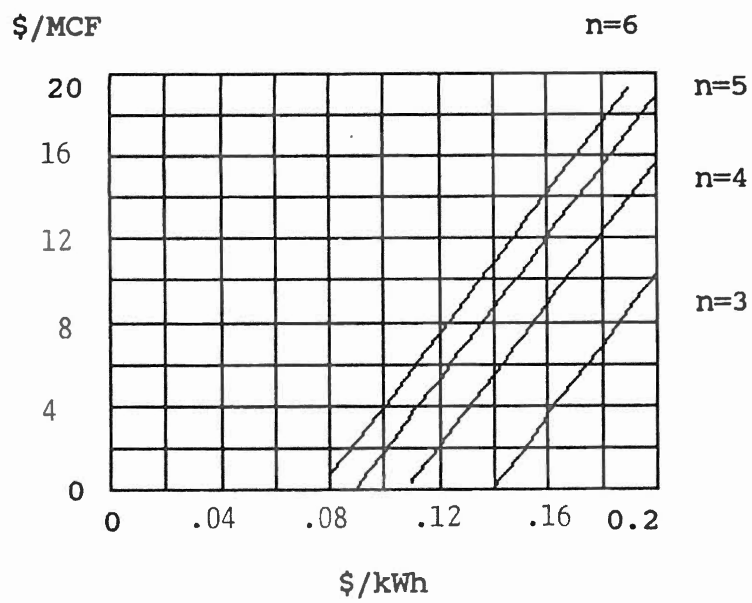

Fig. C6. Payback graph for case II[2], operating 2000 hrs/yr, with a load factor of 0.70, and with waste heat recovery.

2 - A 217 HP GEDC as an alternate for a 200 HP EMDC

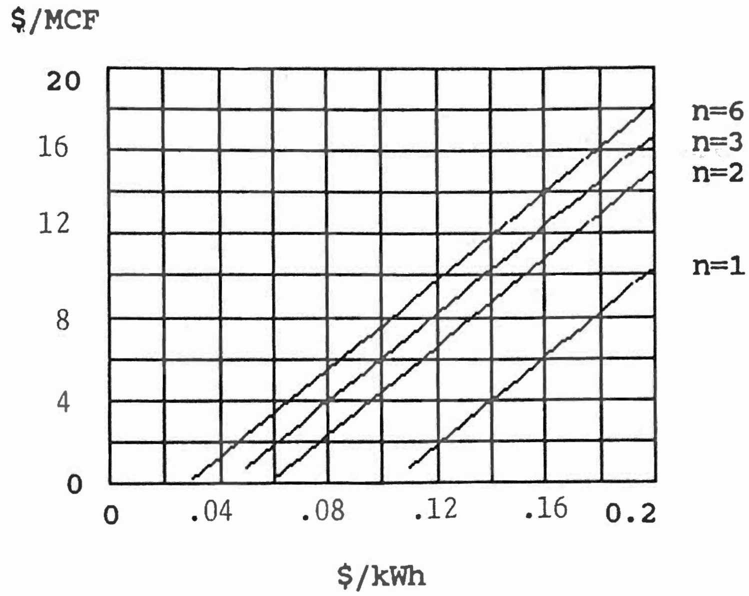

Fig. C7. Payback graph for case II[2], operating 7500 hrs/yr, with a load factor of 0.70, and with no waste heat recovery.

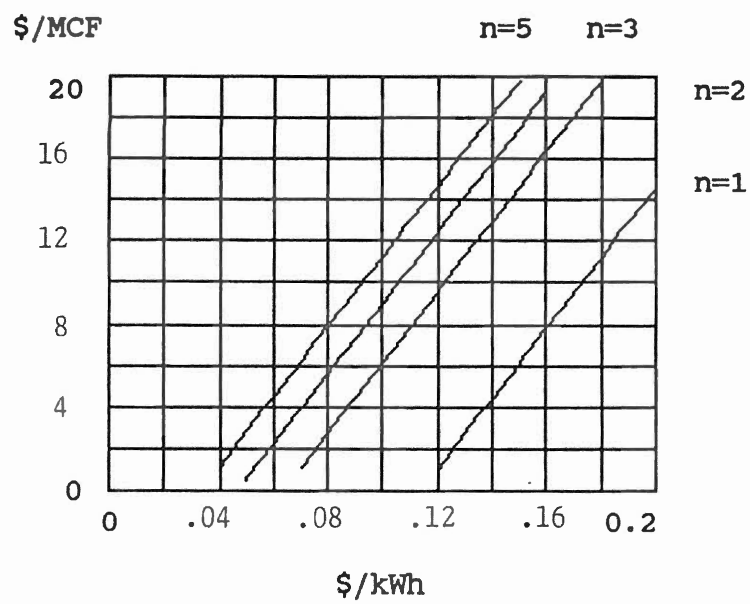

Fig. C8. Payback graph for case II[2], operating 7500 hrs/yr, with a load factor of 0.70, and with waste heat recovery.

