

Experimental investigation of a Diaphragm FPSE design using a novel edge welded bellows-displacer assembly

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ABSTRACT

The energy conversion process of fossil fuels to electrical power is accomplished using inefficient technologies with negative consequences on human health and environment as well as contributing to depletion of finite natural resources. Therefore, there is currently increased societal pressure to adopt more environmentally sustainable and low carbon solutions. The share of energy derived from renewable sources (solar, wind, biomass, etc.) is projected to increase significantly over the next decades to decarbonise the energy sector. However, to achieve this objective, the heat conversion technologies need to make a step improvement in energy conversion. Stirling engine is considered to operate on the most efficient heat cycle capable of operating on a range of fuels including renewables.

This research investigated a novel design of a diaphragm Free-piston Stirling Engine for low temperature power generation in remote and inaccessible regions of the world. The design incorporates a flexible edge welded bellows material to support the displacer and a flat elastomer as a power piston. This mechanical arrangement of the moving parts of the engine eliminates air leakage, mechanical friction of the power piston and cylinder, and low spring losses. A proof-of-concept and a validated

mathematical model was developed as part of this project. The mathematical model was based on solving the energy, mass and momentum conservation equations of the working fluid in different parts of the engine. The performance of the engine was evaluated for different design parameters such as temperature, pressure, operating frequency, etc. A proof-of-concept prototype was built and tested under controlled laboratory settings to measure the energy performance. It was demonstrated the proof-of-concept engine can operate successfully at low to medium temperatures (up to 300°C) at atmospheric pressure and frequency of 16Hz. The tests also showed that under sufficient temperature gradient the engine is self-starting. Though the shaft power output was insignificant for the size of the engine, the design and laboratory results have contributed to advancing the technology and its application.

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Table of Contents

INTRODUCTION.....	1
1.1 BACKGROUND	1
1.2 RESEARCH OBJECTIVES	3
1.3 RESEARCH METHODOLOGY	4
1.4 THESIS STRUCTURE	5
REVIEW OF POWER GENERATION AND RENEWABLE TECHNOLOGIES	7
2.1 INTRODUCTION	7
2.2 CONVENTIONAL POWER GENERATION TECHNOLOGIES	9
2.2.1 Steam (Rankine) Cycle (RC)	10
2.2.2 Gas (Brayton) Cycle (GC)	12
2.2.3 Internal Combustion Engine – Reciprocating Engine.....	14
2.3 RENEWABLE POWER GENERATION TECHNOLOGIES	17
2.3.1 Concentrating Solar Power (CSP)	18
2.4 SUMMARY	30
REVIEW OF STIRLING ENGINE TECHNOLOGY	32
3.1 INTRODUCTION	32
3.2 PRINCIPLE OF STIRLING ENGINE	33
3.2.1 Displacer	35

3.2.2	<i>Power Piston</i>	36
3.2.3	<i>Heater</i>	36
3.2.4	<i>Cooler</i>	37
3.2.5	<i>Regenerator</i>	37
3.3	IDEAL STIRLING THERMODYNAMIC CYCLE	37
3.3.1	Isothermal Compression Process 1-2:	38
3.3.2	Constant Volume regenerative Process 2-3:.....	39
3.3.3.	Isothermal Expansion Process 3-4:	39
3.3.4	Constant volume regenerative process 4-1:	39
3.4	REAL STIRLING CYCLE	40
3.5	THE MECHANICAL ARRANGEMENTS AND CLASSIFICATION OF STIRLING MACHINES	42
3.5.1	Liquid-Piston Stirling Engines:.....	44
3.5.2	Kinematic Stirling Machines:	44
3.5.3	Free piston Stirling engine (FPSE)	46
3.5.4	FPSE Classification and Types	53
3.6	SUMMARY	60
STIRLING CYCLE ANALYSIS AND MODELING		61
4.1	INTRODUCTION	61
4.2	THERMODYNAMIC ANALYSIS.....	62
4.2.1	Zeroth Order (Simple Analysis)	62

4.2.2	The first-order model (Schmidt analysis)	64
4.2.3	Second Order Model (Adiabatic Decoupled Analysis)	76
4.2.4	Simple Analysis.....	83
4.2.4	Third-Order Model	85
4.3	DYNAMIC ANALYSIS	87
4.4	THE PROPOSED MECHANICAL ARRANGEMENT FOR THE FPSE.....	93
4.5	ENGINE DESIGN PARAMETERS	96
4.6	PROPOSED ENGINE MODELLING	98
4.7	COMPUTER MODEL RESULTS	98
4.8	SUMMARY	104
THE PROOF-OF-CONCEPT ENGINE DESIGN, FABRICATION AND ASSEMBLY		105
5.1	INTRODUCTION.....	105
5.2	PRELIMINARY DESIGN CONSIDERATIONS	106
5.3	MECHANICAL SPRINGS AND MATERIAL SELECTION	107
5.4	METAL BELLOWS	109
5.4.1	Bellow Configurations	110
5.4.2	Bellow Design Characteristics.....	113
5.4.3	Bellows Design and Sizing	114
5.4.4	Bellow Mounted Displacer in FPSE	117

5.5	DIAPHRAGMS	118
5.5.1	Diaphragm Design	119
4.5.2	The Diaphragm Material Properties	122
5.6	ENGINE DESIGN AND LAYOUT CONSIDERATIONS	123
5.6.1	Displacer and Power Piston Assembly	124
5.6.2	The Proposed Design Configurations	126
5.7	THE PROOF-OF-CONCEPT PROTOTYPE ENGINE	130
5.7.1	Prototype Components Design and Manufacturing	131
5.7.2	Prototype Assembly Procedure	143
5.8	SUMMARY	147
EXPERIMENTAL TESTING OF THE DIAPHRAGM FPSE		149
6.1	DESCRIPTION OF THE TEST RIG SETUP AND EXPERIMENTAL PROCEDURES	149
6.1.1	Thermocouples	150
6.1.2	Pressure Transducer	151
6.1.3	Liner Variable Differential Transformer (LVDT)	151
6.1.4	Data Logger.....	152
6.1.5	DC Power Supply.....	152
6.1.6	Oscilloscope.....	152
6.2	ENGINE START UP PROCEDURE	153
6.3	MODIFICATIONS AND REFINEMENT	154

6.4	DIAPHRAGM OPERATING FREQUENCY TUNING.....	159
6.5	STEADY STATE TEST RESULTS.....	162
6.5.1	Swept Volume and Pressure Variation	164
6.5.2	PV Diagram	165
6.6	SUMMARY	166
CONCLUSION AND RECOMMENDATIONS FOR FURTHER WORK.....		167
7.1	INTRODUCTION.....	167
7.2	NOVELTY OF THE RESEARCH AND CONTRIBUTION TO ADVANCING FPSE DESIGN	169
7.3	FUTURE WORK	170

List of figures

Figure 2.1: Global electricity generation by fuel [9]	9
Figure 2.2: The ideal Rankine cycle	11
Figure 2.3: Ideal Bryton cycle: (a) Open Brayton cycle (b) T-S diagram	13
Figure 2.4: IC engine for electrical power generation [24]	15
Figure 2.5: Renewable generation by regions in 2050 [26]	18
Figure 2.6: Solar thermal technologies for power generation [27]	19
Figure 2.7: PCT plant [31]	21
Figure 2.8: Schematic view of solar tower power plant [32]	22
Figure 2.9: Ivanpah solar tower power plant in the US [33]	23
Figure 2.10: Schematic view of CSP plant with Linear Fresnel reflectors [37]	24
Figure 2.11: Solar dish with Stirling engine [38]	25
Figure 2.12: The thermoelectric generator (TEG)	27
Figure 2.13: Organic Rankine Cycle (ORC) layout	28
Figure 2.14: Acoustic engine	29
Figure 3.1: Principle of the heat engine	34
Figure 3.2: The main parts of the Stirling engine	34
Figure 3.3: Ideal Stirling cycle processes	38
Figure 3.4: Real Stirling engine cycle (a) Motion profile of the displacer and power piston and (b) the P-V diagram	40
Figure 3.5: P-V diagram of the real Stirling cycle [58]	41
Figure 3.6: Stirling engine classification	43
Figure 3.7: Kinematic Stirling configurations	46
Figure 3.8: The relationship between load and the piston stroke	48
Figure 3.9: Simple diagram shows the main components of the FPSE	50
Figure 3.10: PV diagram of the FPSE [58]	52

Figure 3.11: Single acting two-free pistons configurations	55
Figure 3.12: The way the displacer is driven	57
Figure 3.13: Split-Free-piston parallel cylinders	58
Figure 3.14: Diaphragm free-piston Stirling engine.....	60
Figure 4.1: Beale number for a range of heater temperature	63
Figure 4.2: Ideal Isothermal Model	67
Figure 4.3: Control volume – generalized cell	69
Figure 4.4: Volume variations of compression and expansion spaces for single cycle of Alpha type [68].....	72
Figure 4.5: The ideal adiabatic model	78
Figure 4.6: Heat balance of the engine cells –simple analysis	83
Figure 4.7: Dynamic representation of FPSE	88
Figure 5.1: The proposed design of the FPSE	94
Figure 5.2: PV diagram	99
Figure 5.3: Pressure variation in the working spaces	100
Figure 5.4: Gas mass flow rate between the working spaces.....	102
Figure 5.5: Cyclic heat transferred within the engine cells	102
Figure 5.6: Temperature change in the engine cells with respect to the crank angle .	103
Figure 5.7: Pressure losses in the heat exchanger	104
Figure 6.1: The mechanical springs used in FPSEs	107
Figure 6.2: (a) Formed bellows (b) Edge-welded bellows.....	112
Figure 5.6: (a) axial bellow (b) Lateral bellow (c) angular bellow.....	113
Figure 6.4: Effect of resonance frequency on number of bellow convolutions	116
Figure 6.5: Schematic diagram of the proposed bellows	117
Figure 6.6: Swept volume approximation	118
Figure 6.7: Schematic of a deflected diaphragm under distributed pressure forces ...	119

Figure 6.8: FPSE displacer-power piston arrangements (a) Displacer spring to piston (b) displacer spring to the ground	125
Figure 6.9: Flat metal spring for displacer support.....	126
Figure 6.10: Initial design variant of the FPSE	127
Figure 6.11: Displacer assembly	128
Figure 6.12: Slide power piston.....	129
Figure 6.13: Diaphragm power piston configuration.....	130
Figure 6.14: Diaphragm-Bellow FPSE.....	131
Figure 6.15: Unassembled displacer parts	135
Figure 6.16: Partially assembled displacer	136
Figure 6.17: Displacer assembly (parts not fully tightened)	136
Figure 6.18: Assembled displacer block.....	137
Figure 6.19: Diaphragm configuration (a) diaphragm structure (b) clamping holes (c) clamping flanges	138
Figure 6.20: Sloped diaphragm clamping flange.....	139
Figure 6.21: The engine cylinder	140
Figure 6.22: Compression and expansion space connecting tube	141
Figure 6.23: Heat sources (a) electric ring heater (b) gas burner	142
Figure 6.24: Engine support frame	143
Figure 6.25: Displacer-cylinder assembly	144
Figure 6.26: Sloped flange insertion	145
Figure 6.27: Sloped flange insertion	146
Figure 6.28: Fully assembled proof-of-concept engine	147
Figure 7.1: Laboratory test rig	150
Figure 7.2: Pressure transducer	151
Figure 7.3: DT 500 Data logger.....	152
Figure 7.4: Engine cylinder hot end	155

Figure 7.5: Diaphragm power piston oscillation (a) $T_h=230^\circ\text{C}$ (b) $T_h=250^\circ\text{C}$	156
Figure 7.6: Modified working fluid connection tube location (a) expansion space (b) compression space	157
Figure 7.7: Diaphragm oscillation after modifications (a) $T_h=230^\circ\text{C}$ (b) $T_h=250^\circ\text{C}$	158
Figure 7.8: Initial sustained diaphragm and pressure oscillation (red line: stroke, yellow line: pressure, expansion space temperature $T_h=250^\circ\text{C}$)	159
Figure 7.9: Diaphragm resonance frequency adjustment (a) $m=100\text{ g}$ (b) $m=400\text{ g}$ (c) 500g	160
Figure 7.10: Diaphragm oscillations (freq.=15 Hz, $T_h=270^\circ\text{C}$ and $T_c=35^\circ\text{C}$)	161
Figure 7.11: Engine supporting frame	162
Figure 7.12: Diaphragm oscillation (freq.=16 Hz, $T_h = 300^\circ\text{C}$)	163
Figure 7.13: Diaphragm oscillation (freq.=16 Hz, $T_h = 350^\circ\text{C}$)	164
Figure 7.14: Pressure and volume variation	165
Figure 7.14: Engine PV diagram	166
Figure 8.1: The prototype engine adopted to linear electric generator	171

List of tables

Table 2.1: Solar thermal collectors' classifications [45]	20
Table 2.2: The characteristics of CSP power plants	26
Table 5.1: The approximated dimensions for the main engine parts.....	95
Table 5.2: Operating parameters defined by the user	96
Table 5.3: The numerical results from the computer analysis.....	97
Table 6.1: Bellow characteristics	113
Table 6.2: Natural rubber properties.....	122
Table 6.3: The displacer cylinder design specification.....	132
Table 6.4: The edge welded bellows design specification	134
Table 6.5: physical properties of the diaphragm	138
Table 7.1: The working conditions of the engine during the test.....	164

Nomenclature

Symbol	Description
A	cross sectional area, m^2
B_n	Beale number
C	damping coefficient, N/m
C_p	gas specific heat capacity at constant pressure, J/kg.k
C_v	gas specific heat capacity at constant volume, J/kg.k
E	Young's Modulus of the diaphragm material, Pa
f	frequency, Hz
g_A	mass flow rate
K	spring rate, N/m
K_p	diaphragm spring rate, N/m
M	mass (kg)
$\dot{m}$	mass flow rate (kg/s)
nw	number of convolutions
NTU	number of transfer unites
NST	Stanton number
P	Pressure, N/m^2
P_m	mean pressure, N/m^2
Q	heat transferred, W
R	gas constant, J/mol.K

$\dot{x}$	velocity, m/s
$\ddot{x}$	acceleration, m/s ²
T	temperature, °C
t	thickness, mm
V	volume, m ³
W	output power, W

Greek letter

ε	heat exchanger effectiveness
δ	bellow and diaphragm deflection, mm
λ	natural frequency, Hz
η	thermal efficiency
θ	cyclic angle, degree
μ	Poisson's ratio of the diaphragm
ρ	density, kg/m ³
φ	phase angle, degree

Subscripts

C	cooled temperature
diss	dissipated
H	hot temperature
c	Compression space
k	Cooler

r	Regenerator
h	Heater
e	Expansion space
cl	Clearance volume
sw	Swept volume
i, j	Space directions
i,o	in and out

CHAPTER 1:

INTRODUCTION

1.1 Background

Energy plays a vital role in every field of human activity and it is a significant factor in the development of successive civilisations. This has been reflected in the ever increasing global demand for energy as the population continues to grow, development of energy intensive industries, and increase in prosperity. However, the energy used to power modern living is almost derived from exhaustible fossil fuel reserves which are being depleted at a fast rate. Furthermore, burning these fossil fuels to generate useful energy for power generation, transportation and heat has led to huge quantities of carbon dioxide and other pollutants being released to atmosphere. This has in turn long lasting consequences on global warming, health issues and security of energy supplies.

Therefore, it has become urgent in many developed countries to set up targets in reducing carbon emissions through promoting energy efficient and sustainable technologies. This will require a concerted effort globally if a meaningful impact is going to be made. Leading in

the way, the UK for example, has set an ambitious target of reducing carbon emission by 80% by 2050 compared to 1990 emission level. The road map for achieving this target will require almost a fully decarbonised economy using particularly renewable energy sources, carbon capture and storage in traditional fossil fuelled power plants, developing energy storage technology, and promoting energy efficiency.

Globally, the population growth is expected to reach 10 to 11 billion people by 2050 [1-3], which will put extraordinary pressure on remaining fossil fuels reserves. The primary energy demand, which is approximately 13 terawatt, increases by approximately 1.2% per year, that represents about 36% of the world energy demand between 2008 and 2035 [4] and doubling by 2050. [1] [3] even though there are still over a quarter of the world population (1.5 billion people) have no access to electricity[5].

The vast majority, around 85%, of those people live in remote and rural areas, such as South Asia and sub-Saharan Africa [5, 6].

Therefore, developing sustainable, clean and affordable source of energy and low carbon technologies using renewables such as solar, wind, geothermal, biomass, tidal and wave energy is vital for future

mitigation of climate change, realising global stability and accelerating social and economic development.

It is in this context that this research project was undertaken to develop a simplified configuration of Stirling engine for small scale power generation in remote areas of the world using indigenous source of fuel such as agriculture waste and solar.

1.2 Research Objectives

The main objective of this research is to design, construct and test a simple free piston Stirling engine configuration using metal bellows and a flat diaphragm as a power piston. The specific aims were to:

- (i) Conduct a thorough literature review of power generation focusing on solar and clean technologies and define the research gap within the Stirling engine technology. The literature review covers power cycles of power generation technologies including conventional steam and gas cycles, Otto and Diesel cycle and Stirling cycle. The latter was considered in more details as is a highly efficient technology with potential application in remote parts of the world without access to electricity.
- (ii) Set up the design parameters such as heat source and heat sink temperatures, working fluid type, operating frequency and pressure.

- (iii) Analyse the thermodynamic cycle using a second order mathematical model and develop Matlab computer code.
- (iv) Specify the displacer supporting bellows and the diaphragm power piston parameters.
- (v) Construct and test a small demonstration prototype as a proof-of-concept.

1.3 Research Methodology

The research project investigated practical methods of constructing a free piston Stirling engine using metal bellows and rubber diaphragm for power piston. The research methodology considered in this work was both qualitative and quantitative. The qualitative aspect of the work involves understanding the theoretical concepts both using rule of thumb calculation and developing a comprehensive computer model using high level computer language (Matlab) to understand the Stirling cycle and simulate the proposed design of the Free Piston Stirling Engine. The quantitative part of the work requires material selection, concept outline, construction of a physical prototype and a lab test rig. The test rig was particularly important in that many modifications were required in the initial development stage. Measurements of various operating parameters using thermocouple for temperatures, pressure transducers for pressure variation within the engine, and oscilloscope for the sinusoid signal of the stroke of the moving parts.

1.4 Thesis Structure

The thesis is divided into 7 chapters:

Chapter 1: Introduction: brief background about energy generation and consumption is introduced, as well as, the objectives and the methodology of the research is pointed out.

Chapter 2: Review of power generation technology: in this chapter, both of the conventional and renewable power generation technologies are reviewed with more focus on thermal power generation technologies.

Chapter 3: Review of Stirling engine technology: The Stirling engine cycle, the engine main parts and the Stirling machine types and classifications are reviewed in order to broaden the knowledge of Stirling engine for power generation in general and in more particular Free-piston Stirling engine configuration.

Chapter 4: Stirling cycle computer model: Various thermodynamic Stirling engine cycle analysis methods are reviewed, then using one of the second order thermodynamic analysis methods, the Stirling cycle of the proposed design is modelled using Matlab to approximate the prototype dimensions and to specify the operating parameters.

Chapter 5: the proof-of-concept prototype design, fabrication and assembly: the characteristics and the calculations of the prototype components are defined. The prototype manufacture process is described. The prototype of diaphragm Stirling engine assembly and its outline is presented.

Chapter 6: experimental testing and results discussion: the test rig and the experimental procedure is described. Engine start up and modifications required are discussed. The validation of the results from the computer model are discussed and summarized.

Chapter 7: conclusion and recommendations for further work: this chapter introduce the findings of this research and the recommendations that could be beneficial for future work.

CHAPTER 2

REVIEW OF POWER GENERATION AND RENEWABLE TECHNOLOGIES

2.1 Introduction

Fossil fuels are the main energy source in the world and currently represent more than 80% of the overall energy consumption and are projected to remain the leading energy sources to 2035 [4, 7, 8]. The global demand for primary energy grows by approximately 1.2% per year on average, that would represent about 36% the world energy demand between 2008 and 2035 [4]. Referring to the International Energy Outlook IEO2016 [9], electricity as an end-use energy consumption is the fastest growing form of energy, while the global electricity generation is continuing to increase from 21.6 trillion kWh in 2012 to reach 25.8 trillion kWh by 2020 [9].

The world's demand for electricity has increased by 38% since 2003 [10] with an annual increase of almost 2.3% per year. This global demand for electricity is expected to continue to rise rapidly and more than any other final energy form, with about 80% of this growth is expected to be in developing countries [4]. Although there has been a

substantial share of electricity being generated from fuels such as natural gas that increased considerably after the 1980s and generation from nuclear power that increased significantly in the 1970s and 1980s. However, coal continues to be the most widely used fuel in conventional power plants accounting for 37% to 40% of the global fuel requirements for electricity generation [9, 11].

In order to meet the demand growth for electric energy, additional generation capacity of about 5,900 GW will be required by 2035 [12]. Recent concerns of environmental impact of greenhouse gas emissions have driven interest in generating power from renewable sources that release significantly less pollutants per generated kilowatt-hour. According to the IEO2016 [9] reference case, generation from natural gas, nuclear and renewable energy sources are continually improving, with renewables being the fastest growing source of energy for electricity generation. The world electricity generation from renewable sources accounted for 22% in 2012 and projected to increase up to 29% in 2040 in which the generation from non-hydropower renewables is the predominant source accounted for 5% of total world electricity generation in 2012 and predicted to increase up to 14% in 2040 [9]. The world net electricity generation by fuel is presented in Figure 2.1.

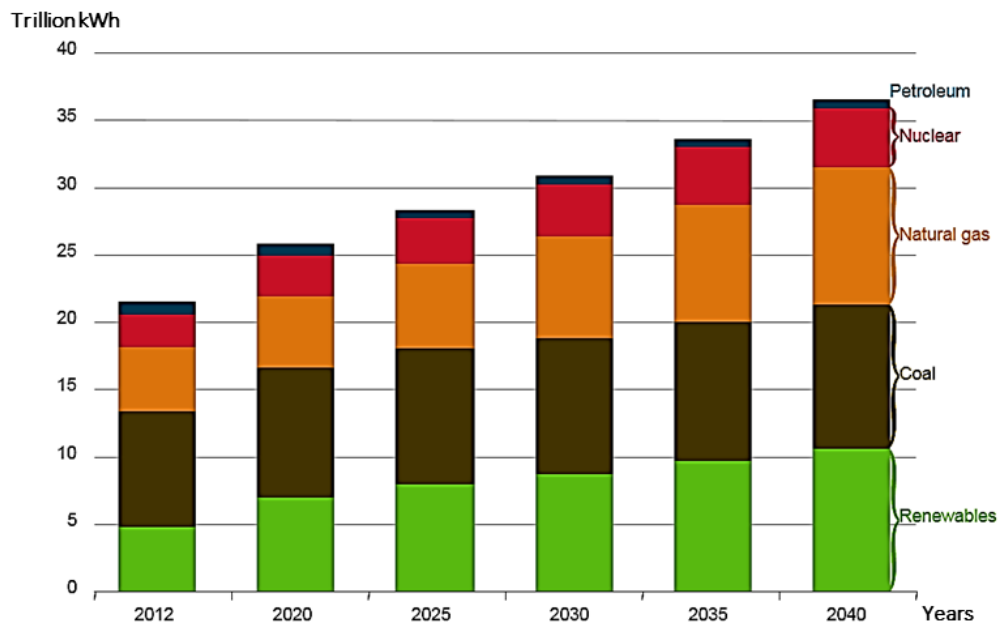


Figure 2.1: Global electricity generation by fuel [9]

2.2 Conventional Power Generation Technologies

Power plants which are fuelled by fossil fuels, such as gas-turbine (GT), steam-cycle and Combined Cycle (CC) generate approximately 66% of world electrical power [13]. However, about 80% of this power generation around the world is produced by conventional steam cycle [14]. Moreover, currently power generation plants are built using Combined cycle Gas turbine (CCGT) and supercritical steam cycle where the power plant operates at supercritical conditions of pressure and temperature (i.e., steam pressure higher than 22.06 MPa) to improve efficiency towards the 60% mark [15].

2.2.1 Steam (Rankine) Cycle (RC)

The steam turbine which operates according to Rankine thermodynamic cycle converts the energy from the superheated steam driving a steam turbine to mechanical work to generate electricity. The Rankine cycle is mainly used for power generation in conventional centralised power plants using heat sources such as nuclear, coal, natural gas or oil fuels. Today, steam turbine power stations account for 80% of the generated electric power worldwide [14].

A steam power plant consists of four main components, namely a steam boiler, a steam turbine, a condenser and a water circulating pump, as shown in Figure 2.2 (a).

The ideal Rankine cycle is a closed thermodynamic cycle in which water is vaporized at high pressure and temperature and expanded to low pressure through a steam turbine to produce mechanical work.

Figure 2.2 (b) shows the ideal thermodynamic processes shown on a T-s diagram that water undergoes to complete the Rankine cycle.

Process 1-2 is the isentropic compression of water in the boiler water feed pump from the condenser to the boiler pressure level. Process 2-3 is the heat addition in the boiler at constant pressure. At the exit of the boiler, the steam is in superheated state which is expanded isentropically in the turbine that is described by process 3-4 in Figure

2.2 (b) to convert thermal energy to mechanical energy. The steam at the turbine outlet is partly wet and is rejected to the condenser where heat is removed at constant pressure that is described in process 4-1 where the cycle is complete [15].

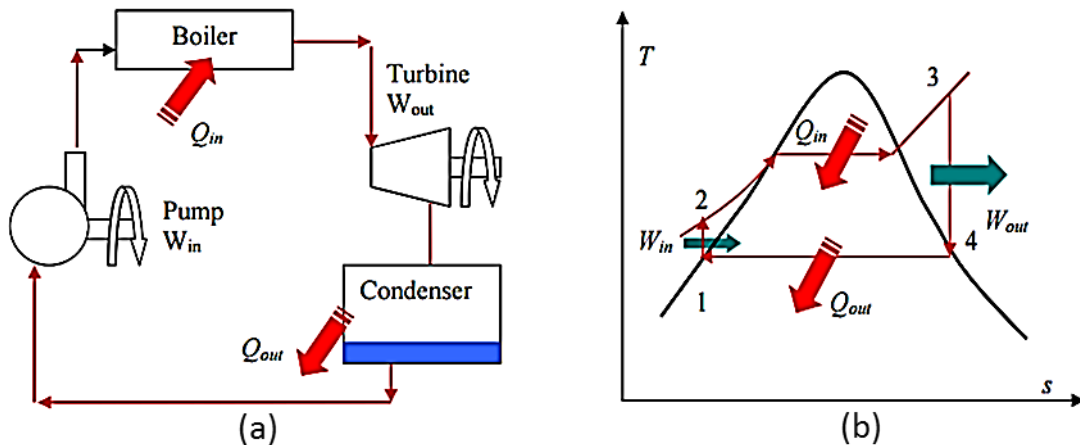


Figure 2.2: The ideal Rankine cycle

The thermal efficiency of Rankine cycle is up to 42% for modern power stations and its efficiency is mainly limited by the steam temperature and the operating pressure. The steam temperature at the inlet to the turbine is typically around 550 °C that gives efficiency up to 63% of Carnot efficiency. The main practical method of improving simple Rankine cycle efficiency is through steam reheating between expansion stages and the working fluid heat regeneration using steam trapped from hot side of the cycle to obtain the saturated liquid.[16].

The conventional heat sources have an advantage of producing energy at high density and high grade heat with typical temperatures of

approximately 565 °C which may be increased up to 700° C in modern supercritical cycles [14]. The power output capacity of steam power stations ranges from 100 to 1500 MW.

Current research in this area is focused on developing organic refrigerants with low boiler temperatures for power generation using for example waste heat and solar energy at low temperatures. This is referred to as Organic Rankine Cycle (ORC), which will be explored later in the renewable power generation technologies section.

2.2.2 Gas (Brayton) Cycle (GC)

Brayton cycle is used in gas turbine particularly in aeroplane engines for air transport and in stationary power generation. In its simple form, a compressor draws in fresh air at ambient conditions and compresses it into the combustion chamber at high pressure and temperature where the fuel is burned at constant pressure. The combustion products expand adiabatically in the turbine, producing mechanical power. The exhaust gases are released to the atmosphere in an open thermodynamic cycle. Figure 2.3 shows a simplified diagram of the main components of Brayton cycle, namely a compressor, a combustion chamber and a turbine and T-s diagrams of the thermodynamic cycle.

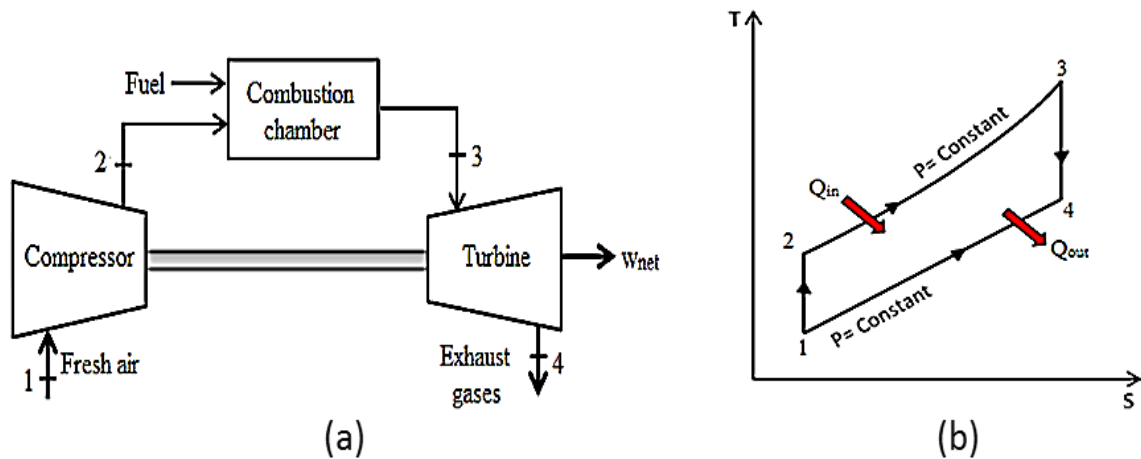


Figure 2.3: Ideal Bryton cycle: (a) Open Brayton cycle (b) T-S diagram

Compared to Rankine cycle, the Brayton cycle has lower thermal efficiency, particularly at low air pressure ratios. This is partly due to large amount of heat is rejected in the exhaust gases at the turbine outlet. The efficiency of the Brayton cycle is strongly influenced by pressure compression ratio and turbine inlet temperature.

Furthermore, modern gas turbines are equipped with heat recovery recuperator in which compressed air is preheated by exhaust gases that reduces the heat supply for combustion often referred to as regenerative Brayton cycle [17, 18]. In addition to the regenerative method of improving efficiency of the gas turbine cycles, other techniques are also used including reheating and intercooling. The gas turbine cycle is simple, flexible and reliable and is fast to start and is increasingly used to meet base load for power generation [19].

2.2.3 Internal Combustion Engine – Reciprocating Engine

Internal Combustion Engine (ICE) use reciprocating pistons to convert fuel energy into mechanical power. For convenience, the piston reciprocating motion is usually converted into rotary motion by a crankshaft mechanism. Reciprocating IC engines are split into two basic categories, spark-ignition (SI) and compression-ignition (CI) engine, which operates according to Otto and Diesel thermodynamic cycle engine respectively. They also can be of a two-stroke and four stroke engine [15, 20].

The development of IC, both spark ignition and compression ignition type, has been driven by the vast investment by the automobile industry with constant strive for high power density, efficiency and low emission [7]. The IC engine is also very popular for stationary applications and represents the largest share of the small power generation market and available in a range of applications including emergency and standby power, base-load power, peaking shaving power and combined heat and power (CHP). Importantly, IC engines are available in many sizes and power outputs, ranging from a few kilowatts to over 20 MW [21]. For example, the annual installed capacity of diesel gensets is project to reach 82 GW by 2018 [22].

Furthermore, concerns of dwindling conventional fuel sources and environmental pollution, concerted efforts are being directed to developing alternative bio-fuels derived from agriculture products. Current studies investigated that high octane number fuel such as ethanol is suitable for spark-ignition engine, whereas bio-fuel and vegetable oil are suitable for diesel engines [23]. Figure 2.4 illustrates a general layout of an IC engine for electrical generation.

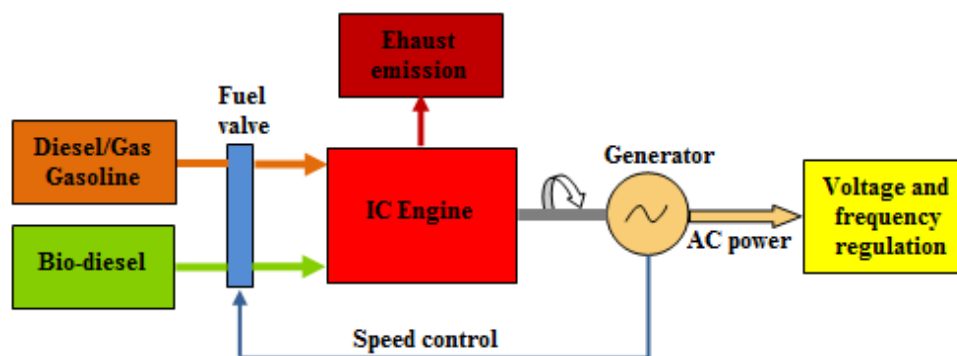


Figure 2.4: IC engine for electrical power generation [24]

There are several techniques and gas treatment processes that have been implemented particularly through better design, controlling the combustion process and using exhaust catalysts.

Spark ignition engines operate at high temperatures of up to 120° C [25] and consume either petrol (gasoline) or clean gaseous fuels such as natural gas. They release more heat to the cooling system and correspondingly less heat to the exhaust gases than diesel engines that will be introduced later. The efficiency of SI engines is relatively

low at around 35% [25] and the output shaft power limited to about 5 MW

Diesel engines consume a wide variety of fuels including the less combustible types, and turns more of the fuel's heat energy into work-shaft but rejects less wasted heat compared with SI engine.

Therefore, Diesel engines are more economic and efficient than SI engine.

Today, Diesel gensets remain the preferred power generators for isolated communities, stationary power applications and emergency and back-up power source for sensitive loads such as those found in hospitals, IT centres, banks and control rooms in power plants.

Moreover, Diesel prime mover has the advantages of quick starting .

Diesel generators require low initial investment costs, although their output power is expensive due to the high fuel prices and the associated fuel transportation costs, regular maintenance needs, and relatively low life cycle. However, electricity produced by diesel generator becomes even more costly when it runs under weak operation at partial load for a long time.

2.3 Renewable Power Generation Technologies

The energy extracted from natural replenished sources such as solar, wind, moving water (hydro power), geothermal and biomass are clean and renewable. Concerns over energy security because of the unpredictable fuel prices and the negative environmental impact of greenhouse gases emission have led many countries to invest heavily in renewable and low carbon energy technologies. These include solar, wind, hydro, geothermal and biomass as well as clean coal technologies such as carbon capture and storage. Solar power (particularly photovoltaic (PV) technologies) as well as wind power have seen an extraordinary growth, with hundreds of Gigawatts of new capacities have been installed in recent few years because of the drop in costs and new policies that address market development. Figure 2.5 highlights the future of renewable generation according to the analysis of International Energy Agency. The percentage shown above the bars represents the contribution of renewable energy technologies over the total electricity generation.

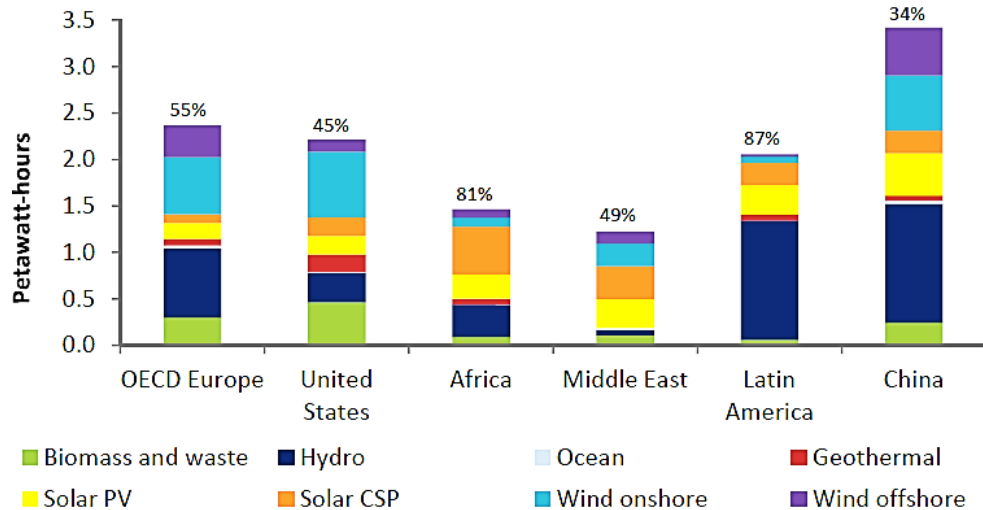


Figure 2.5: Renewable generation by regions in 2050 [26]

In the following sub-sectors, renewable technologies for electric power generation will be reviewed. However, the technologies used in conjunction with the large scale steam cycle power generation will be introduced in more details.

2.3.1 Concentrating Solar Power (CSP)

Conversion of solar energy to electrical power in a solar concentrating power plant is accomplished through the installation of an array of concentrating mirrors with or without tracking systems to concentrate sunlight onto a heat generator that it can be used to generate electricity.

CSP technologies are feasible in countries with abundant sunshine and clear skies like Southern Europe, some regions of North America, the Middle East and North Africa.

The advantage of CSP is that it can integrate thermal energy storage (TES) to extend the operation hours of the plant even though this tends to increase the overall cost of the plant. The most used technologies for concentrating solar energy are parabolic trough, linear Fresnel lens, solar dish and solar tower, as shown in Figure 2.6.

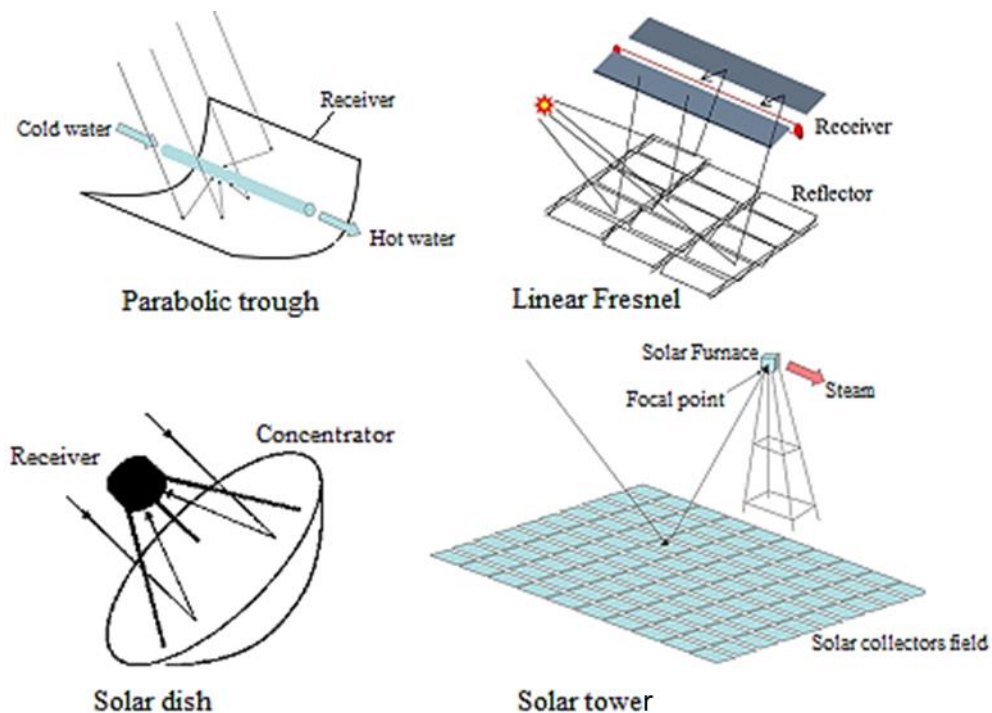


Figure 2.6: Solar thermal technologies for power generation [27]

Common to all these systems is the concentration of Direct Normal Irradiation (DNI) onto a focal line or point to heat up a working fluid to generate steam. The main characteristic of each is given in Table 2.1.

Table 2.1: Solar thermal collectors' classifications [45]

Technology	Operational Temperature range (°C)	Tracking mechanism
Parabolic trough	70-400	One-axis
solar tower	500-800	Two-axis
Fresnel reflector	100-400	One-axis
Solar dish	500-1200	Two-axis

2.3.1.1 Parabolic Trough CSP

Parabolic trough collectors (PTC) are the most mature technology and lowest cost among the CSP technologies that attracts the greatest interest. PTCs are used for power generation have a geometrical concentrating ratio of between 20 to 30 and operating temperature up to 580°C [28, 29] [30].

Figure 2.7 shows a schematic of a PCT power plant where the solar field consists of parabolic reflectors that concentrate solar energy onto a fluid carrying focal line receiver, a stainless steel pipe with solar radiation absorbing selective coating. The power generation block is made up of a steam turbine cycle with a dry cooled condenser.

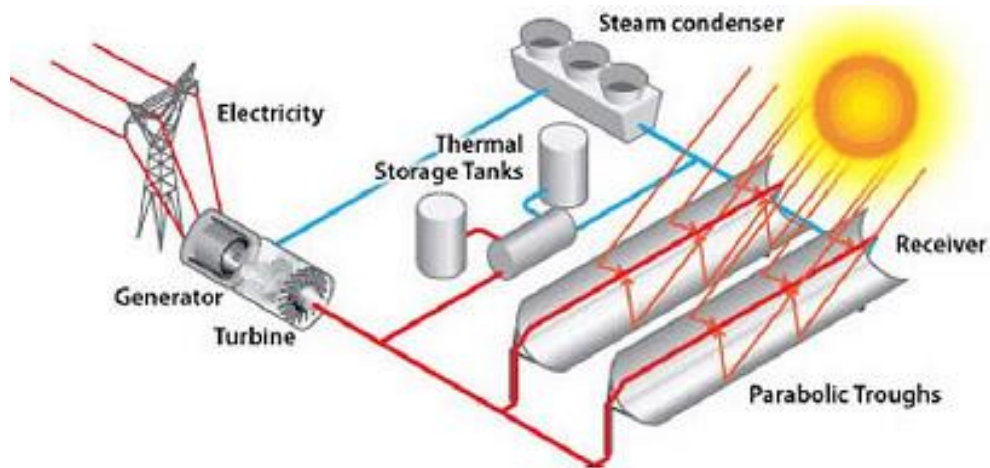


Figure 2.7: PCT plant [31]

The average annual conversion efficiency of a PCT plant is around 13 to 15% [1, 30, 31].

2.3.1.2 Solar Tower

In a solar tower plant, the solar field is made up of flat heliostats with two axes tracking to concentrate sunrays onto a receiver at the top of tower at the centre of the field. Solar towers produce a superheated steam directly as a result of the high temperature that can be achieved in the receiver of up to 1200°C [27, 32]. Figure 2.8 illustrates a schematic view of the solar tower power plant.

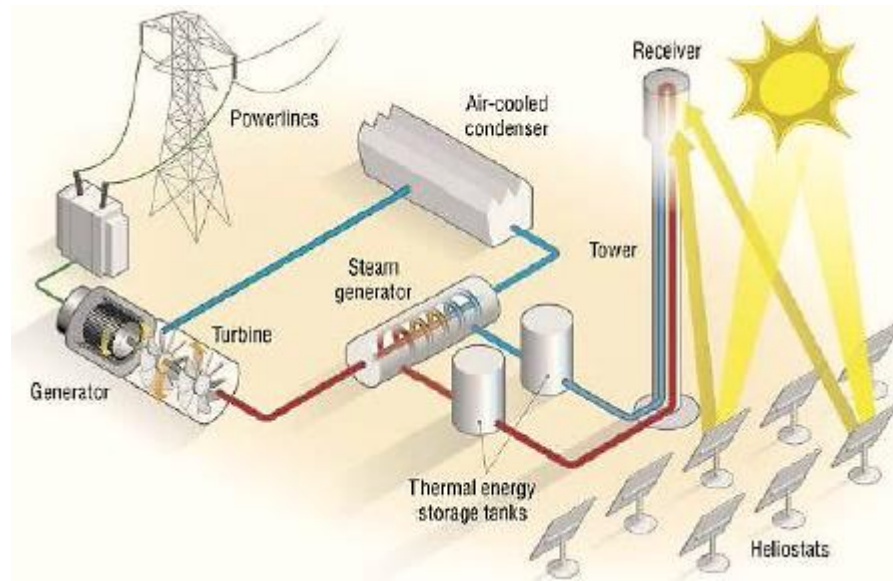


Figure 2.8: Schematic view of solar tower power plant [32]

The solar tower (ST) technology is preferable for direct steam generation for high range temperature of 500 to 1000 °C. However, the initial cost of the ST is high in addition to its design complexity and the large land occupancy requirements [32]. Solar tower accounts for about 10% of the total number of the installed CSP plants around the world including approximately 30% still under construction for both direct steam generation (DSG). The share from the ST generation is close to a capacity of 500MW with an expected capacity of 542 MW. The Ivanpah power plant showed in Figure 2.9 is considered to be one of the largest solar tower plants in the world.



Figure 2.9: Ivanpah solar tower power plant in the US [33]

2.3.1.3 Linear Fresnel CSP

There are two types of Fresnel collector, linear Fresnel reflector (LFR) and the Fresnel lens collector (FLC) [34]. Linear Fresnel reflectors (LFRs) are flat or slightly curved mirrors but they are similar to parabolic troughs in their operation concept. The optical system (mirrors) concentrates the solar radiation onto receivers comprising of a cylindrical parabolic tube at their focus line. Consequently, the working fluid of water is heated up to 450°C and hence converted to saturated steam. The steam is transferred under pressure of around 70 bar to drive a steam turbine for electricity generation [32, 35]. The first commercial LFR plant in the world was built in Spain by the German company Novatec Biosol. The plant has a capacity of 1.4 MW and was connected to the local electricity grid in 2009 [34]. The LFRs produce temperatures of 250 – 390 °C with an annual average

conversion efficiency of 14 – 18% and thermal efficiency of up to 30% [33, 34, 36].

Figure 2.10 illustrates the general layout of the integration of LFCs a field to produce steam for a steam turbine utilising the Rankine power cycle.

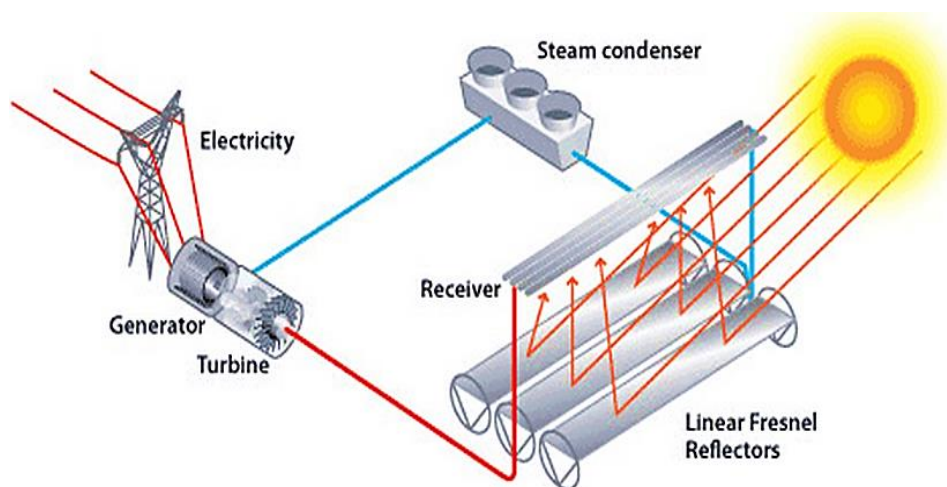


Figure 2.10: Schematic view of CSP plant with Linear Fresnel reflectors [37]

2.3.1.4 Solar Dish (Parabolic Solar Dish PSD)

A Parabolic dish concentrates solar radiation into a receiver at the focal point above the centre of the dish, as shown in Figure 2.11. The solar dish has a heat engine (Stirling engine) mounted at its focal point.

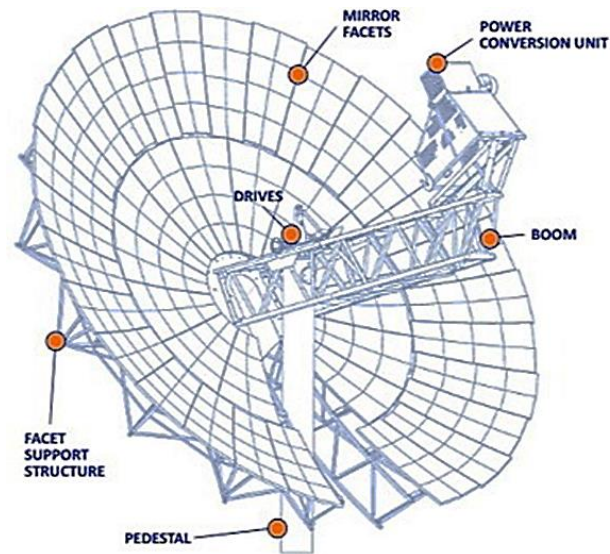


Figure 2.11: Solar dish with Stirling engine [38]

The Stirling engine mounted dish tracks the sun in two axes. The engine dish is used in a small scale system with a typical power output capacity of around 25 kw [33, 38]. However, multiple units can be used to form a solar power generation field.