2 - A 217 HP GEDC as an alternate for a 200 HP EMDC

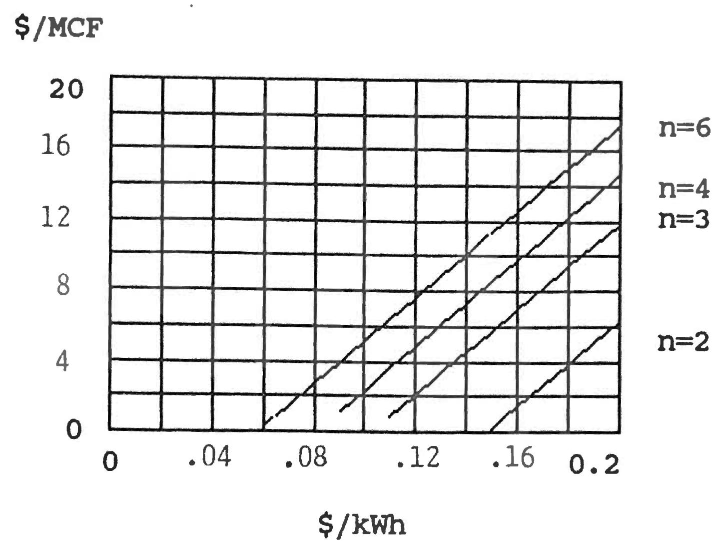

Fig. C9. Payback graph for case III[3], operating 2000 hrs/yr, with a load factor of 0.70, and with no waste heat recovery.

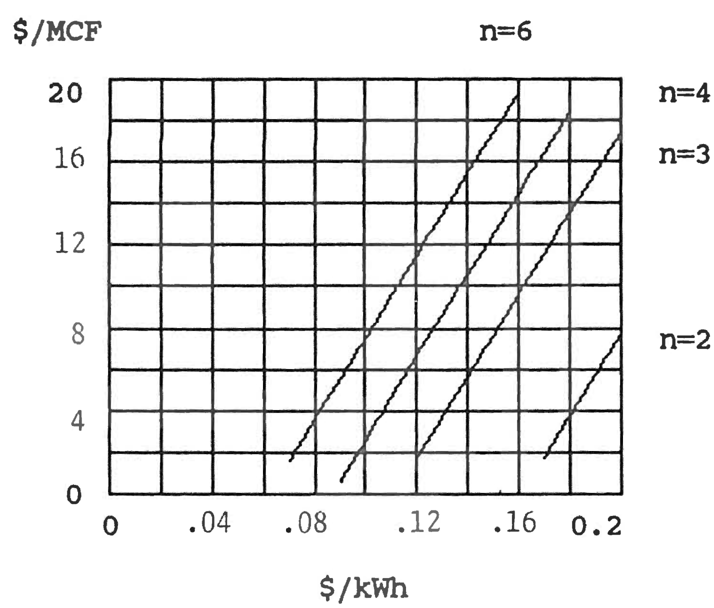

Fig. C10. Payback graph for case III[3], operating 2000 hrs/yr, with a load factor of 0.70, and with waste heat recovery.

3 - A 352 HP GEDC as an alternate for a 350 HP EMDC

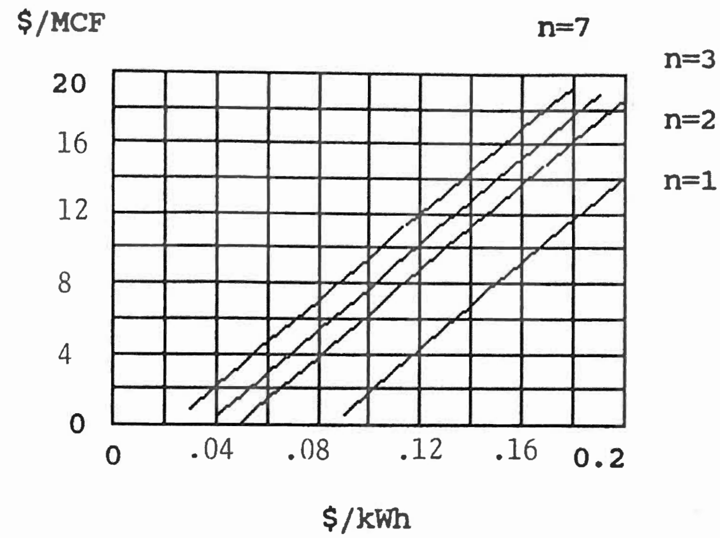

Fig. C11. Payback graph for case III[3], operating 7500 hrs/yr, with a load factor of 0.70, and with no waste heat recovery.

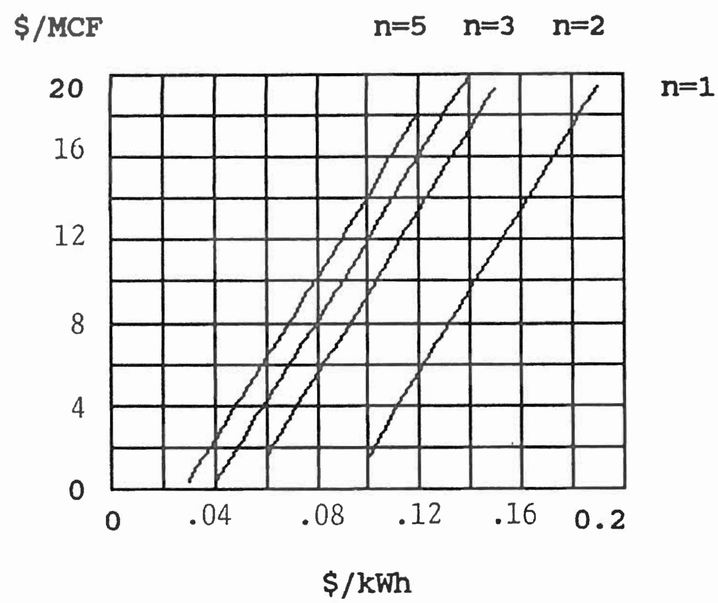

Fig. C12. Payback graph for case III[3], operating 7500 hrs/yr, with a load factor of 0.70, and with waste heat recovery.

3 - A 352 HP GEDC as an alternate for a 350 HP EMDC