A summary of the main characteristics of solar concentrated power plants is given in Table 2.2:

Table 2.2: The characteristics of CSP power plants

Property	PTC	LFR	ST	PDC
Maturity	High	Medium	Medium	Low
Operating temperature (°C)	≤380	≤450	≥450	≥580
Plant efficiency (%)	13–15	9–13	14–18	22–
Typical capacity (MW)	100	50	50–100	3–30
Land occupancy	Large	Medium	Medium	Low

2.3.1.5 Other Power Generation Technologies

There is a whole host of other small scale power generation technologies that can be found mostly in specific and niche applications.

i) Thermoelectric Generator (TEG)

Thermoelectric generator (TEG) showed schematically in Figure 2.12 are devices that convert thermal energy directly into electrical energy. They are made of a semi-conductor material and operate on the principle of the Seebeck effect where on application of a temperature gradient between a p-n junction an electric voltage is established. The device have many advantages including no moving parts, silent operation and no maintenance. However, the thermal energy to electrical power conversion efficiency is very low (3-5%).

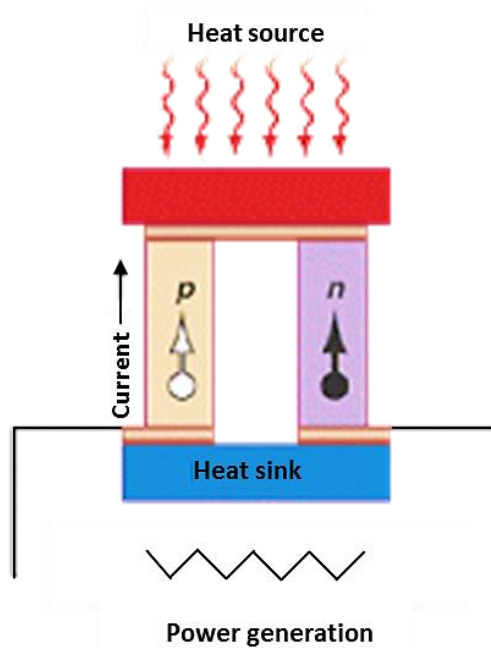


Figure 2.12: The thermoelectric generator (TEG)

ii) The Organic Rankine Cycle (ORC)

The ORC operates on the same principle as conventional Rankine cycle with the working fluid being of low boiling temperature. The selection of the working medium is very important in organic Rankine cycle since it influences the power plant performance, design and cost. A wide range of advanced organic fluids such as n-pentane, benzene, ammonia, trichlorofluoromethane and dichlorodifluoro-methane have been trialled for ORC technology [39]. These fluids require less heat for the evaporation as the evaporation process occurs at low pressure and temperature (below 300°C). The main desirable characteristics of the fluid are thermal and chemical stability, high latent heat, high density, high specific heat, low condensing and evaporating pressure and low environmental impact.

ORC base systems are increasingly sought for power generation in remote and isolated areas with heat sources from biomass, waste heat recovery from industrial processes, geothermal and solar collectors. These systems however have relatively low thermal efficiency of 5 to 10% [40-43].

Figure 2.13 shows an example schematic of a solar concentrator driven ORC.

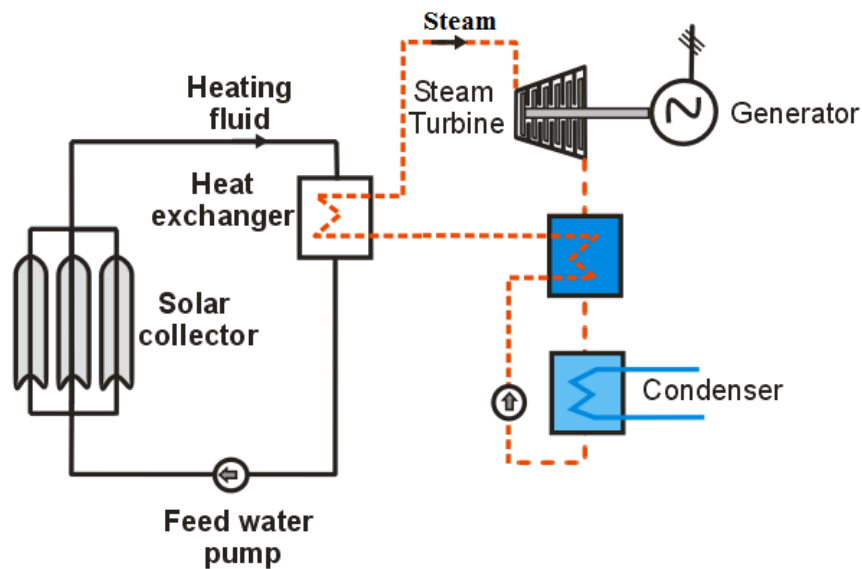


Figure 2.13: Organic Rankine Cycle (ORC) layout

iii) Thermo-acoustic Engines

The thermo-acoustic phenomenon was invented in the early 1950s by Kaiser. However, the first thermoacoustic machine was demonstrated by Yazaki [44-46]. The engine converts thermal energy into sound waves without moving parts except the oscillating working fluid. Both ends of the regenerator are connected to appropriate acoustic transfer

tubes where the working medium is by the acoustic wave displaced through the regenerator [47, 48]. The distinguishing part in the engine is the stack where the acoustic power is generated by the difference in temperature between its ends. The stack is made of porous material and located inside the resonator as shown in Figure 2.14 [49].

Thermo-acoustic Stirling engines have low efficiency (20%) limits their commercial application [50].

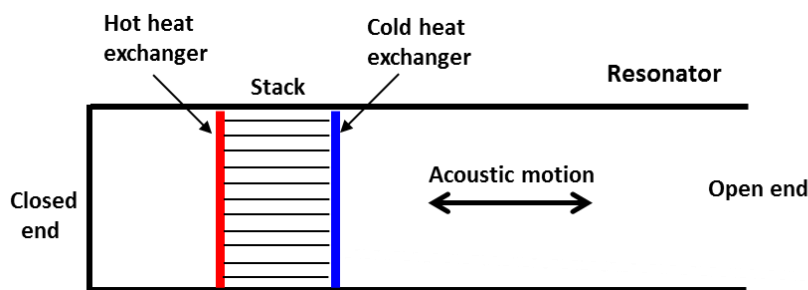


Figure 2.14: Acoustic engine

iv) Stirling Engine

Stirling engine is considered as the most efficient low-power range thermal converter [51] and is particularly attractive for small and remote power generation applications. In developing countries fuels such as diesel and bottled gas are often expensive and difficult to obtain and Stirling engine could be used to run on biomass fuel or solar energy. In developed countries it can be used as power generator in leisure and camping sites, and for micro-generation in houses [47, 51, 52].

Many configurations of Stirling engine have been successfully demonstrated since Robert Stirling's first design in 1816. These will be covered in detail in the next chapter.

2.4 Summary

Power cycles for heat engines with more focus on sustainable technologies for small scale power generation are reviewed in order to improve heat engine working on Stirling cycle appropriate for small-scale power generations applications operating at low temperature, low pressure and low frequency. The recent undergoing research devoted to develop Free Piston Stirling Engine as it's considered to be as one of promising technologies for small scale sustainable regenerative power generation applications. Thermal efficiency of the FPSE particularly for those are working at low pressure and low temperature are affected by some factors such as mechanical and gas hysteresis due to the interface friction and damping losses, air leakage between the working spaces and inefficient heat transfer. In this research a flexible metal bellow is proposed to be incorporated into a simple design of a Free Piston Stirling Engine configuration operating at low temperature, low pressure and low frequency is proposed for small-scale power generation in remote areas of developing countries. Using the metal bellow instead of the conventional mechanical springs or gas springs will facilitate separating working chambers so that the air

leakage will be eliminated. The bellow will replace the conventional mechanical springs, therefore, the hysteresis and imbalance of using the mechanical springs could be avoided. The bellow in terms of its shape is more suitable to form gas housing and improve heat exchange and regeneration.

CHAPTER 3

REVIEW OF STIRLING ENGINE TECHNOLOGY

3.1 Introduction

The Stirling engine is an external combustion heat engine that operates on closed regenerative thermodynamic cycle where a working fluid in a gas form undergoes a compression and expansion process at two temperature levels thereby converting thermal energy into mechanical energy. The Stirling engine is distinguished among the heat engines by its high ideal theoretical efficiency that approaches the Carnot cycle efficiency. Even though the concept of Stirling engine was invented by Robert and his brother James Stirling back in 1816 [53], its initial success was reversed by the advent of the internal combustion engine and its application is limited. However, the advantage of high efficiency, the flexibility of operating on a wide range of heat energy sources and the low requirements for maintenance has attracted renewed interests in the technology. In addition, the Stirling engine operate more quietly and has low vibration as fuel is burned continuously in the combustion chamber.

The Stirling engine has found applications in marine vessels, power units in space missions, combined heat and power, and cryogenic cooling [54, 55].

The main limitations of the Stirling engine that have hampered its competitiveness include [55] low power to weight ratio, slow thermal response to load variation and continuous exposure of the hot side material to high temperature, requiring expensive metal alloys.

3.2 Principle of Stirling Engine

The basic concept of Stirling engine operation is similar to that of other heat engines in that a portion of the thermal energy supply from a high temperature heat source is converted to mechanical energy and the remaining energy is reject in lower temperature heat sink. Figure 3.2 shows a simplified schematic of a heat engine operating between two temperature levels and producing work [53, 56].

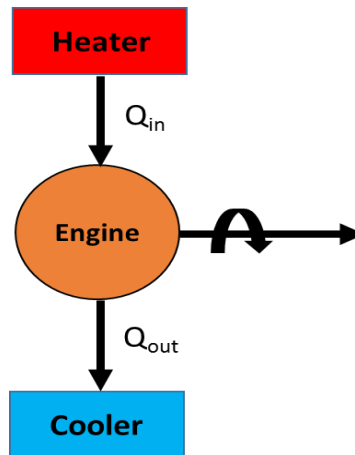


Figure 3.1: Principle of the heat engine

In its simplest forms, a Stirling engine is composed of two moving pistons, two heat exchanges and a thermal regenerator. Figure 3.2 shows a diagram of a Stirling engine where it is shown that the two pistons, a displacer at the hot end and a power piston at the cold end, are housed in the same cylinder.

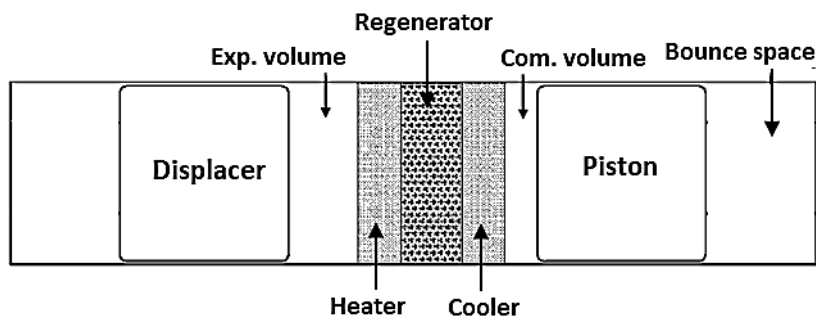


Figure 3.2: The main parts of the Stirling engine

The role and function of each component is described below.

3.2.1 Displacer

It is a lightweight piston that reciprocates between the compression and expansion spaces shuttling the working fluid, back and forth through the heat exchangers (cooler, heater and regenerator). The displacer serves to move the working fluid and hence consumes little mechanical power of the engine. The pressure difference (ΔP) between the displacer ends is due the working fluid viscous losses through the regenerator. The displacer is subject to high temperature gradient (ΔT) as it separates the expansion space at high temperature and the cold space at low temperature. To minimise heat conduction, the displacer is typically constructed in a ratio of its length to diameter of 2.5:1 with additional inner baffles to minimise heat radiation [57].

Other forms of displacer designs have also been produced by filling the displacer void with thermally insulating lightweight foam materials. In high pressure Stirling engine, a small orifice (relief vent) is drilled to link the inner volume of the displacer with the working space that allow the working fluid to flow in and out of the displacer in order to maintain the average pressure inside the displacer close to that of the working space. To prevent the working fluid by-passing the heat exchangers, a seal between the displacer and the cylinder wall is used. In small selected engines, the regenerator is included in the displacer,

so there is no need for the seals and the working fluid passes through the displacer annulus formed with the housing cylinder [58].

3.2.2 Power Piston

The power piston is the heavier of the two pistons. It is particularly subject to high pressure difference (ΔP) and low differential temperature (ΔT). The working temperature for the power piston is close to that of the heat sink. The power piston converts the engine pressure fluctuation into a reciprocating movement, which in turn drives a load. The engine working fluid is sealed using piston sealing rings between the piston and the cylinder wall, though some form of lubrication may be required to prevent friction [58].

3.2.3 Heater

The heater is the part of the engine through which all heat released from the fuel is transferred to the enclosed working fluid in the engine. In small engines, the heater is simply the hot end of the cylinder and the surrounding area (a gap heat exchanger) whereas in large units a larger heat transfer area will be required for sufficient heat transfer. In the latter, a complex bundle of tubes is often used and located within the combustion chamber [53, 59, 60]. One of the limitations of Stirling engines is that the heat must be transferred by conduction through the heat exchanger wall, creating high thermal inertia [59, 61].

3.2.4 Cooler

Similar to the heater, the cooler is a heat exchanger through which heat is rejected from the engine once the work is done on the power piston. This could be a simple air cooled fins around the compression space of a bundle of tubes immersed in a water-cooling jacket [53, 62].

3.2.5 Regenerator

The regenerator is often made of a matrix of metal wires or strips. In high performance Stirling engines the regenerator is the form of a sintered structure to store heat as the working fluid is moved from the expansion to the compression space and release an equivalent heat in the reverse flow. The regenerator is usually specified by its porosity. Hence, an effective regenerator needs to have a good compromise between porosity, void volume and pressure drop [53, 62].

3.3 Ideal Stirling Thermodynamic Cycle

The ideal Stirling thermodynamic cycle is considered as a reversible cycle in which all heat transfer processes are assumed to be reversible, all heat loss through friction, viscous and in the regeneration process are neglected, the power piston and displacer motion is discontinuous to achieve a prescribed working fluid distribution and all dead volumes are omitted [57]. Referring to Figure 3.3, which shows the ideal

Stirling cycle PV diagram and the movement of the pistons, the cycle is made up of the following thermodynamic processes.

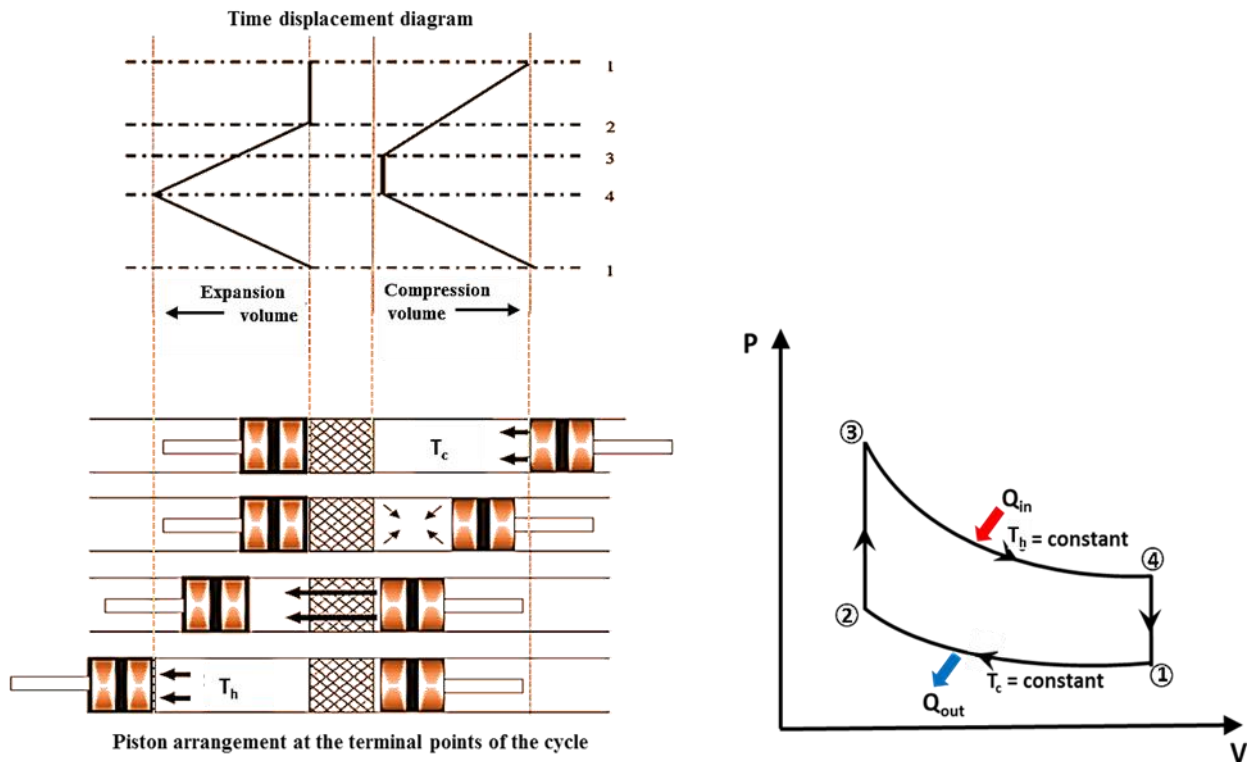


Figure 3.3: Ideal Stirling cycle processes

3.3.1 Isothermal Compression Process 1-2:

All working fluid is assumed to be initially in the compression space at its maximum volume V_1 and minimum pressure and temperature P_1 and T_c respectively. The working fluid is compressed isothermally at the heat sink temperature T_c with heat rejection to the surrounding. During this process, the displacer is at the top end of the cylinder while the power piston compresses the working fluid from pressure P_1 , to P_2 .

3.3.2 Constant Volume regenerative Process 2-3:

The working fluid is moved swiftly from the compression space to the expansion space and the heat transferred from the regenerator to the working fluid at constant volume. The movement of the displacer from the hot end to cold end displaces all the working fluid through the heat exchangers into the expansion space, where the working fluid temperature and pressure increased to T_h and P_2 , respectively.

3.3.3. Isothermal Expansion Process 3-4:

The working fluid is expanding isothermally at the high temperature T_h while heat is added to the working fluid in the heater. The expansion of the working fluid does work on the power piston, which reverses its direction and starts a power stroke. At the end of the stroke, the working fluid occupies the maximum volume, V_4 .

3.3.4 Constant volume regenerative process 4-1:

The working fluid is moved from the expansion space to the compression space with both pistons moving in tandem so that the volume remains constant. The working fluid temperature and pressure decrease to their lowest point, T_c , and P_4 respectively. The working fluid transfers heat to the regenerator on its way back to the compression space.

3.4 Real Stirling Cycle

In a real Stirling engine, most of the assumptions of the ideal cycle are not possible as the power piston and the displacer motion is continuous. This makes the boundaries of the thermodynamic processes not clearly distinguishable and represented by a smooth elliptical shape diagram as shown in figure 3.4 (a) and 3.4 (b).

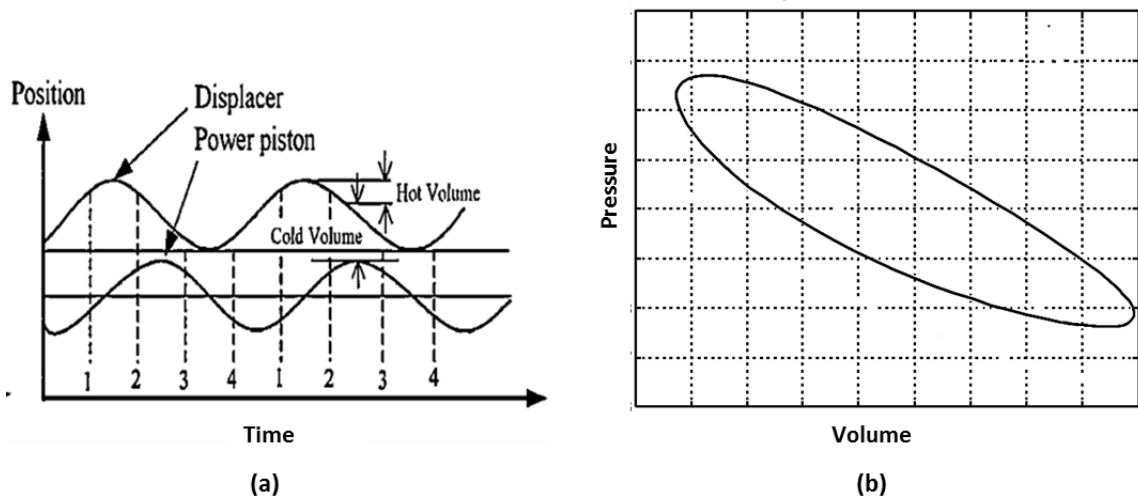


Figure 3.4: Real Stirling engine cycle (a) Motion profile of the displacer and power piston and (b) the P-V diagram

Evaluation of the performance of the Stirling engine is usually accomplished by quantifying the mechanical power delivered by the power piston and displacer as the working fluid is compressed and expanded. The combination of the work done on/by the power piston and displacer results in the actual power output.

Figure 3.5 shows the PV diagram of an actual Stirling engine. It can be seen that the area of the PV diagram of the compression space in

Figure 3.5 (a) is markedly larger than that of the expansion space (b). It is also noticeable that the presentation of the PV diagram in the compression and expansion cases do not take into account the dead space. Figure 3.5 (c) represents the PV diagram of the actual cycle and also shows the dead space volume as the distance from the origin of x-axis to the closest point on the diagram [52, 53, 57, 63].

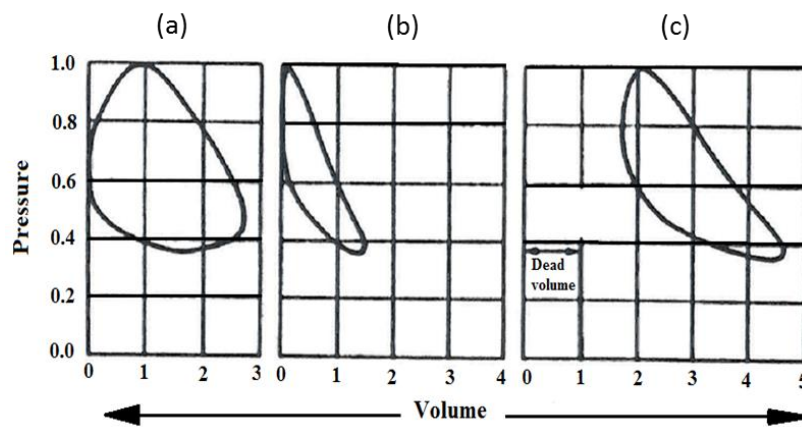


Figure 3.5: P-V diagram of the real Stirling cycle [58].

In the ideal cycle where the compression and the expansion processes are isothermal and the friction losses are neglected, the total area enclosed by the PV diagram equals to the difference between the area of the PV diagram of the expansion and compression spaces. However, in practice such equivalence is not attainable due to the aerodynamic flow losses in the heat exchangers causing instantaneous pressure differences between the compression and expansion spaces. This leads to flow losses which decrease the aggregated PV diagram enclosed

area and reduces the overall cycle power output and thermal efficiency [57].

Incorporating effective heat exchangers (cooler, heater and regenerator) into Stirling engines improve heat exchange between the working fluid and the heat source and sink. However, this often introduces higher aerodynamic flow losses and increases dead volume. The operating speed of a Stirling engine has also a detrimental impact on the thermodynamic processes as an ideal thermodynamic cycles is only realisable if the engine speed is very low (i.e., discontinuous motion of the power piston and displacer), whereas in real engines the speed is typically above 1000 rpm rendering it difficult to achieve adiabatic compression and expansion processes. All these have a negative impact on the overall performance of the engine [53, 57].

3.5 The Mechanical Arrangements and Classification of Stirling Machines

The essential approach of the Stirling engine is two working spaces in which their volumes are varied at the same frequency and are coupled by regenerator to enable heat transfer to and from the working medium. Two Stirling engine designs are common which are generally distinguished by the mechanism involved motion,

couplings namely the kinematic drive and free piston variates.

However, further subdivision are recognized according to [47, 59]:

- (a) Their operation approach;
- (b) Arrangement of pistons;
- (c) Motion mechanisms and piston couplings.

Figure 3.6 shows the classifications of the Stirling engine.

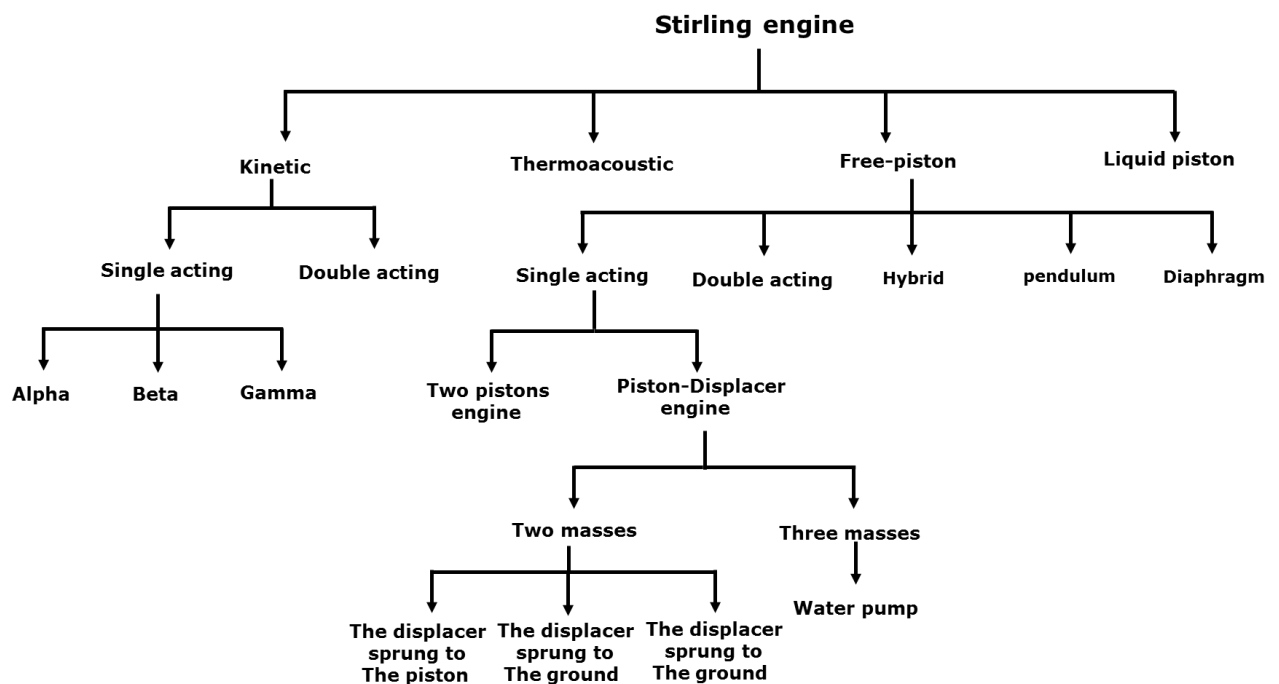


Figure 3.6: Stirling engine classification

As part of this review of Stirling machines, the four main categories of the Stirling engines shown above will be explored very briefly, while the free-piston Stirling engine will be introduced in more detail as the research will be focusing on this kind of Stirling configuration.

3.5.1 Liquid-Piston Stirling Engines:

The fluidyne liquid piston engine is a simple free-piston Stirling engine in which the pistons are formed of liquid. Liquid pistons fit the cylinders without any machining or tight tolerances and mechanical seals or bearings. However, they operate at low frequencies, typically 0.5 to 1 Hz, and at a low pressure close to atmospheric pressure. As a result, the power density is low. The potential application for this type of free-piston Stirling machine is in pumping and desalination of water [57].

3.5.2 Kinematic Stirling Machines:

Kinematic Stirling engines can be categorized depending on their capacities into single acting (suitable for small capacities) and double acting (used for large capacities) [59].

i) Single acting

these are designed for small power capacities up to 10 kW [53] and are made up of a two piston arrangement or a displacer and power piston in one or two cylinders.

ii) Double acting

These are designed for large capacities and large power wherein a series of single acting cylinders are coupled to a common crankshaft. There is an alternative form of the double acting configuration designed by Finkelstein [53], two cylinders, one cylinder contains both compression spaces and the other contains both expansion spaces. Each cylinder has a double-acting piston that combine together to operate as a twin-cycle system [53]. In kinematic double acting Stirling engines, the displacer and the power piston both are coupled to a kinematic drive mechanism such as crank-slider or swash-plate moving in approximately sinusoidal manner with a specified fixed phase shaft.

The kinematic Stirling engines according to their configurations are classified into:

iii) Alpha (α) arrangement:

Where the two pistons are mounted at each end of a common cylinder with the heater, regenerator and cooler inserted in between them as it is shown in Figure 3.7 (a). Both pistons transfer work and must be sealed to contain high pressure working fluid [47, 53].

iv) Beta (β) arrangement:

Where the displacer and power piston are both installed in the same cylinder as shown in Figure 3.8 (b). The displacer shuttle the working gas back and forth to the hot space and does not transfer work.

v) Gamma (γ) arrangement:

It is similar to Beta type, but having the displacer and the power piston in two separated cylinders as is shown in Figure 3.7 (c).

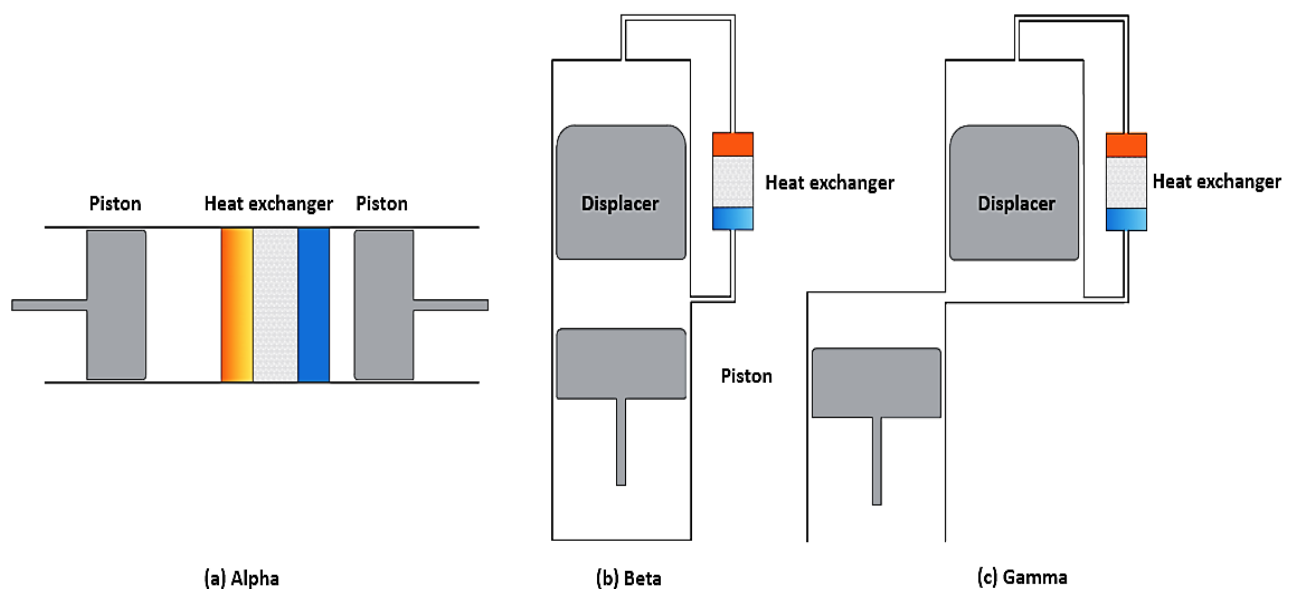


Figure 3.7: Kinematic Stirling configurations

3.5.3 Free piston Stirling engine (FPSE)

The free-piston Stirling engine is a dynamic resonant system operating at almost constant frequency and was invented by William Beale in

1964 and developed by the Sunpower Corporation. The FPSE configuration is considered as one of the most promising and advanced design application [64]. It is distinguished by its unique and simple mechanical arrangement in which the kinematic mechanisms and crank shafts are eliminated. Therefore, the displacer and power piston oscillate by virtue of the gas pressure and/or mechanical springs. The FPSE has a lightweight displacer and heavy power piston drives a permanent magnet linear alternator for power generation. The FPSE is simple in terms of design and can be produced at low cost. The side forces exerted by the reciprocating components against the cylinder wall are very limited or even omitted. The FPSE often is self-starting and has a high performance with long lifespan in comparison with other Stirling engine configurations. It is claimed that the FPSE requires no maintenance services during its lifespan of operation. The casing cylinder of the engine is hermetically sealed thereby the problems of dynamic seals is eliminated in advanced FPSE arrangements [64].

Self-starting occurs when the expansion space is heated by an external heat source, it only requires a small random movement to start fluid movement and then it continues in its steady dynamic vibrating mode. The output work extracted from the piston is regulated by the piston stroke and the resistance forces acting on the piston; however, when

the acting forces increase, the stroke decreases and vice versa, but the output work is almost similar. The output work can be to some extent approximated to the rectangular shape as is illustrated in Figure 3.8.

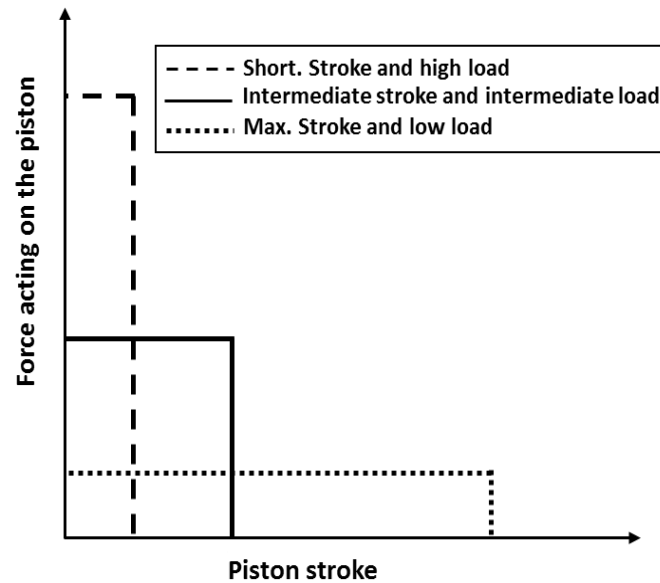


Figure 3.8: The relationship between load and the piston stroke

Despite this, the development and optimisation of the FPSE is a challenging task because of the following practical difficulties [65]:

- (i) No mechanical connections fix the moving parts; however, they are driven dynamically by the working fluid pressure and the spring forces (gas springs or mechanical springs);
- (ii) The strokes and phase angles are affected and set by the dynamic and thermodynamic parameters;

- (iii) The volume of and the pressure in the chambers and hence the pressure losses through the heat exchangers are modified by the displacer and the piston displacements.

A simple schematic drawing points the main parts of the FPSE is shown in Figure 3.9. The essential components of the FPSE are: a light displacer piston attached to the displacer rod and a heavy power piston inside a cylinder forming the working chambers. The system is hermetic in which the cylinder is sealed at one end and a linear load is connected to the power piston. This piston uses the mechanical power to drive a water pump or using a linear electric generator. The regenerator is an essential part for Stirling engines to improve performance comprising a long thin ring between the displacer and the cylinder wall. It acts as the regenerative heat exchanger and the space below the piston called the bounce space.

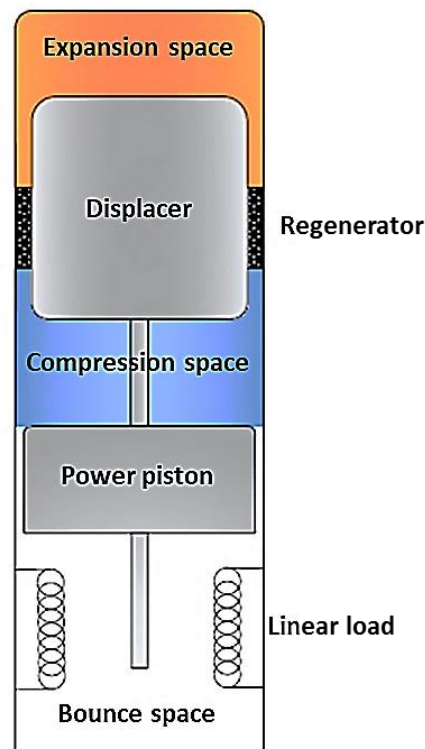


Figure 3.9: Simple diagram shows the main components of the FPSE

Assuming the initial state of the system as it is in Figure 3.9, the displacer and the piston are at the rest positions in which the pressure and temperature are similar in the all spaces. The pressure of the system (in the all spaces) will be the atmospheric pressure. However, when the expansion space is heated by adding heat from an external heat source, the temperature and as result the pressure P_e of the working fluid in the expansion space will increase while the pressure in the bounce space is assumed to remain constant and this process is represented by 0-1 on Figure 3-10. When taking into consideration that the piston and displacer rod are effectively sealed to prevent the working fluid leakage between the working spaces and the bounce

space, the pressure of the working fluid increases and starts to move both the displacer and the piston down the cylinder. However, since the displacer is lighter than the piston (typically 1:10), it accelerates 3 times more quickly than the piston. The working fluid in the compression space is swept into the expansion space through the regenerator. The temperature and hence the pressure increase, which is represented by state 1-2 on Figure 3.10, due to heat addition into the expansion space causes an acceleration of the piston and the displacer. Meanwhile, as the displacer approaches the piston and they both move together so the all working fluid is swept to the expansion space. The pressure in the expansion space stabilizes before beginning to fall (process 2-3) where gas (or air) expands.

The expansion process continues until the pressure in the working space and the bounce space become similar at point 4 and the acting forces of the gas on the movement parts is equal to zero, but both of the piston and displacer continue in motion by the effect of the inertia of the heavy piston leading to a further expansion of the working fluid and a decrease in the pressure in the working space below the pressure of the bounce space, which represented by 4-5 in Figure 3.10.

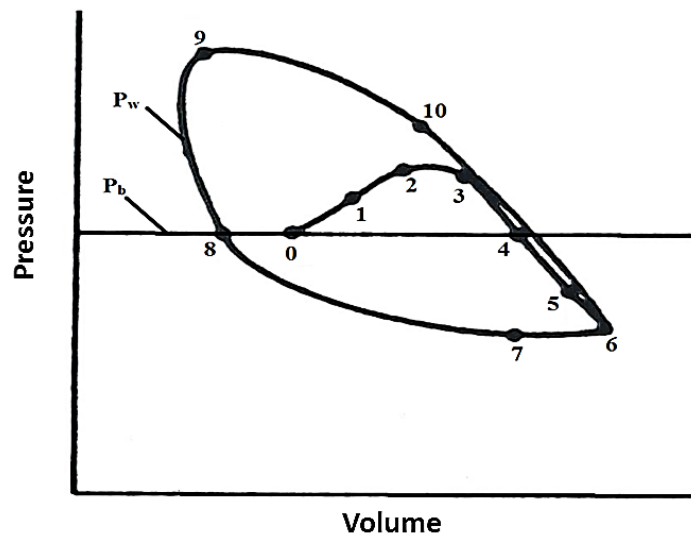


Figure 3.10: PV diagram of the FPSE [58]

As a consequence of the lower pressure in the working space than the bounce space, forces reverse and decelerate the heavy piston and the lightweight displacer, which separates from the piston and reaches the top of the cylinder faster and this process represented by 5-6. The power piston continues moving because of its mass causing further expansion of the working fluid that further decreases its pressure. When the displacer reaches the top of the cylinder, all the working fluid must be moved through the regenerator to the compression space so that the volume of the expansion space is at its minimum and the compression space is at its maximum value. The working fluid temperature decreases during the transfer from the hot space to the cold space through the regenerator which leads to a further decrease in the working space pressure. The expansion process is completed

when the piston reaches 6 and beyond, the process of compression is continues up to 8 where the pressure in the whole system become similar. During the processes 6-7 and 7-8, the displacer is held at the top of the cylinder by the effect of gas forces acting on the displacer rod by mean of the pressure differential ($P_e - P_b$).

Next, the piston continues in motion under the effect of its inertia force beyond the pressure balance at 8 which results in an increase in the working space pressure as shown in 8-9. Then the displacer starts to move down toward the piston under the effect of gravity and the acting gas forces moving the working fluid from the compression space to the expansion space. The cycle then repeated but without the starting stages 0-1 and 1-2.

3.5.4 FPSE Classification and Types

There are various arrangements and types of FPSE, and they share a common principle of having at least one oscillating part driven by the fluid (gas or liquid) pressure forces. Based on the operation principles of FPSEs, they are generally classified into two main groups:

- a. Resonant engine or simple harmonic system.**
- b. Non-resonant, occasionally, known as an over-driven engine.**

Numerous designs and layouts of the FPSE were recognized and developed particularly by Sunpower since Beale has built his first FP engine. However, they can be classified according to their capacity, the arrangement of their cylinders, the number of oscillating masses and the forces driving oscillating elements.[57, 63]

A further classification refers to the power output and capacity of the machine, where the FPSE can be classified into the single-acting FPSE and double-acting FPSE.

I Single-acting FPSE:

The vast majority of the FPSE arrangements are of the single-acting piston, which can be further divided into: (a) two pistons in one single cylinder and, (b) piston and displacer enclosed in one cylinder or being separated in two cylinders [57].

(a) Two pistons machine:

Two-free-pistons are arranged opposed to each other in different forms. Figure 3.11 shows three different common configurations in which both pistons oscillate by virtue of the gas pressure variations in

the engine spaces including expansion space volume, regenerator volume and compression space volume, and the gas or mechanical springs in the bounce space [57].

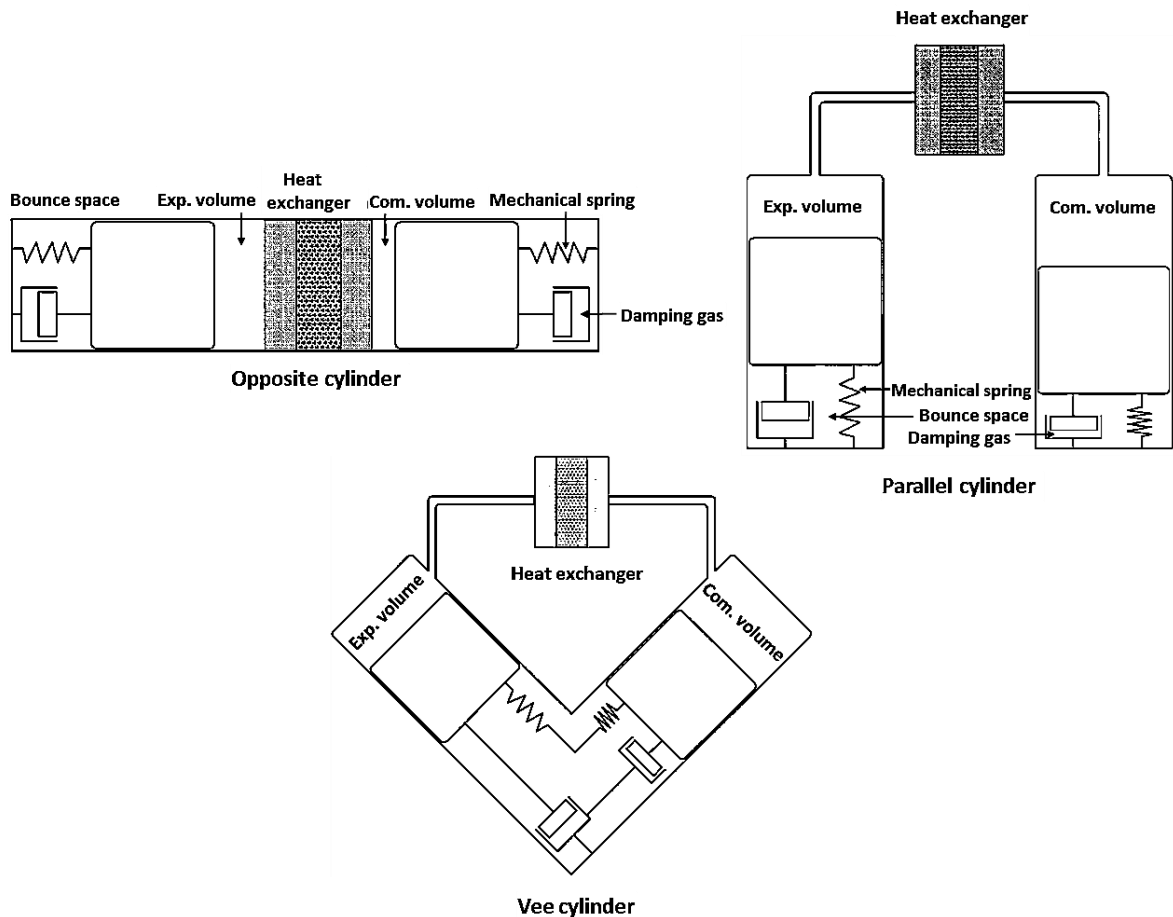


Figure 3.11: Single acting two-free pistons configurations

(b) Piston-displacer arrangements:

The piston and the displacer are set in one cylinder comprise three separate masses, however, in many cases one of these masses is large and it is considered as infinite mass compared with the other two. Commonly, the casing cylinder has a large mass, hence it assumed to

be a rigid component and only the dynamics of the displacer and the piston are considered in the analysis (so called two-mass system analysis). The three-mass design is usually used for water pumps where the cylinder oscillates to drive the plunger of a reciprocating water pump [57].

Mechanical springs are commonly used in the free-piston Stirling machines, however, various designs have been realised based on the way the displacer is driven, which can be divided into three main groups (these are illustrated schematically in Figure 3.12). (a) the displacer sprung to the piston, the displacer and the piston can be coupled by a gas or a mechanical spring; (b) the displacer sprung to the ground, in this design, the dynamic forces acting upon the displacer are independent of the other forces acting on the power piston; (c) in some applications, particularly when the displacer and the piston are required to operate out of phase, the displacer and the power piston are sprung to common ground [57].

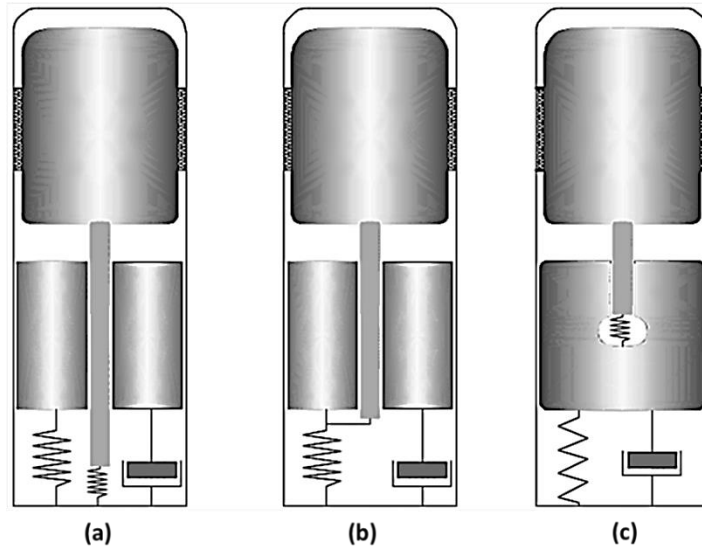


Figure 3.12: The way the displacer is driven

II Split-Stirling machine:

Where the piston and displacer are built in separate cylinders , in-line cylinder, parallel cylinder and orthogonal cylinder are the most common configurations of the split-systems, free-piston Stirling engine. Split-Stirling engines are widely used for Stirling cryocoolers to achieve cryogenic temperatures. Another application for the split-Stirling machine is the hybrid system where the crank and the connecting rod drive one piston and the other reciprocating element is driven by dynamic forces [57]. Figure 3.13 shows the split-Stirling, parallel cylinder free-piston as an example for this kind of configuration.

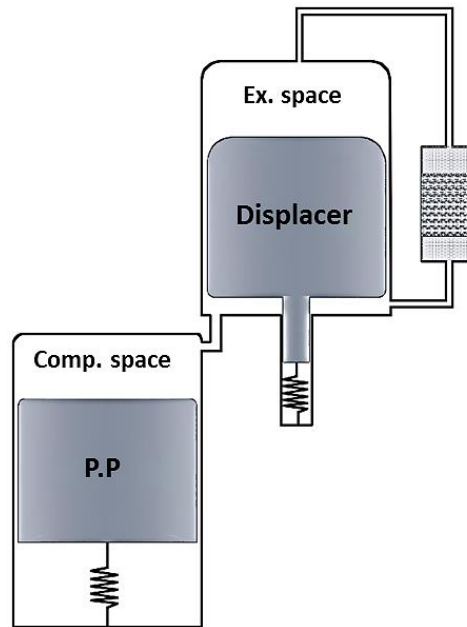


Figure 3.13: Split-Free-piston parallel cylinders

III Double-acting FPSE:

The engine may be built up of multi-systems with only one reciprocating element per system, which is an advantage over the single- acting where two elements reciprocate in one system. Usually, the double-acting FPSE is designed in two forms, one where the engine has two cylinders and is called a Franchot engine or the engine has more than two cylinders where it is called a Siemens engine [57].

IV Diaphragm free-piston Stirling machine:

One major difficulty in Free-piston Stirling engines is wear and friction in the moving parts and gas leakage between the working spaces. This is excepted because, lubricants cannot be used because they find their way into the hotter regions and oxidise. The first heat engine

operating on the Stirling thermodynamic cycle was invented by Cooke-Yarborough in 1967. He replaced the piston by a metal diaphragm oscillating at high frequencies of 50 to 140 Hz to evaluate the difficulties of wear and friction [57]. The concept of using metal diaphragms for free-piston Stirling cryocooler to avoid using sealants has been proposed in order to allow the displacer to oscillate without rubbing or moving seals, to improve the thermal efficiency and the mechanism life span [66].

A Stirling engine for power generation purposes that has been demonstrated recently replaced the power piston with a flexible diaphragm made of natural rubber which was clamped at its circumference to the engine cylinder. This type of engine eliminates problems of sealing the working fluid. Figure 3.14 shows of such arrangement [67].

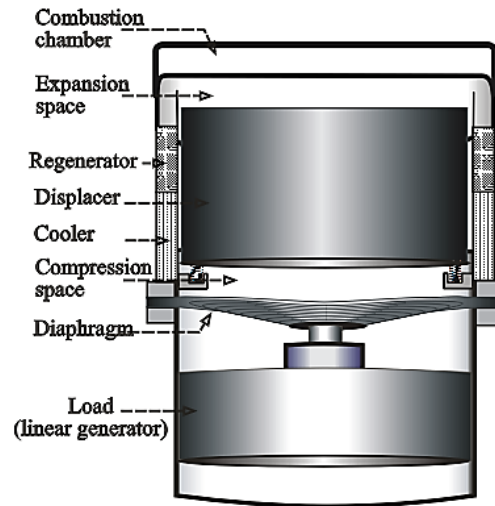


Figure 3.14: Diaphragm free-piston Stirling engine

3.6 Summary

In this chapter, the principle work of the Stirling regenerative cycle from the thermodynamic point view and the dynamic mechanism of the engine have been presented in detail. The design and the aspects of the engine components were reviewed and introduced, then the deferent mechanical arrangements of the Stirling engine for power generation purpose have been classified with particular focus on Free Piston configurations as this research will be investigating the demonstration of improved design of FPSE for power generation applications.

CHAPTER 4

STIRLING CYCLE MATHEMATICAL ANALYSIS

4.1 Introduction

The operation and the behaviour of FPSEs requires an understating of both the dynamic interaction of the moving parts and thermodynamic of the energy flow. While the variation of the working volume and pressure are directly related to the quantities of energy supplied and extracted from the engine, the steady state equilibrium of the engine piston stroke and the operating frequency are determined by its dynamics.

The theoretical analysis and mathematical formulation of the Stirling cycle has been developed with varying degrees of sophistication. The mathematical models and computer simulation use parameters such as heat energy inputs, power output, thermal efficiency, etc.to predict the operating parameters and approximate the preliminary design factors or simulate the engine operating characteristics . In FPSEs, the models usually integrates the thermodynamic and dynamic models.

4.2 Thermodynamic Analysis

The thermodynamic models are based on the mathematical formulation of the Stirling cycle. Depending on the accuracy and complexity of the model, four main different orders of analysis of the Stirling thermodynamic cycle can be distinguished [53, 68]. These include zeroth order, first order, second order and third order.

4.2.1 Zeroth Order (Simple Analysis)

The simplest models is referred to as the Zeroth order analysis which uses an empirical formulae proposed by West [69] to predict the power output of the Stirling engine, as follows:

$$W = B_n p_m f V_{sw} \quad (4.1)$$

Where W is the output power in watts, P_m is the mean cycle pressure in bar, f is the frequency of operation in Hz, V_{sw} is the swept volume of the power piston cm^3 . The dimensionless parameter B_n referred to as Beale number [53] is expressed as a function of the heat source and heat sink temperatures as follows [31].

$$B_n = 0.035 \frac{T_h - T_c}{T_h + T_c} \quad (4.2)$$

Where T_h is the hot temperature at the expansion space, T_c is the cold temperature at the compression space. Temperatures are in (K).

Equation 4.1 is very useful for rule of thumb calculation to determine the output power or the size of new Stirling engine design. The Beale number of 0.015 (or 0.15 for SI units) is often used as an approximation for Stirling engine configurations at high temperature [70]. On the other hand, the Beale number for medium and low temperature differential MTD and LTD is approximated to be 0.002 to 0.008 [70] Figure 4.1 shows a broad range of Beale numbers as a function of heat source temperature T_H [71].

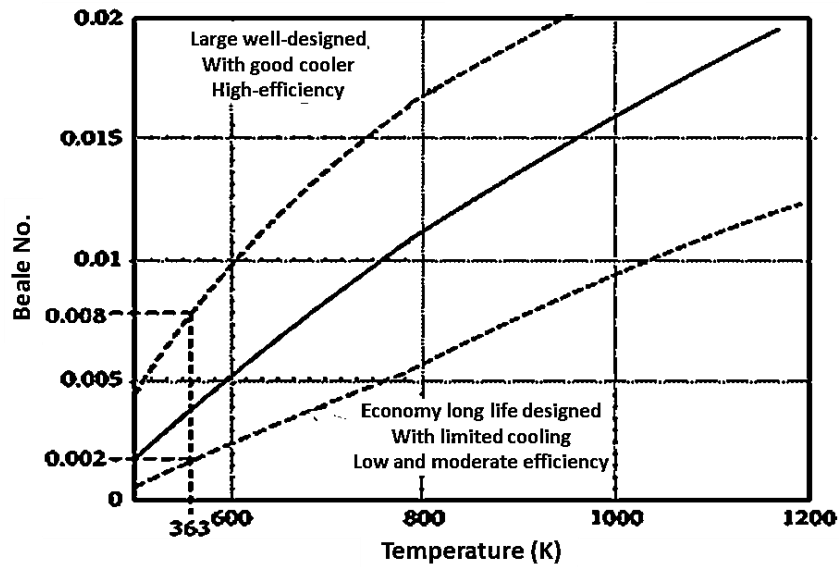


Figure 4.1: Beale number for a range of heater temperature

Furthermore, West improved the Beale equation to include the effect of the thermal irreversibilities in the heat source and heat sink [72].

This is expressed as follows:

$$W = 0.025pfV_{sw} \frac{T_h - T_c}{T_h + T_c} \quad (4.3)$$

Where P is the mean pressure (bar), f is the operation frequency (Hz), V_{sw} is the swept volume (cc), T_h is the hot temperature at the expansion space and T_c is the cold temperature at the compression space (K)

4.2.2 The first-order model (Schmidt analysis)

The first order design methods are preliminary calculations based on algebraic equations of mass and energy conservation to predict the engine performance and to estimate the relationship between the overall size of the engine and its output power [70]. A coherent mathematical model of an ideal Stirling cycle was first developed by Gustav Schmidt in 1871 which as well is occasionally referred to as the classical Stirling cycle analysis. This assumes that the heat addition and rejection from the cycle occur isothermally and is often referred to as the First order analysis [53, 73-75].

The Schmidt analysis is considered the typical and the simplest method of the Stirling cycle and approximates the operation of the real Stirling engine cycle since it provides sinusoidal volume variation of the working spaces in the reciprocating engine, and produces closed-form solutions for its performance. However, Schmidt made his analysis for three configurations of the Stirling engine, alpha, beta and gamma. Despite the fact that Schmidt theory is highly idealized due to

the certain assumptions required for its implementation, it is more realistic than the simple analysis. Although there was a slight improvement in the analysis after 1960 and till today, the Schmidt analysis is widely used for sizing and designing the Stirling engine, particularly for preliminary design of the small scale engines. The Schmidt analysis depends on certain assumptions and conditions to be applied and implemented [51, 53, 56, 73]:

- (i) The compression and expansion processes are considered isothermal (constant heat source and heat sink temperatures);
- (ii) Pressure and working fluid flow losses and leakage are neglected;
- (iii) The engine speed is constant;
- (iv) The working fluid obeys ideal gas law and the expansion and compression processes are reversible;
- (v) The temperature gradient in the heat exchangers is negligible;
- (vi) The regenerator is ideal with effectiveness of unity.

The formulation of Schmidt analysis takes into account the following variables and factors [53]:

- (i) The pistons and other reciprocating elements movement, which is sinusoidal motion in a realistic cycle;
- (ii) The compression and expansion swept volumes;

(iii) The phase angle and the volume variations in the compression and expansion spaces;

(iv) Dead space in the working space.

The Schmidt analysis depends on the isothermal analysis of the Stirling cycle to formulate a closed-form solution for the special case of the sinusoidal volume variations of the working spaces. Figure 4.2 illustrates the ideal isothermal model of an Alpha-type Stirling engine. The engine working space is divided into a compression space (c), cooler (k), regenerator (r), heater (h) and expansion space (e). All the spaces are connected in series and each component considered as a cell of constant properties. The working gas characteristics such as density, temperature, and pressure in each cell are considered homogeneous. The volumes of the compression and expansion space changes with the motion of the power piston and displacer with all other cell volumes are constant and hence constitute the engine's dead volume.

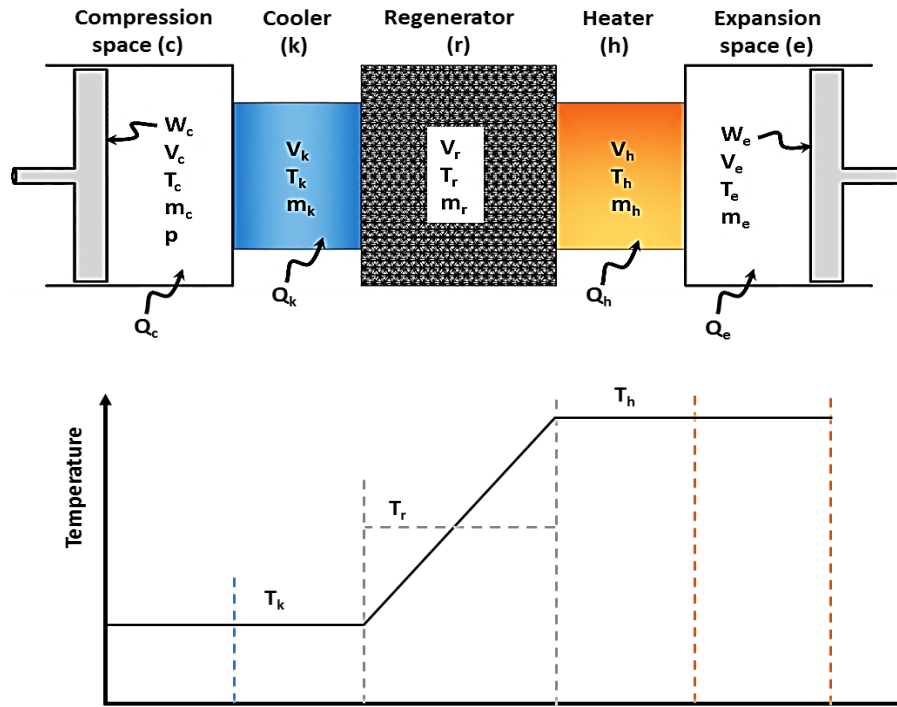


Figure 4.2: Ideal Isothermal Model

In the analysis, it was also assumed that the ideal gas law is applied in each of the control spaces, as follows:

$$PV = mRT \quad (4.4)$$

Where the working fluid properties are described by pressure, P (N/m^2), volume V (m^3), temperature T (K), mass m (kg) and the gas constant R J/kg.K .

The total mass of the working fluid M in the sealed space of the engine is constant, and can be expressed as:

$$M = m_c + m_k + m_r + m_h + m_e \quad (4.5)$$

When indices, (c), (k), (r), (h) and (e) refer to the working fluid property in the compression space, the cooler, the regenerator, the heater and the expansion space. Substituting Equation 4.4 into Equation 4.5 and the latter can be rewritten as:

$$M = \frac{P \left(\frac{V_c}{T_k} + \frac{V_k}{T_k} + \frac{V_r}{T_r} + \frac{V_h}{T_h} + \frac{V_e}{T_h} \right)}{R} \quad (4.6)$$

Given that the regenerator temperatures is limited by the heater and cooler temperatures and acts as an instantaneous thermal store as the working fluid is shuttled between the expansion and compression space, in the ideal analysis, it is assumed that the temperature in the regenerator is linearly distributed and the effective temperature T_r is given by:

$$T_r = \frac{(T_h - T_k)}{\ln\left(\frac{T_h}{T_k}\right)} \quad (4.7)$$

Substituting equation 4.7 in equation 4.6 and rearranging for working pressure in the engine p as a function of the volume variations V_c and V_e is given by equation 4.8:

$$P = \frac{MR}{\left(\frac{V_c}{T_k} + \frac{V_k}{T_k} + \frac{V_r \ln\left(\frac{T_h}{T_k}\right)}{(T_h - T_k)} + \frac{V_h}{T_h} + \frac{V_e}{T_h} \right)} \quad (4.8)$$

The mechanical work output of the engine (w) is by means of volume variations in the working spaces V_c and V_e , and hence the total work done over a complete cycle is computed from the PV diagram of the thermodynamic cycle as the sum of the area of the expansion and compression PV diagrams performed by the displacer and power piston. This is expressed as follows:

$$W = W_e + W_c = \oint P dV_c + \oint P dV_e = \oint P \left(\frac{dV_c}{d\theta} + \frac{dV_e}{d\theta} \right) d\theta \quad (4.9)$$

Where θ is the angular position of the moving pistons.

Heat transfers to and from the working spaces at the hot and cold temperatures T_h and T_c respectively. Figure 4.3 shows the generic control volume schematically which may either be represent the working spaces cell and/or the heat exchanger cell.

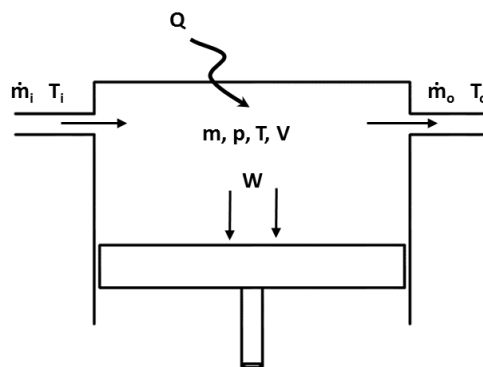


Figure 4.3: Control volume – generalized cell

Referring to the energy conservation equation, enthalpy is transferred between the cells by virtue of mass flow g_A and temperature T in

which the suffixes (i, o) refer to (in) and (out) respectively, resulting in change in the internal energy of the cell. The general energy equation for the working gas in the cell with neglecting potential and kinetic energy can be expressed in differential form as:

$$DQ + (C_p T_i \dot{m}_i - C_p T_o \dot{m}_o) = DW + C_v D(mT) \quad (4.10)$$

Where D is the differential, Q is the transferred heat in (W), C_p and C_v are the gas specific heat capacity at constant pressure and constant volume respectively (J/kg.K) and $\dot{m}$ is the mass flow rate in (kg/sec).

Considering the isothermal model, the temperature between the interfaces in the compression space and cooler cells, as well as, in the expansion space and the heater cells does not change, thus, $T_i = T_o = T$. For non-steady flow and from mass conservation point view, $\dot{m}_o - \dot{m}_i$ is equal to the rate of mass in the cell, for the ideal gas $C_p - C_v = R$. Therefore, by simplifying and integrating the equation 4.10, the net heat transferred over the cycle is given by:

$$Q = RT \oint Dm + \oint DW \quad (4.11)$$

However, the steady state over the complete cycle is reasonable assumption, and hence the cyclic change in working gas mass for each cell is equal to zero. Hence, applying the equation 4.11 on each cell of

the isothermal model, the neat heat transferred to and from the working spaces is given by:

$$Q_e = W_e = \oint P \left(\frac{dV_e}{d\theta} \right) d\theta \quad (4.12)$$

$$Q_c = W_c = \oint P \left(\frac{dV_c}{d\theta} \right) d\theta \quad (4.13)$$

Similarly, the Equation 4.13 applies on the heat exchanger cells where the heat is instantaneously captured from and released to the working gas in which no work is done.

$$Q_k = 0 \quad (4.14)$$

$$Q_h = 0 \quad (4.15)$$

$$Q_r = 0 \quad (4.16)$$

Thermal efficiency is given by

$$\eta = \frac{W}{Q_e} \quad (4.17)$$

The ideal efficiency is only dependent on the heat source and heat sink temperature. In theory, the Stirling cycle efficiency is equal to Carnot's cycle.

$$\eta = \frac{W}{Q_e} = 1 - \frac{T_k}{T_h} \quad (4.18)$$

Schmidt started his analysis from the previous equations and found the solutions of these equations for the special case of sinusoidal volume variations of the working spaces with respect to the cycle angle θ . In this analysis, the volume variation of the expansion and compression space is assumed to be sinusoidal, as shown in Figure 4.4. It can be seen that the volume variations of the working spaces V_c and V_e over one cycle. The angle α is the phase shift angle with the expansion space leads the compression cycle by $\pi/2 = 90^\circ$, which is one quarter of the cycle [76]. In the actual cycle, the phase angle α varies between 60° to 120° without severe effect on the performance.

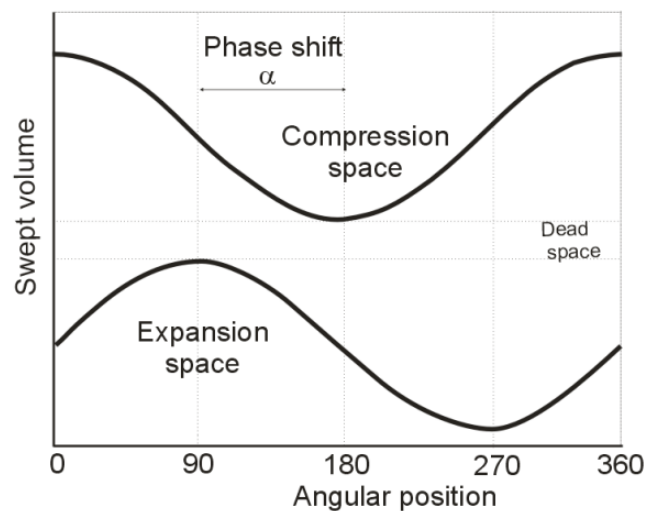


Figure 4.4: Volume variations of compression and expansion spaces for single cycle of Alpha type [68]

The sinusoidal variations of compression and expansion space are expressed by the Equations 4.19 and 4.20:

$$V_c = V_{clc} + \frac{V_{swc}(1+\cos \theta)}{2} \quad (4.19)$$

$$V_e = V_{cle} + \frac{V_{swe}(1+\cos(\theta+\alpha))}{2} \quad (4.20)$$

Where θ is the cyclic angle and V_{cl} and V_{sw} are the clearance and swept volume respectively in (m³).

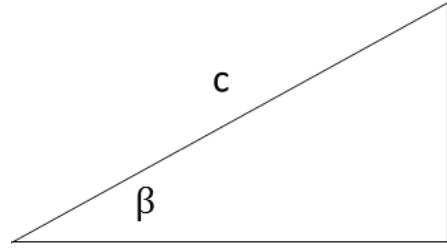
By substituting Equations 4.19 and 4.20 in the Equation 4.8, the pressure as a function of the volume variations in the working spaces is expressed as follows:

$$P = \frac{MR}{\left[S + \left(\frac{V_{swc}}{2T_k} + \frac{V_{swe} \cos \alpha}{2T_h} \right) \cos \theta - \left(\frac{V_{swe} \sin \alpha}{2T_h} \right) \sin \theta \right]} \quad (4.21)$$

Where

$$S = \left[\frac{V_{swc}}{2T_k} + \frac{V_{clc}}{T_k} + \frac{V_k}{T_k} + \frac{V_r \ln\left(\frac{T_h}{T_k}\right)}{(T_h - T_k)} + \frac{V_h}{T_h} + \frac{V_{swe}}{2T_h} + \frac{V_{clc}}{T_h} \right] \quad (4.22)$$

The pressure equation above can be simplified using trigonometric calculations. New variables β and c are defined as it is illustrated in the following right-angled triangle.



$$c \sin \beta = \frac{V_{swe} \sin \alpha}{2T_h}$$

$$c \cos \beta = \frac{V_{swe} \cos \alpha}{2T_h} + \frac{V_{swc}}{2T_k}$$

$$\beta = \tan^{-1} \left(\frac{\frac{V_{swe} \sin \alpha}{T_h}}{\frac{V_{swe} \cos \alpha}{T_h} + \frac{V_{swc}}{T_k}} \right) \quad (4.23)$$

and

$$c = \frac{1}{2} \sqrt{\left(\frac{V_{swe}}{T_h} \right)^2 + 2 \frac{V_{swe}}{T_h} \frac{V_{swc}}{T_k} \cos \alpha + \left(\frac{V_{swc}}{T_k} \right)^2} \quad (4.24)$$

Then substituting for β and c in the pressure Equation 4.21 we obtain the simple equation of the pressure at the working spaces:

$$P = \frac{MR}{S(1 + b \cos \varphi)} \quad (4.25)$$

Where

$$\varphi = \theta + \beta \text{ And } b = c/S$$

At $\varphi = 0$ and π , the pressure is at its minimum and the maximum respectively and expressed by:

$$P_{\min} = \frac{MR}{S(1+b)} \quad (4.26)$$

$$P_{\max} = \frac{MR}{S(1-b)} \quad (4.27)$$

And the average pressure over the cycle can be calculated from

$$P_{\text{mean}} = \frac{MR}{2\pi S} \int_0^{2\pi} \frac{1}{(1+b \cos \varphi)} d\varphi \quad (4.28)$$

The integrated equation is given by:

$$P_{\text{mean}} = \frac{MR}{(S\sqrt{1-b^2})} \quad (4.29)$$

The heat transferred over the cycle is equal to the work done by the engine as a result of the varying volumes of the compression and expansion spaces and it is expressed as follows:

$$Q_c = W_c = \int_0^{2\pi} \left(P \frac{dV_c}{d\theta} \right) d\theta \quad (4.30)$$

$$Q_e = W_e = \int_0^{2\pi} \left(P \frac{dV_e}{d\theta} \right) d\theta \quad (4.31)$$

The volumes derivatives are expressed as following:

$$\frac{dV_c}{d\theta} = -\frac{1}{2} V_{\text{swc}} \sin \theta \quad (4.32)$$

$$\frac{dV_e}{d\theta} = -\frac{1}{2} V_{\text{swe}} \sin(\theta + \alpha) \quad (4.33)$$

The work done over the complete cycle is the algebraic sum of the work done of the compression and expansion spaces.

By substituting the equations 4.32 and 4.33 in the equations 4.30 and 4.31 respectively, we obtain

$$W_c = -\frac{V_{swc} M R}{2s} \int_0^{2\pi} \frac{\sin \theta}{1 + b \cos(\beta + \theta)} d\theta \quad (4.34)$$

$$W_e = -\frac{V_{swe} M R}{2s} \int_0^{2\pi} \frac{\sin(\theta + \alpha)}{1 + b \cos(\beta + \theta)} d\theta \quad (4.35)$$

Urieli and Berchowitz [76] solved the equations 4.34 and 4.35 using Fourier series and obtained:

$$W_c = \pi V_{swc} P_{mean} \sin \beta \frac{(\sqrt{1-b^2}-1)}{b} \quad (4.36)$$

$$W_e = \pi V_{swe} P_{mean} \sin(\beta - \alpha) \frac{(\sqrt{1-b^2}-1)}{b} \quad (4.37)$$

The cycle efficiency is expressed as following:

$$\eta = \frac{W}{Q_e} = 1 - \frac{T_k}{T_h} \quad (4.38)$$

4.2.3 Second Order Model (Adiabatic Decoupled Analysis)

The second order-analysis calculates the thermal and friction losses whilst cooler, regenerator and heater are treated as perfect heat

exchangers and the compression and expansion spaces are assumed perfectly insulated. Therefore, the adiabatic analysis is more realistic than the Schmidt analysis, particularly for the large engines operating at high frequencies [53, 73-75]. Finkelstein developed the first second-order model using an adiabatic analysis in 1960 [71]. In 1980s, Urieli and Berchowitz developed the adiabatic model by incorporating the effects of non-ideal heat exchangers and applying the heat transfer equations to calculate cooler and heater gas temperature, however, the assumptions of the isothermal process are still valid in the all heat exchangers in which the perfect gas laws is applicable in each control volume and the flow is one-dimensional.

The expansion and compression control volumes were assumed to be sinusoids and adiabatic. The temperature of the heater and cooler were assumed to be constant and known and defined as initial conditions. The temperature gradient in the five control volumes of the engine is presented in Figure 4.5

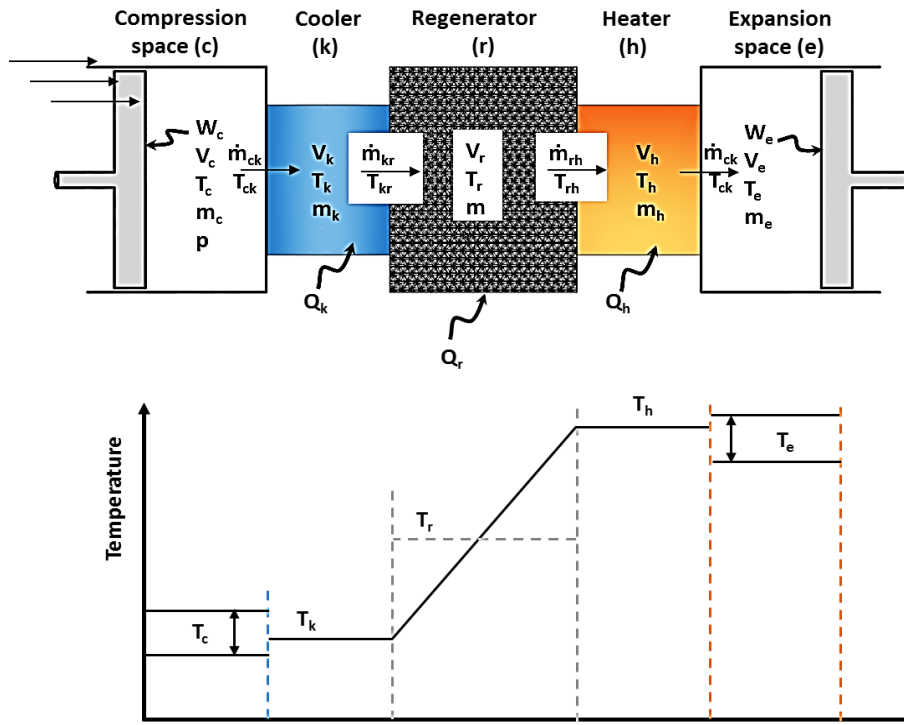


Figure 4.5: The ideal adiabatic model

The adiabatic model formulation was derived by applying the perfect gas laws, conservation of energy, mass, and momentum to each control volume as follows [77, 78]:

$$dQ = dW - RTdm \quad (4.39)$$

$$PV = mRT \quad (4.40)$$

$$dm = m_i - m_o \quad (4.41)$$

The differential form of the equation of state is given by equation 4.42:

$$\frac{dP}{P} + \frac{dV}{V} = \frac{dm}{m} + \frac{dT}{T} \quad (4.42)$$

Following the similar approach of the isothermal analysis by assuming the total mass of the working gas in the engine is constant and applying the equations of energy and state to each control volume and then the obtained equations are linked by applying the continuity equation to the entire engine.

The equation of mass balance in the differential form is expressed as [52]:

$$\Delta m_c + \Delta m_k + \Delta m_r + \Delta m_h + \Delta m_e = 0 \quad (4.43)$$

The differential form of the equation of state if considered that volumes and temperatures are constant is reduced to:

$$\frac{\Delta m}{m} = \frac{\Delta p}{p} \quad (4.44)$$

Substituting the equation of the ideal gas and applying the equation 4.44 to the heat exchanger cells and substituting in equation 4.43, the equation of mass balance can be written as [52]:

$$\Delta m_c + \Delta m_e + \frac{\Delta p}{R} \left(\frac{V_k}{T_k} + \frac{V_r}{T_r} + \frac{V_h}{T_h} \right) = 0 \quad (4.45)$$

By applying the energy equation 4.11 to the compression and expansion control volumes and simplifying using the ideal gas

relations, the energy equations for the working spaces are given by [52]:

$$\Delta m_c = (p\Delta V_c + \frac{V_c \Delta p}{\gamma}) / (RT_{ck}) \quad (4.46)$$

$$\Delta m_e = (p\Delta V_e + \frac{V_e \Delta p}{\gamma}) / (RT_{he}) \quad (4.47)$$

Substituting the energy equations 4.46 and 4.47 in the mass balance equation 4.45 and simplifying to obtain an explicit equation in D_p [52]:

$$\Delta p = \frac{-\gamma p (\Delta V_c / T_{ck} + \Delta V_e / T_{he})}{V_c / T_{ck} + \gamma \left(\frac{V_k}{R_k} + \frac{V_r}{T_r} + \frac{V_h}{T_h} \right) + V_e / T_{he}} \quad (4.48)$$

Once the pressure p is evaluated from equation 4.48, the mass of the gas in the working spaces is calculated from the equations 4.46 and 4.47, and the rest of the variables are obtained from the ideal gas equation and the mass balance equation.

The interface temperatures T_{ck} and T_{he} are conditional on the direction of the flow and defined as follows [52]:

If $\dot{m}_{ck} > 0$ then $T_{ck} = T_c$ otherwise $T_{ck} = T_k$

If $\dot{m}_{he} > 0$ then $T_{he} = T_h$ otherwise $T_{he} = T_e$

The temperatures T_c and T_e are evaluated by applying the equation of state [52]:

$$T_c = pV_c/(Rm_c) \quad (4.49)$$

$$T_e = pV_e/(Rm_e) \quad (4.50)$$

Applying equation 4.44 to the heat cells to obtain the mass accumulations within each cell of the heat exchanger that are expressed as follows [52]:

$$\Delta m_k = m_k \Delta p/p \quad (4.51)$$

$$\Delta m_r = \frac{m_r \Delta p}{p} \quad (4.52)$$

$$\Delta m_h = \frac{m_h \Delta p}{p} \quad (4.53)$$

The heat exchangers are isothermal, thus the heat transferred to the working gas is evaluated using the energy equation 4.39 [52]:

$$\Delta Q_k = V_k \Delta p c_v / R - c_p (T_{ck} \dot{m}_{ck} - T_{kr} \dot{m}_{kr}) \quad (4.54)$$

$$\Delta Q_r = V_r \Delta p c_v / R - c_p (T_{kr} \dot{m}_{kr} - T_{rh} \dot{m}_{rh}) \quad (4.55)$$

$$\Delta Q_h = V_h \Delta p c_v / R - c_p (T_{rh} \dot{m}_{rh} - T_{he} \dot{m}_{he}) \quad (4.56)$$

Urieli and Berchowitz did further development on their adiabatic model by dividing the regenerator into two cells in which the heat exchanger gas temperatures are no longer equal to their associated wall temperatures, and are not constant over the cycle. However, the heater and cooler walls are maintained at the constant source and sink temperatures T_{wh} and T_{wk} respectively. The steady-flow conditions are considered for each cell of the engine, hence it is named the quasi steady model, which result in a set of complex equations requiring numerical methods and computer programmes to be solved [52].

The quasi-steady-flow model is relatively complex, therefore, Urieli and Berchowitz tried to simplify this model by using the ideal adiabatic results and the empirical correlations for friction factor and heat transfer coefficient to evaluate the heat transfer rates and the pressure drop in the three heat exchanger elements. New gas temperatures are defined through the heat transfer rates, which then used in another iteration of the ideal adiabatic analysis. A convergence is achievable by repeating these steps. This is a combination of the ideal adiabatic analysis and simple heat exchanger analysis approach simplify the quasi-stead flow model with relatively similar accuracy. Figure 4.6 shows heat balance of a typical control volume cell.

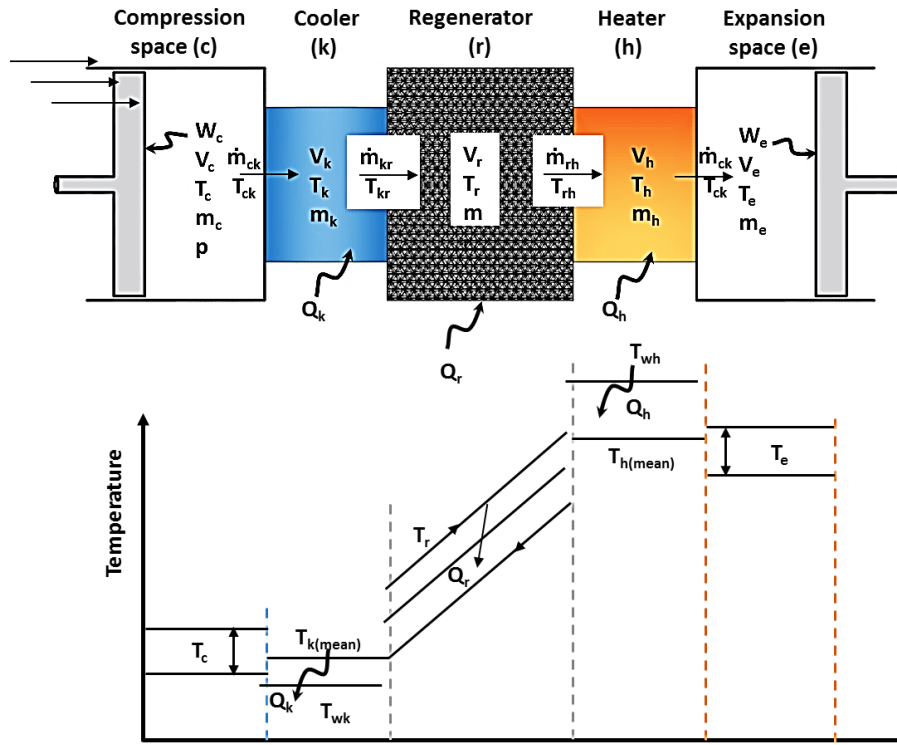


Figure 4.6: Heat balance of the engine cells –simple analysis

The values of heat transfer Q for the heater and cooler cells are then used to determine the actual operating temperature of the working fluid in the heater and cooler, as below:

$$T_k = T_{wk} - \frac{Q_k}{(h_k A_{wgk})} \quad (4.57)$$

$$T_h = T_{wh} - \frac{Q_h}{(h_h A_{wgh})} \quad (4.58)$$

4.2.4 Simple Analysis

The Simple analysis, which was developed by Berchowicz and Urieli, is used to calculate irreversibility due to heat, pressure and friction losses in the heat exchanger (cooler, heater and regenerator). Their

analysis based on the number of transfer units (NTU) method to calculate the heat exchangers effectiveness using equation 4.59:

$$\varepsilon = \frac{NTU}{(1+NTU)} \quad (4.59)$$

Where, the NTU is expressed as:

$$NTU = (\frac{A_w}{A})/2 \quad (4.60)$$

And Stanton Number is given by:

$$N_{ST} = \frac{h}{\rho u c_p} \quad (4.61)$$

The working fluid “pumping losses” expressed as pressure drop across the heat exchangers is evaluated with reference to the compression space pressure as:

$$\Delta p = \frac{f \rho L u^2}{2 d_h} \quad (4.62)$$

Where f is the frequency (Hz), ρ is the working fluid density (kg/m^3), L is the heat exchanger length (m), d_h is the hydraulic diameter (m) and u is the velocity of the fluid (m/s).

Therefore, the effective engine work output could be given as follows:

$$W = \oint P(dV_e + dV_c) - \oint \sum \Delta P dV_e = W_i + \Delta W \quad (4.63)$$

Where W_i is the work done per cycle that is obtained from the ideal adiabatic analysis and ΔW is the pumping loss per cycle.

4.2.4 Third-Order Model

It is a very detailed and complicated analysis, which commonly is referred to as nodal design analysis methods. For a known set of operating parameters such as temperatures, mean pressure, and frequency, Schmidt analysis can be used to yield a better estimate of the engine thermal performance. This analysis, does not however quantify heat losses involved in the engine (i.e. heat loss due to conduction, convection, shuttle, and friction). Therefore, for a comprehensive design of Stirling engine a computer model to optimize heat transfer components (heater, regenerator, and cooler) and yield the highest efficiency would be required. Existing thermal models use mathematical analysis based on nodal method [53, 69, 73-75]. The method subdivides the working fluid flow-passages into a network of control volumes (nodes). Each control volume is then defined by a set of differential equations of conservation of energy, mass, and momentum

Applying energy conservation governing principle to each control volume as follows:

$$\frac{dQ_j}{dt} - Q_{dissj} + C_p(T_i m_i - T_{i-1} m_{i-1}) = C_v \frac{d}{dt}(m_j T_j) + dW_j \quad (4.64)$$

Where, Q_j is heat energy (J) in each node (control volume), Q_{dissj} is heat dissipated from each node, C_p is working fluid specific heat capacity at constant pressure (J/kg K), T_i is the temperature (K) transferred between nodes (at control volume boundary), T_j is the temperature of each node, m_i is the mass transferred (kg) between nodes, m_j is mass (kg) of each node, C_v is working fluid specific heat capacity at constant volume (J/kg K) and W_j is the work in each node.

As explained previously, the total mass of the working fluid, m , in the engine is assumed constant hence the mass conservation is expressed as:

$$\sum_1^n m_i = M \quad (4.65)$$

In this mathematical formulation, it is also considered that the working fluid across the control volumes interfaces obeys the mass continuity equation, which can be given by:

$$\frac{dm_i}{dt} = m_i - m_{i-1} \quad (4.66)$$

Finally, the conservation of the working fluid momentum is observed in each control volume. This is given by the following.

$$P_i - P_{i-1} = f_i \frac{L_i}{D_{hi}} \rho_i v_i^2 \quad (4.67)$$

Where, P_i is the pressure (N/m²) between nodes, f is the friction factor, L_i is node (control volume) length (m), D_{hi} is hydraulic diameter (m) of each control volume, ρ_i is the density (kg/m³) of the working fluid and v_j is the velocity (m/s) of the working fluid.

This method needs expertise in heat transfer and computer modelling to be able to obtain a converging model solution. However, the results from the simulation are difficult if not impossible to be measured experimentally.

4.3 Dynamic Analysis

The dynamics of the power piston and the displacer in the Stirling engine is a complex process, however, the kinematic type is controlled by a mechanical mechanism. The dynamic analysis of the FP type is further complicated owing to the fact that the power piston and the displacer dynamics are caused by the springs and gas pressure. For that reason, the dynamics of the FPSE is coupled to the thermodynamics of the working fluid [57]. Therefore, the dynamic analysis is particularly important in design of a FPSE in which the engine is usually represented by three oscillating masses (i.e., displacer, power piston and the engine casing mass). Each of the

components is subject to a driving force, an inertial force and a damping force, as shown in Figure 4.7. The main driving forces are a result of pressure variation in the engine. The frequency of the moving components is also determined by mass, the restoring spring stiffness, and damping forces imposed by loading driven by the engine and a variety of mechanical and aerodynamic frictional forces.

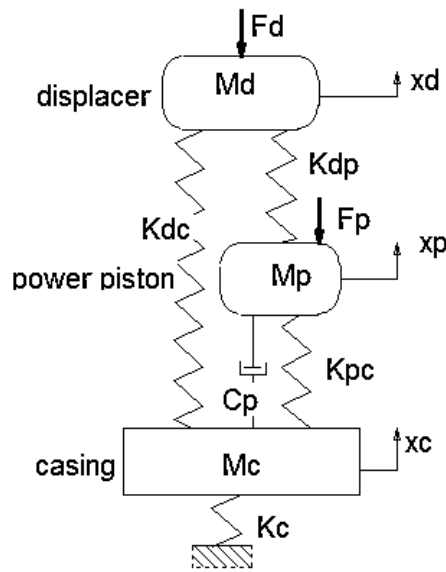


Figure 4.7: Dynamic representation of FPSE

In thermodynamic analysis, the working space pressure is a function of the volume variations for a particular geometry, gas type and temperature distribution. Additionally, the volume varies with the oscillation of the piston, displacer and casing. Therefore, the pressure variation can be written as a function of the power piston, displacer and casing motion [52].

$$p = f(x_p, x_d, x_c) \quad (4.68)$$

The dynamic behaviour of the FPSE is significantly influenced by the pressure variation, which acts as if it is a damping load, due to the dissipative (damping) forces, which in turn have a major effect on the power output. The pressure gradient effects, which act as if they are damping a load, can be represented by equivalent linear dampers as a function of the reciprocating components (displacer, piston and casing) velocities [52]:

$$\Delta p = f(\dot{x}_p, \dot{x}_d, \dot{x}_c) \quad (4.69)$$

$$A_D \Delta p = C_p \dot{x}_p + C_d \dot{x}_d + C_c \dot{x}_c \quad (4.70)$$

Where Δp is the instantaneous pressure difference (bar), A_D is the cross sectional area of the displacer, C is the damping coefficient (N/m), $\dot{x}_p$, $\dot{x}_d$ and $\dot{x}_c$ are the velocities (m/s) of the piston, displacer and the casing.

The other damping forces such as the internal hysteresis loss of the mechanical springs and the gas spring hysteresis loss may also be accounted for by linear dampers.

For the displacer:

$$f_d = C_{Hdc}(\dot{x}_d + \dot{x}_c) \quad (4.71)$$

For the power piston:

$$f_p = C_{H_{pc}}(\dot{x}_p + \dot{x}_c) \quad (4.72)$$

Where $(\dot{x}_d + \dot{x}_c)$ and $(\dot{x}_p + \dot{x}_c)$ are the relative velocities with respect to the casing for the displacer and the power piston respectively, C_H is the gas spring damping coefficient (N/m).

By considering that the oscillation of the power piston and the displacer are approximated by a sinusoid wave, while the oscillation of the cylinder is neglected. The behaviour of a FPSE can be formulated into state equations by combining the dynamic analysis of the moving parts of the engine and the thermodynamic process of the working fluid in the engine [69].

The motion for the oscillating piston and displacer, taking in consideration the gas spring losses, is determined from the dynamic equations 4.73 and 4.74 respectively [52, 69]:

$$M_p \ddot{x}_p = (p_c - p_b)A_p - (C_{pc} + C_{H_{pc}})\dot{x}_p - (C_{pc} + C_{H_{pc}})\dot{x}_c \quad (4.73)$$

$$M_d \ddot{x}_d = (C_d + C_{H_{dc}})\dot{x}_d + C_p \dot{x}_p + (C_c + C_{H_{dc}})\dot{x}_c + A_R(p_c - p_d) \quad (4.74)$$

Where, M_p , M_d is the mass (Kg) of the piston and the displacer respectively, $C_{pc}(\dot{x}_p + \dot{x}_c)$ is the forces exerted by the load, P_p is the

pressure in the compression space, P_b , P_d and P_c are the pressure (N/m) in the bounce space, the expansion space and compression space, A_p , A_d , A_r are the cross sectional areas m^2 of the power piston, the displacer and the displacer rod.

The instantaneous pressure in the engine given by equation 4.75 is a function of the position of the displacer and power piston in the working space and can be rewritten as [79]:

$$P_c = P_m \left(1 - \frac{A_p x_p - A_d x_d}{S T_k} - \frac{A_d x_d}{S T_h} \right) \quad (4.75)$$

Where, P_m is the mean pressure (N/m²) in the engine working space when the displacer and the power piston are at mid-stroke, A_p , A_d is cross-section area (m²) of the piston and the displacer respectively, x_p and x_d is the motion (m) of the piston and displacer respectively.

The general forms of the dynamic equations (equations of motion) represent the stability of the power piston, displacer and casing motions are written by Urieli and Berchowitz [81]:

$$\ddot{x}_p = S_{pp}x_p + S_{pd}x_d + S_{pc}x_c + D_{pp}\dot{x}_p + D_{pd}\dot{x}_d + D_{pc}\dot{x}_c \quad (4.76)$$

$$\ddot{x}_d = S_{dd}x_d + S_{dp}x_p + S_{dc}x_c + D_{dd}\dot{x}_d + D_{dp}\dot{x}_p + D_{dc}\dot{x}_c \quad (4.77)$$

$$\ddot{x}_c = S_{cc}x_c + S_{cp}x_p + S_{cd}x_d + D_{cc}\dot{x}_c + D_{cp}\dot{x}_p + D_{cd}\dot{x}_d \quad (4.78)$$

The general equations of motion can be simplified to:

$$[\ddot{x}] = [S][x] + [D][\dot{x}] \quad (4.79)$$

Where $[S]$ and $[D]$ are the coefficients of springs and dampers respectively.

4.4 Summary

The different orders of the thermodynamic analysis of the Stirling cycle were mathematically formulated. Four main different orders depending on accuracy and complexity have been introduced in detail. The mathematical equations control the system dynamics and the oscillating components of the engine were extracted and expressed.

CHAPTER 5

THE PROOF-OF-CONCEPT ENGINE DESIGN AND CALCULATIONS

5.1 The Proposed Mechanical Arrangement for the FPSE

The new design of low temperature difference Free Piston Stirling Engine introduced and described in this study are based on the innovative design concept of using a flexible diaphragm as a power piston and replacing the mechanical springs in other FPSE designs by using a special type of flexible metal bellow with high reliability and long life-time. Initially, a number of designs for the FP engine with bellows that can be practically manufactured have been proposed. However, the assembly and manufacture issues were taking into considerations in order to simplify the assembly and reduce the total cost. The displacer is located between the hot space the cold end of the engine, thus, it will experience a large temperature gradient, and it must achieve a specific operating frequency and stroke. Therefore, it is designed to be made of a light weight sealed metal cylinder supported by a specially shaped metal bellows that act as a mechanical spring. Similarly, the power piston is proposed to be mounted on another

metal bellow to form a mass-spring oscillating component so that it is in a position of free movement. The bellows generate the compression and expansion strokes, therefore the clearance volumes have been calculated according to the bellows specifications, so the clearance volumes are minimised and the two moving parts oscillated freely. The mechanical arrangement of the components of the engine is shown in Figure 5.1. The two working spaces are designed to be coupled by two gas passages for the air to be shuttled in between them.

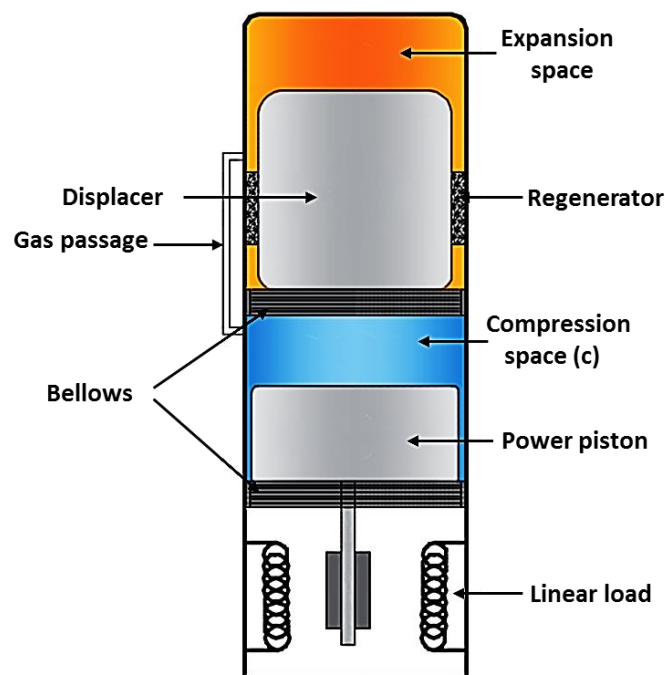


Figure 5.1: The proposed design of the FPSE

The displacer, the power piston and the engine casing can be machined from stainless steel tubes with the approximated dimensions in Table 5.1. The displacer is designed to be clamped to the top bellow flange (the small flange), whereas, the bottom bellow flange (the large

flange) will be used as the firm base in which the flange of the engine casings will be mounted to. The power piston is designed to be clamped to the other bellow (the small bellow). The dimensions of the displacer and the power piston cylinders were chosen based on the bellow's size, and the length (height) of the bellows was based on the number of the convolutions (which were calculated based on their stiffness and maximum stroke) to achieve the required movement for the displacer and the power piston to shuttle the air in between the working spaces. The bellows diameter was chosen according the availability off the-shelf, but the bellow diameter could not be less than half of its length to avoid wobbling. The engine casing cylinder was designed to be heavier to be more rigid and stand the temperature gradient.

Table 5.1: The approximated dimensions for the main engine parts

Part description	Part dimensions (mm)
Displacer cylinder	95D × 50L × 1t
Bellow clamped to the displacer	OD 95 × ID 76
Engine casing cylinder (top part)	97D × 90L × 1.5t
Power piston	74D × 30L × 1t
Bellow clamped to the power piston	OD 66 × ID 48
Engine casing cylinder (bottom part)	76D × 55L × 1.5t

5.2 Engine Design Parameters

The engine shown in Figure 4.8 was analysed and simulated in order to evaluate and validate its performance and calculate the output power with taking into consideration the pumping losses caused by the fluid friction as well as the imperfection of the heat exchangers. A range of operating parameters were used in the computer model, however, the values are listed in table 5.2 are appropriate to meet the design and size of the engine to achieve output power at low temperature ranges and at atmospheric pressure.

Table 5.2: Operating parameters defined by the user

Variable	Value	Unit
Cold side wall temperature T_k	30~50	°C
Hot side wall temperature T_h	250~300	°C
Operating frequency f	10~30	Hz
Mean operating pressure P_{mean}	2	bar

Table 5.3 presents the numerical results obtained from the simulation. The temperature gradient in the cooler and heater and the ideal thermal performance of the engine (heat and frictional losses has yet not considered) are listed in the top of Table 5.3. However, the actual engine performance (heat and frictional losses are considered) and the imperfect specifications of the regeneration are listed in the bottom of the Table.

Table 5.3: The numerical results from the computer analysis

Variable name	Value	Unit
User defined cold side wall temperature (T_{wk})	35	°C
Simulated cold side gas temperature (T_k)	40	°C
User defined hot side wall temperature (T_{wh})	280	°C
Simulated hot side gas temperature (T_h)	265.9	°C
Gas mass	0.35	g
Pressure phase angle (beta)	32.7	degree
Schmidt thermal efficiency	44	%
Schmidt total outpour power	0.98	W
Thermal efficiency of thermodynamic cycle (Ideal Adiabatic)	41.6	%
Total output power of thermodynamic cycle (Ideal Adiabatic)	0.76	W
Heat power added by heater (Simple Analysis)	2.13	W
Heat power rejected by cooler (Simple Analysis)	-1.38	W
Regenerator effectiveness (Simple Analysis)	20.8	%
Regenerator net enthalpy loss (Simple Analysis)	69.8	W
Regenerator wall leakage (Simple Analysis)	125.2	W
Actual total output power (Simple Analysis)	0.76	W
Actual heat power required (Simple Analysis)	197.1	W
Actual efficiency (Simple Analysis)	~1	%

5.3 Proposed Engine Modelling

Equations 4.39, 4.40, 4.41 and 4.42 in chapter 4 were set up for every control volume and then solved simultaneously using a second order Runge-Kutta method. It was assumed that the movement of the displacer and power piston are sinusoids and the operating frequency, working fluid mass, fluid type, and the geometry of the system were specified. The model was set up so that according to Urieli [2] it could be solved as an initial-value problem by assigning an appropriate initial conditions and integrating the differential equations until a steady state was reached.

The simulation computes all engine parameters over one complete thermodynamic cycle including the pressure and volume variation, cyclic energy input and output of the engine components, working fluid temperatures variation, and overall efficiency of the cycle.

5.4 Computer Model Results

A computer model was developed based on the second order mathematical formulation by applying the simple analysis developed by Urieli and Bertchwiz.

The enclosed area of the PV diagram shown in Figure 5.2. is representing ~~equal to~~ the work produced by the engine. Although the

PV diagrams are not in phase, it can be seen that the enclosed area of the both diagrams are almost similar. As consequence, the output powers of the both analyses are relatively similar. Referring to Table 4.3, the output power of Schmidt stands at 0.98 W and that of the simple analysis equals to 0.76 W. The highest pressure reached in the engine is of 2.009 bars at the end of the compression cycle and the lowest is 1.986 bar, whereas the mean pressure is 2 bars. It is notice that the power calculated by the simple analysis is lower than that of the Schmidt analysis due to the effect of losses and imperfect regeneration that were calculated in the simple analysis.

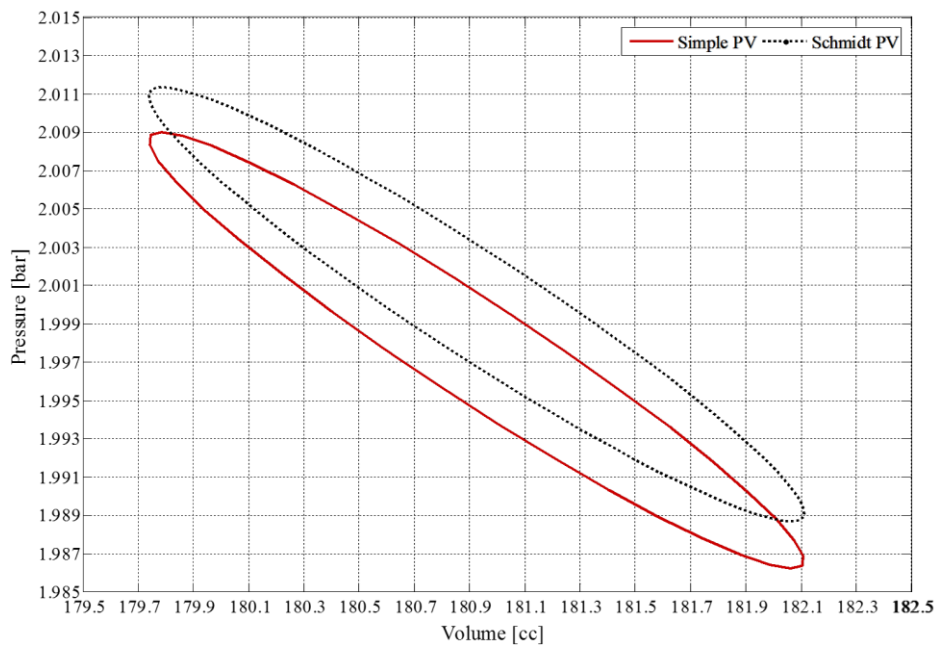


Figure 5.2: PV diagram

Figure 5.3 shows pressure variation in the expansion and compression space of the engine and the two pressure waveforms are nearly in

phase, which underpins low pressure losses in the heat exchangers. On the other hand, the shift of the pressure variations of the Simple analysis from the Schmidt pressure variation is due to the perfect regeneration assumptions and considering the pressure losses in the Schmidt analysis. Furthermore, it is observed that the maximum pressure is at 140° phase angle for both analyses.

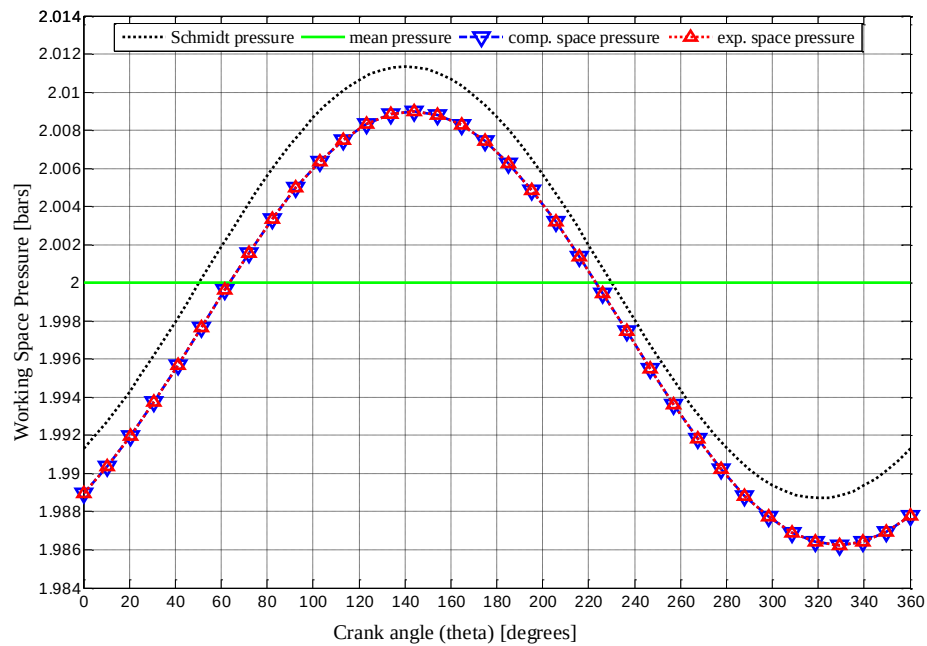


Figure 5.3: Pressure variation in the working spaces

The flow rate of the gas g_A that swept by the displacer and power piston between the engine cells is shown in Figure 5.4. The flow of the gas treated as quasi-steady-flow, thus the pressure at any moment is constant. The phase-shift angle is assumed to be 90° , throughout the engine, though virtually varies between 60° to 120° . Furthermore, it is noticeable that the maximum and the minimum flow rate of the gas

between the hot and cooled ends of the engine are at 140° and 320° respectively (total flow gas flow rate – black line) where the gas flow rate of the hot end of the engine (red lines) is equal to that of the cold end (blue lines) Heat transferred between the working spaces and by the regenerator is presented in Figure 5.5. It is observed that the heat rejected by the cooler Q_k and the heat added by the heater Q_h , were not large values due to limited differential temperatures applied to the

ends of the engine particularly for small size engine and imperfect regeneration.

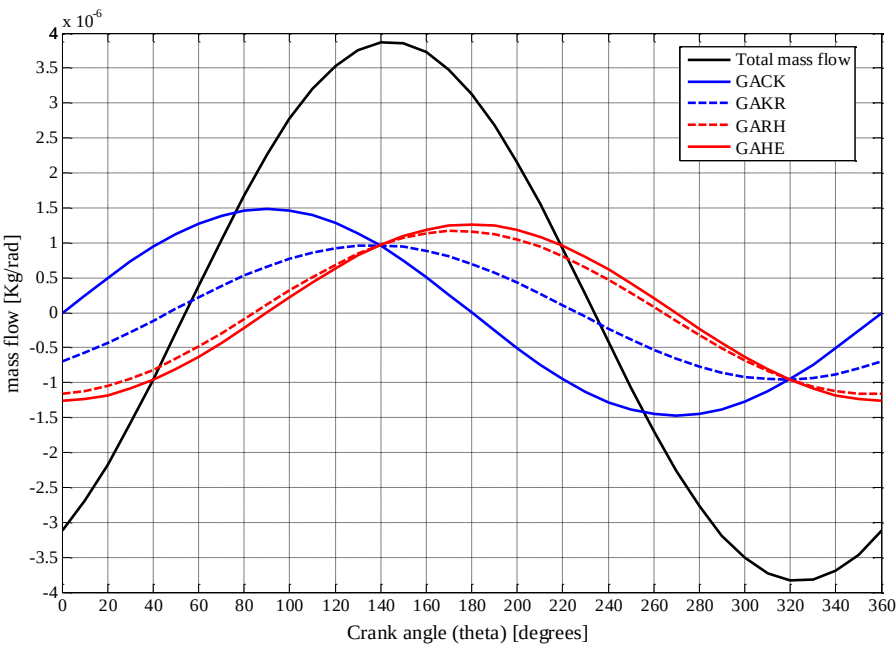


Figure 5.4: Gas mass flow rate between the working spaces

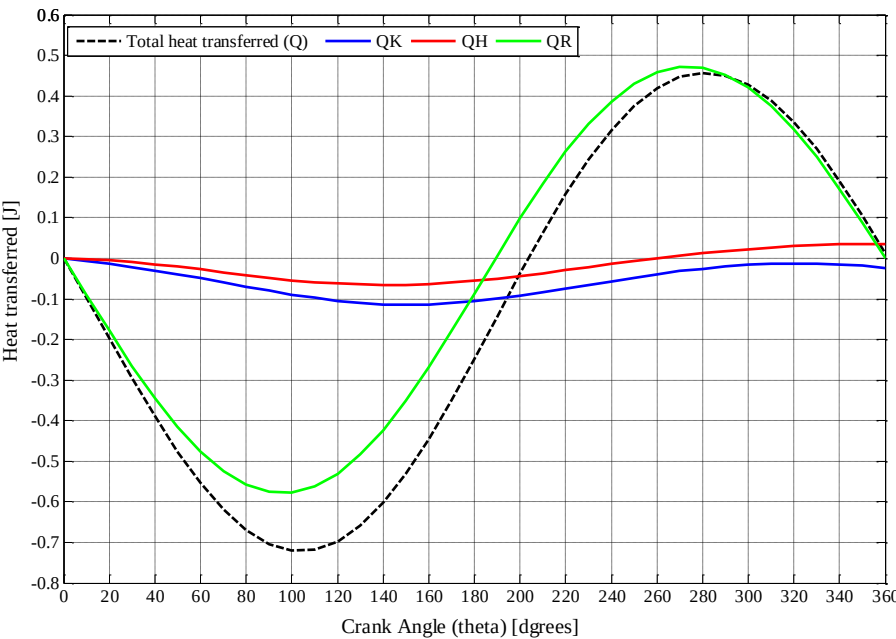


Figure 5.5: Cyclic heat transferred within the engine cells

The temperature variations of the simple analysis model are introduced in Figure 5.6. It should be noted that the mean effective temperature (gas temperature) for the real heater T_h and cooler T_k are respectively lower and higher than that of the heat exchanger wall temperatures T_{wh} and T_{wk} . Therefore, the engine operates on temperature limits lower than that originally specified; consequently, the performance of the engine is negatively affected.

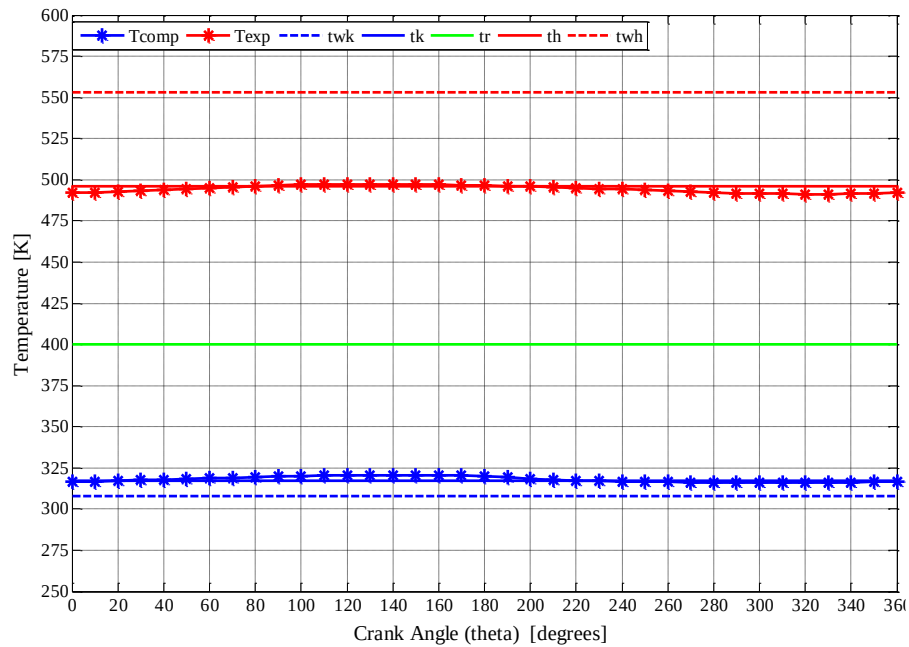


Figure 5.6: Temperature change in the engine cells with respect to the crank angle

Figure 5.7 shows the variations of pressure drop in the heater, the regenerator and the cooler. Large effective heat transfer area and small dead volumes cause high pressure drop across the heat exchanger. The pressure drop at the hot side of the engine comparing to that at the cooler is smaller, the reason behind is the temperature

and the displacer is moving faster than the power piston. The increase in the pressure drop across the heat exchanger reduces the output power of the engine and hence its performance.

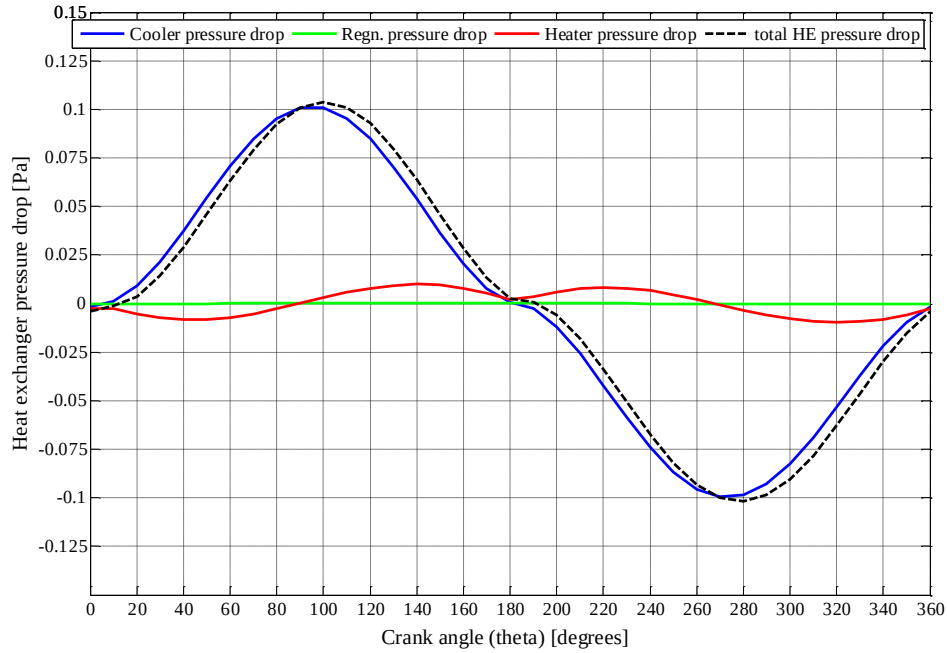


Figure 5.7: Pressure losses in the heat exchanger

5.5 Summary

A preliminary design of novel arrangement for low temperature FPSE is schematically presented and generally described where its substantial dimensions and the main parameters are identified. The results from different simulations including Schmidt and adiabatic were compared to those from Simple analysis. The results from the computer model proved that small size prototype of the proposed engine could be designed to work at medium and low temperature ranges at low pressure (atmospheric pressure) and low frequency.

CHAPTER 6

THE PROOF-OF-CONCEPT ENGINE DESIGN, FABRICATION AND ASSEMBLY

6.1 Introduction

Since the invention of Beale's FPSE, many configurations have been produced, tested and demonstrated for applications ranging from power generation to cryogenics. Common to all FPSEs is the simple linear mechanical oscillation of power piston and displacer in a housing cylinder without mechanical linkage. In this work, a simplified mechanical configuration of the displacer and power piston assembly and fabrication was proposed. The design seeks to address some of the disadvantages associated with FPSEs, such as working fluid seal, the hysteresis losses in air springs and dry friction losses caused by moving parts. The displacer component of the proposed design is to be integrated into an elastic metal bellows component with long service life.

The structure of the cylindrical flexure bellows acts as a mechanical support springs and assist in eliminating the leakage issues without need for sealing. However, two different designs for the power piston

were considered using the metal bellow and the flexure rubber diaphragm.

6.2 Preliminary Design Considerations

The displacer is a key component in FPSE design. The displacer is generally made of a light long sealed cylinder out of low thermal conductivity materials. The displacer is mounted on mechanical or gas springs to form a mass-spring resonance system. Therefore, the displacer design affects the dynamic stability of a FPSE, in which its mass, spring stiffness, viscous damping and friction need to be carefully designed.

The preliminary design focuses on the structure of the displacer and the power piston and their respective spring assemblies with the aim to:

- (i) Produce a lightweight displacer;
- (ii) Eliminate air leakage from the expansion to compression space;
- (iii) Eliminate mechanical seals and friction between the displacer and cylinder;
- (iv) Create pressure-driving forces for the displacer at atmospheric pressure;
- (v) Minimise the dead volume;
- (vi) Operate at low differential temperature.

6.3 Mechanical Springs and Material Selection

In a typical FPSE, the displacer oscillations are possible because of the use of mechanical spring, air springs or both. The displacer mounting springs can take many forms including helical (coil), planar or flat springs. Examples of the mechanical springs used in FPSE designs are shown Figure 5.1 [80, 81]. Figure 5.1 (a) is a commonly used compression helical springs that operates under compression loads and its use in FPSE requires a pre-compression setting. Figure 6.1 (b) represents a planar spring with opening grooves in its plane to form a flexible structure, however, the oscillation amplitudes with this type of spring is usually small.

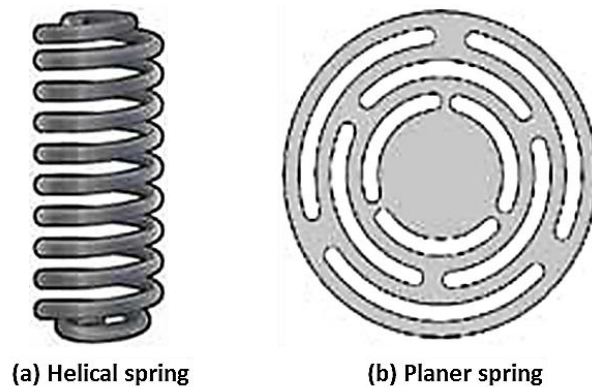


Figure 6.1: The mechanical springs used in FPSEs

Gas springs on the other hand are less prone to mechanical fatigue but still suffer from leakage and hysteresis losses causing a decrease in the output power and the engine performance. In addition, the gas

spring mechanism is subject to the alignment problems leading to frictional losses between the oscillating components [82, 83].

The spring constant of the coil springs does not change with charging pressure, but it is required to be pre-compressed even when the engine is not in operation to maintain the displacer in the correct location. The pre-load increases the stress levels which result in a reduction of spring life. Moreover, the static friction increases due to the side forces and impedes the initial engine starting [80, 83]. Planar springs provide both good support for the oscillating parts, good alignment and excellent life span. They have been used in a number of FPSE designs and if oscillation amplitude is kept below the material's threshold fatigue then the material should not fail [83, 84]. Furthermore, planar springs have constant spring rate and are only affected slightly by temperature [83].

In this study, the focus is on other types of flexible materials (metallic and non-metallic) to be studied and implemented in the proposed FPSE design eliminate some of the disadvantages of mechanical springs. In this study, two other types of spring mechanism and materials for the displacer and power piston support were investigated.

6.4 Metal Bellows

A metal bellow is a thin cylindrical element that has a corrugated wall with high pressure, torsion and corrosion resistance capable of withstanding a high temperature difference. The metal bellows are flexible and at the same time pressure resistant. The corrugations give the bellow flexible axial, lateral and angular deformation [85]. Bellows are manufactured from ductile metallic materials and alloys for various purposes and applications, such as the mechanical seals, flexible mechanical couplings and as circuit breaker in high-voltage facilities [85, 86]. Metal bellows are made of different metallic materials such as austenitic stainless steel, nickel, nickel-brass alloys, copper and bronze. They are available in various sizes, stiffness and with different connecting parts (end fittings) and shapes.

Metal bellows as displacer mounting flexure structure have also been implemented in some of the FPSE designs particularly in cryocoolers [87].

The metal bellows once configured properly is robust and can operate safely for a long service life without maintenance requirements. However, the flexibility and the number of cycles of the bellow is dependent on the material and the load applied.

6.4.1 Bellow Configurations

Flexibility and service cycle, pressure resistance, temperature resistance, tightness and corrosion resistance are the main requirements for the metal bellows.

A design compromise usually involves selecting a bellows material that can withstand the required operating pressure and temperature as well as corrosion resistance if deployed in a corrosive environment. The air or fluid tightness on the other hand is dependent on the fabrication process. Pressure-resistance and life cycle, however, are fulfilled by following a proper bellows design, which can be verified by calculating stress, operating loads, torsion and deformation.

Metal bellows can be produced by rolling, welding, chemical deposition, hydroforming pressure and electroforming processes [88]. Metal bellows can be categorised according to the manufacturing processes and by means of their deflection.

i. Fabrication process

Metal bellows are fabricated using forming and welding processes. The formed bellows, also called “*convoluted bellows*”, are manufactured from a thin-walled cylindrical tube by means of mechanical pressure,

hydraulic pressure or electroforming process. Figure 6.2 (a) shows an example of a formed bellows with two different types of end fittings.

The welded bellows, also called edge-welded bellows, are essentially made of diaphragm discs (moulded thin metal plates) welded at the inner and outer diameter to form a corrugation (convolution) in which a number of convolutions form the bellows. Edge-welded bellows can be fabricated from various materials and alloys, however, they are relatively expensive compared to metal formed bellows. They are manufactured with thin walls with high specific expansion compensation (up to 80% of the overall length) because of their low stiffness comparing to the formed type. Edge-welded bellows are complex, sensitive and their welded joints might be affected by the high temperature of welding, thus their resistance to the pressure is limited [89-91]. Figure 6.2 (b) shows an example of an edge welded bellows.

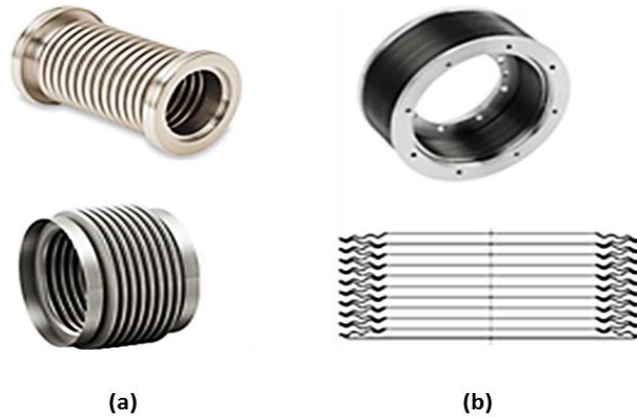


Figure 6.2: (a) Formed bellows (b) Edge-welded bellows

ii. Deflection and movement

Metal bellows have a high degree of movement and deflection. This includes axial, lateral and angular motion. In an axial motion, the bellows is compressed and expanded in an axial line parallel to its centreline to form an axial deflection and provide the axial stroke. The compression and expansion movement is usually limited so that no permanent deformation of the structure of the bellows is sustained. Figure 6.3 (a) shows a metal bellows with axial movement.

In lateral motion , bending forces are applied to the bellows (sideway motion). This occurs when the ends of the bellows move perpendicular to its axis. Figure 5.6 (b) shows the lateral movement of a bellows.

Finally, bellows undergo an angular deflection when a torque is applied around the radius as illustrated in Figure 6.3 (c) [92].

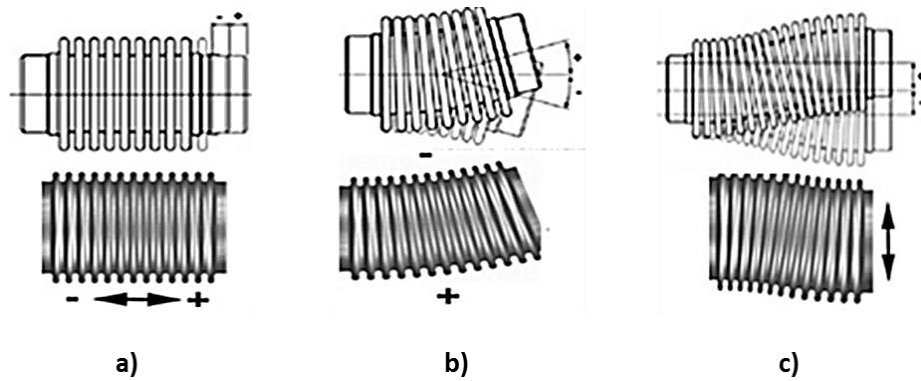


Figure 5.6: (a) axial bellow (b) Lateral bellow (c) angular bellow

6.4.2 Bellow Design Characteristics

When considering the application of metal bellows, there are many design properties that need to be considered. Table 6.1 gives the main design characteristics that need to be accounted for when selecting, sizing and implementing metal bellows in engineering applications.

Table 6.1: Bellow characteristics

Properties	Description
Mechanical stress:	Mode of operation: extension, compression torque bending, swivelling, and offset
Spring stiffness:	Compromise for flexible movement and stroke per convolution
Temperature:	Operating temperature in excess of 400 °C
Life expectancy: The number of cycles before failure	

Operating pressure: The pressure between inside and outside the bellow	Depends on material, frequency, and stroke. Infinite life span is possible for deflection below threshold.
	Important for high pressure applications
Axial movement: The maximum axial stroke (compression and expansion)	
Assembly method: Joining methods (end fitting)	The total axial movement depends on the number of convolutions. Compromise between length and dynamic stability.
Geometric constraints: The restrictions on size of the bellow, such as maximum outer diameter minimum inner diameter and maximum length	Welding, soldering, gluing or bolting using flanges and cuffs.
	Length is more significant than diameters to avoid swivelling and side forces

6.4.3 Bellows Design and Sizing

Sizing the bellows correctly is important to improve the dynamic stability and life expectancy. The design involves selecting suitable length, inner and outer diameters, the number of convolutions, stiffness etc. [85]. In selection of bellows, the number of the convolutions depends strongly on the bellow diameter, particularly at low frequencies.

Similarly, the deflection of the bellow is directly proportional to the number of convolutions, n_w , and stiffness, k , of the material of the bellow. The bellow's length is restricted by the available axial space and stability, as long bellows are prone to wobbling and swerving. The

number of convolutions, n_{ω} , of a bellow is calculated from equation 6.1 [85].

$$n_{\omega} = \frac{2\delta}{2\delta_n} \quad (6.1)$$

Where 2δ is the bellow deflection and $2\delta_n$ is the deflection per convolution.

The stiffness spring rate of the convolution is directly proportional to its length and inversely proportional to its thickness. The stiffness of the bellows are equivalent to having the convolutions mounted in series and the overall spring rate, k , be determined by equation 6.2.

$$k = \frac{k_n}{n_{\omega}} \quad (6.2)$$

Where, k_n , is the spring rate per convolution (N/m) and n_{ω} is the number of the convolutions.

The number of convolutions in a bellow is inversely proportional to the bellow diameter at fixed frequency. Figure 6.4 shows the relationship between the number of convolutions and the resonance frequency of bellows. For low operating frequency, a high number of convolutions needs to be assembled to reduce the overall stiffness. At high

frequencies, the effect of number of convolution on resonance frequency diminishes.

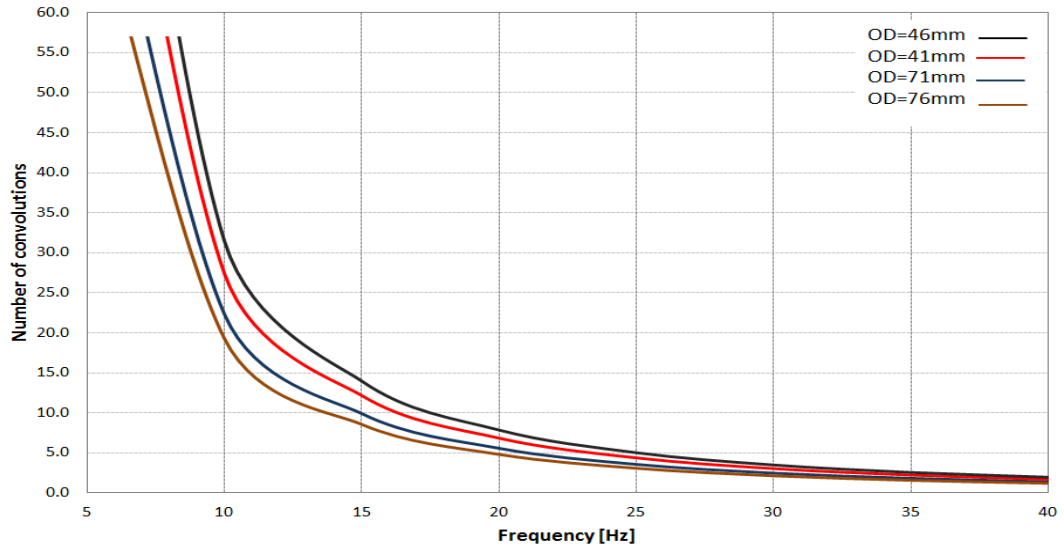


Figure 6.4: Effect of resonance frequency on number of bellow convolutions

The operating frequency, f , of a bellows-mass system is given by:

$$f = \frac{1}{2\pi} \sqrt{\frac{k}{m}} \quad (6.3)$$

Where, the bellow spring constant, k , given by equation 6.2 depends on the geometry and material of the bellows.

The number of convolutions per bellow, n_ω , is calculated from the equations 6.2 and 6.3 as follows:

$$n_\omega = \frac{k_n}{4\pi^2 f^2 \cdot m} \quad (6.4)$$

6.4.4 Bellow Mounted Displacer in FPSE

The bellows can be manufactured from different flexure materials with properties similar to those of typical mechanical springs in terms of flexibility and vibration. However, its operating life span can be longer than the conventional mechanical springs when properly designed. The advantage of bellows is that they can be manufactured as sealed cylindrical structures, allowing separation of internal working fluid regions and eliminating fluid leakages. In addition, bellows offer compact designs, easiness for coupling, dynamic stability and robustness. Figure 6.5 shows an example of a flanged edge-welded bellows that can be used as part of the displacer and power piston assemblies in an FPSE.

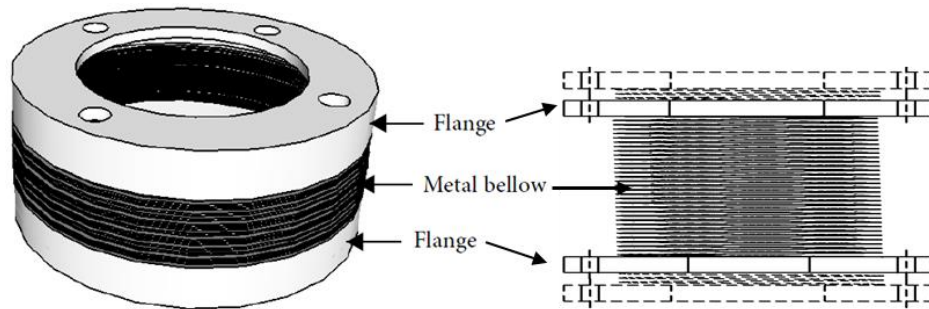


Figure 6.5: Schematic diagram of the proposed bellows

Referring to the operating conditions and the functions of the vibrating components in the FPSE, edge welded stainless steel bellows could be suitable options for this application on account of [93]:

- (a) Low spring forces per convolution, hence high flexible bellows;

- (b) Lowest assembly dimensions due to the shorter length of corrugation;
- (c) Long service life, ranges between 1000,000 to 10,000,000 load cycles [85].

6.5 Diaphragms

Diaphragms are flexible plates with an elastic structure that can deflect when subject to pressure loads [94]. There are two main categories of diaphragm structures: flat and corrugated diaphragms. Compared to flat diaphragms, corrugated counterparts can achieve higher deflection amplitudes. An example of a flat diaphragm is shown in Figure 6.6.

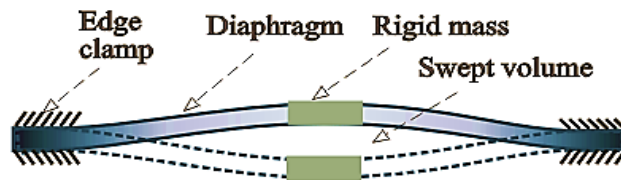


Figure 6.6: Swept volume approximation

Diaphragms have been used in various applications, such as measuring instruments, MEMS (microelectromechanical systems), hydraulic pumps, etc. [90, 94, 95]. Furthermore, a metal diaphragm was first used successfully as power piston in Stirling engines by Cooke-Yarborough of Harwell labs, UK in 1967, for the design of Thermo-mechanical generators (TMG) [96]. Metal flat diaphragms

have also been used to replace sliding pistons in some cryocoolers [16].

6.5.1 Diaphragm Design

The design of diaphragms falls under the well-developed theory of plate vibration under different free and clamped arrangements [97]. The analysis of diaphragms includes establishing maximum deflection, resonance frequencies, materials stresses, etc. In this work, the focus will be on design of a circular diaphragm with clamped edges and free movement at the centre. The analysis will also be limited to thick diaphragms (i.e., the central deflection is smaller than the thickness of the diaphragm).

It is well established that when an edge clamped circular diaphragm is subject to a distributed pressure force upon its surface, it deflects with maximum deflection occurring at the centre. Figure 6.7 shows a schematic of an edge clamped diaphragm with radius, a , and maximum central deflection, w_0 .

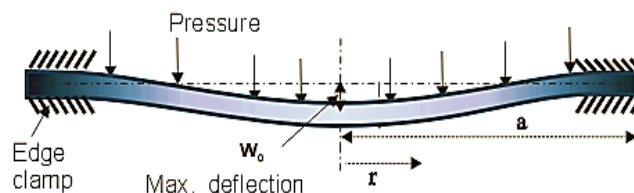


Figure 6.7: Schematic of a deflected diaphragm under distributed pressure forces

The deflection of the diaphragm can be expressed as follows [18].

$$w = \frac{3P(1-\mu^2)}{16Et^3} (a^2 - r^2)^2 \quad (6.5)$$

Where, w , is the deflection at radius, r , P is the uniform applied pressure, t is the diaphragm thickness, E and μ are the Young's Modulus and Poisson's ratio of the diaphragm material.

The maximum deflection of the diaphragm is at its centre (i.e., $r = 0$).

Hence, the maximum deflection is determined as:

$$w_0 = \frac{3Pa^4(1-\mu^2)}{16Et^3} \quad (6.6)$$

The maximum diaphragm deflection, w_0 , from the normal flat plane position is assumed to be less than the thickness of the diaphragm; accordingly, the diaphragm is approximated by a cone shaped dome. Therefore, the swept volume between the normal flat position and the maximum deflection can be calculated by integrating equation 6.5 and the following expressed can be [90].

$$V = \frac{\pi Pa^6}{16Et^3} (1 - \mu^2) \quad (6.7)$$

The total swept volume of the diaphragm when subject to alternating pressure forces (i.e., the deflection is between $\pm w$) is then double that given by equation 6.5.

Similarly, the diaphragm under pressure exerts a restoring force, which is due to its material stiffness. The stiffness is defined as the ratio of the applied load to the corresponding deflection [94].

$$k_p = \frac{pA}{w} \quad (6.8)$$

Where k_p is the diaphragm spring rate (kg/m), p is the applied pressure applied load (N/m²), A is the effective area (m²) and w is the deflection (m).

The application of an alternating excitation pressure force combined with the diaphragm material characteristics allows the diaphragm to oscillate at one of its natural resonance frequency modes. This is usually expressed in terms of a dimensionless natural frequency parameter λ_{ij} as given in the following [91]:

$$f_{ij} = \frac{\lambda_{ij}^2}{2\pi a^2} \sqrt{\frac{Et^2}{12\rho(1-\mu^2)}} \quad (6.9)$$

Where, f_{ij} , represents the resonance frequency at different model indexes (i : number of nodal diameters and j is the number of nodal circles), ρ is the plate material density, and t is the plate thickness.

For a circular flat diaphragm clamped at its edges and, the fundamental oscillation frequency $i = 0$ and $j = 0$ corresponds to the dimensionless natural frequency parameter $\lambda_{0,0}^2 = 10.22$.

6.5.2 The Diaphragm Material Properties

A wide range of flat and corrugated diaphragms that are fabricated from metal and rubbery materials can be found in many engineering applications. The rubber-type diaphragms are of particular interest in the current design. These include for example nitrile, silicone, fluorosilicate, polyacrylate and natural rubber (NR). The latter has high tensile strength, low compression resistance, excellent resilience and tear strength, low damping losses and can be simply bonded or clamped to metal substrates. These properties make the rubber diaphragm an adequate option for applications in machine vibration. The general properties of NR diaphragm are presented in table 6.2 [98].

Table 6.2: Natural rubber properties

Density	920 – 930 kg/m ³
Poisson's ratio	0.5
Young's modulus	1.5 – 2.5 MPa
Yield stress	20 – 30 MPa

Tensile strain	22 – 23 MPa
Maximum operating temperature	120°C

6.6 Engine Design and Layout Considerations

The design of the engine components and layout arrangements was conducted so that to keep the manufacturing process simple and the cost of components low. The criteria used in the selection and sizing of the components was based on the following:

- (i) Use of off-the shelf and available parts with minor adaptation if required;
- (ii) Use of simple manufacturing process and tools available within the workshop;
- (iii) Make the design simple and components conform to numerical calculation.
- (iv) Follow a clear step-by-step assembling of the manufactured components;
- (v) Select design to be easy to assemble and dis-assemble for testing and maintenance.

Initially, a number of design sketches, structure of the dynamic components (displacer and power piston), and type of materials, manufacturability and cost were considered. To assemble all the components into a functioning engine, initial performance estimates

were carried out based on the computer model developed in *Chapter 3*. A trial and error process was followed to narrow down the many options for arranging the engine moving parts.

6.6.1 Displacer and Power Piston Assembly

The displacer assembly is one of the critical parts in the design of an FPSE. The displacer in an FPSE is often mounted on a spring and needs to be of lightweight and low mechanical friction to form a resonant system with low damping [99]. This work reviewed the many published design concepts such as that of FPSEs. One of the earlier designs, produced and refined by Sunpower Inc. uses a gas spring as the link between the displacer and power piston, as shown in Figure 6.8 (a) [99]. The gas spring is represented by a purpose-machined cavity within the power piston and provides a ground spring to the displacer through its rod. One of the limitations of using gas springs in such arrangements is that the spring constant is non-linear; it experiences significant gas hysteresis losses, and suffers from alignment and the friction.

Another configuration, which has the hallmark of an early Beale's engine, is the one shown in Figure 5.8 (b). The displacer rod protrudes through the power piston and is sprung to the engine body (ground) in the bounce space which forms part of the gas spring. In this

arrangement, the hysteresis losses are significantly reduced by the mechanical friction and sealing problems still represent a limitation of the design [99].

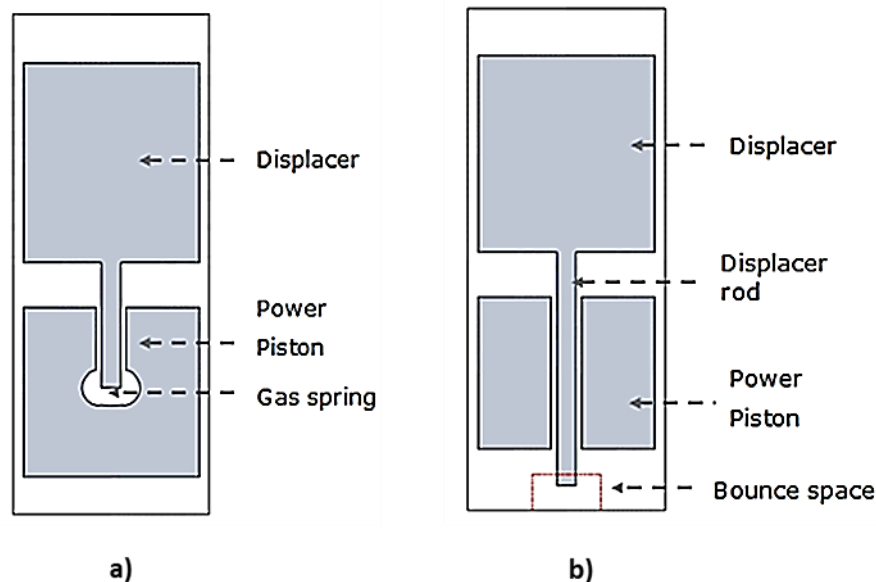


Figure 6.8: FPSE displacer-power piston arrangements (a) Displacer spring to piston (b) displacer spring to the ground

In other FPSE developments, the air spring arrangements were abandoned in favour of mechanical springs, as design and material reliability of mechanical springs improved [100]. Sunpower Inc. introduced planar springs to support both the displacer and power piston. In 1967, Cook-Yarborough introduced a flat metal spring to support the displacer and replaced the power piston by a flat metal membrane that resonated at a frequency of about 100Hz and a stroke of 1 mm. He successfully demonstrated that such arrangements are mechanically simple and of high reliability. Figure 6.9 shows a schematic of the engine.

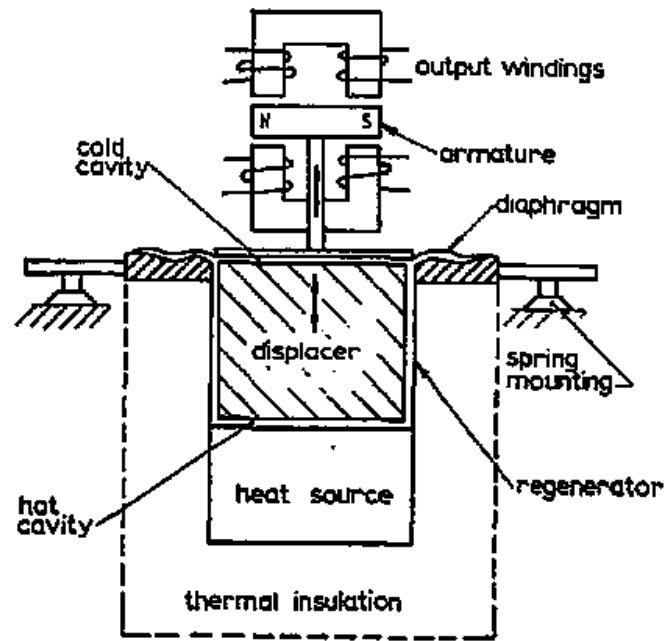


Figure 6.9: Flat metal spring for displacer support

6.6.2 The Proposed Design Configurations

From early on at the start of the project, two design options of the dynamic parts were considered. The first option was to consider the use of a cylindrical displacer and power piston and in the second case, a diaphragm type power piston was chosen.

6.6.2.1 Cylindrical Displacer/Power Piston Configuration FPSE

In this arrangement, an in-line design for the displacer assembly is outlined in Figure 6.10. The main component of this design is the use of metal bellows to support both the displacer and the power piston that are housed in a single cylinder.

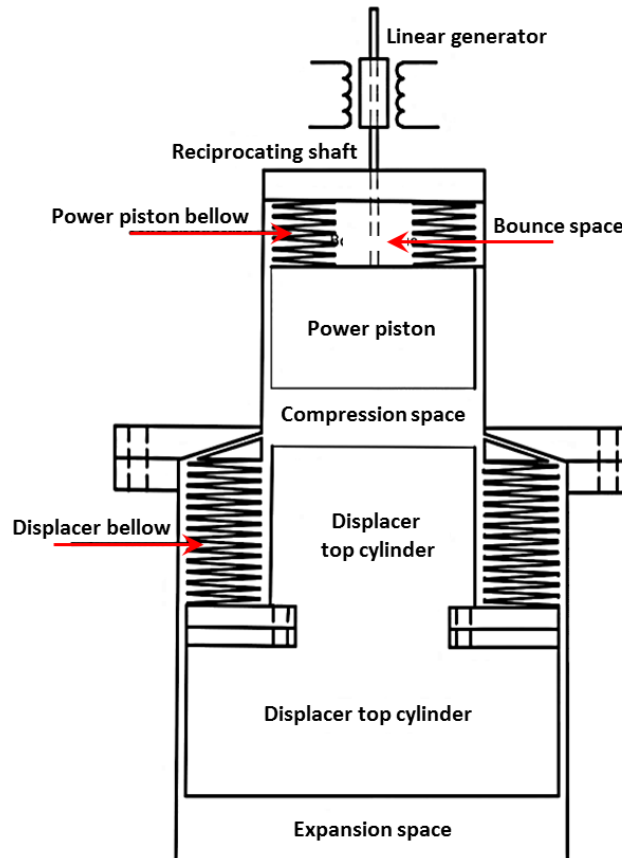


Figure 6.10: Initial design variant of the FPSE

i) Displacer assembly

The displacer was made of two cylindrical parts. The lower part is adjacent to the heat source (expansion space) and the upper part is located in the compression space. The upper displacer part is made of a smaller diameter compared to the lower part. The two parts of the displacer were fitted with an inner flange through which they are fastened and supported by a common bellows as in Figure 6.11. The bellow itself has a dedicated flange that connects the displacer assembly to the engine body. The whole displacer-bellow structure forms a mass-spring resonant system when suspended in the engine

cylinder. The air gap between the engine cylinder and the lower part of the displacer and its mounting bellow is made very small ($<1\text{mm}$) and allow the working fluid to flow between the expansion and compression space.

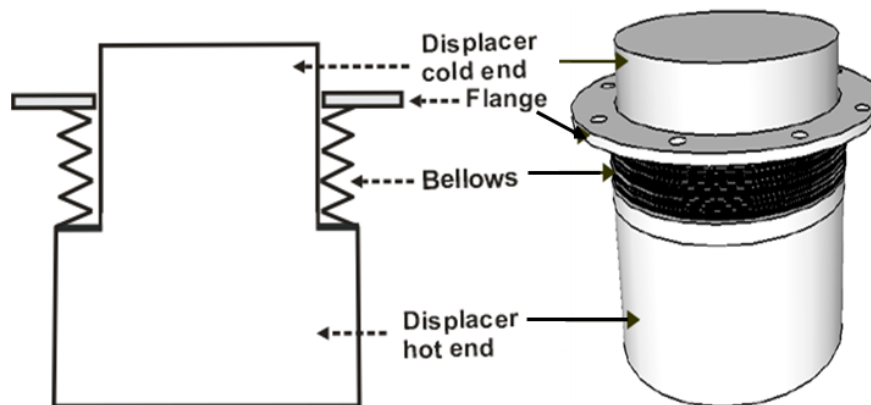


Figure 6.11: Displacer assembly

ii) Power piston assembly

In this version of the engine design, the power piston was made of a stainless steel cylinder and bellow. Like the displacer, the power piston assembly also forms a spring-mass resonate system. The power piston is connected to the load outside the working space through a liner shaft, as shown in Figure 6.12. The power piston is however, designed to be heavier than the displacer to provide the required inertia to the system. In addition, the power piston resonance frequency was selected so that it matches that of the displacer. The use of a flanged bellows to support the power piston has the advantage of eliminating working fluid leaks outside the working space without the need for the

piston seals. The bellow however, is required to withstand the pressure peaks generated within the engine.

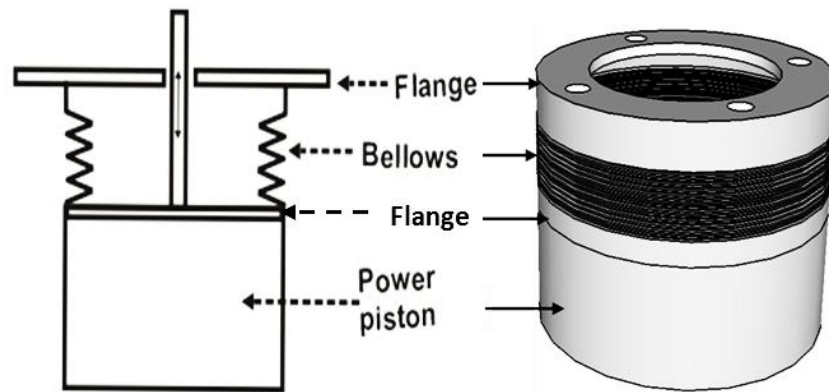


Figure 6.12: Slide power piston

6.6.2.2 Diaphragm Power Piston Configuration

The design of the displacer assembly is similar to that presented in the previous section. However, in this configuration, the power piston was replaced by a simple flat diaphragm, as shown in Figure 6.13. The diaphragm made from flexible elastomeric material, is clamped around its circumference and supports a metal rod at its centre to drives a linear load. The use of a diaphragm reduces the complications associated with sliding type pistons including working fluid leakage, seal life and friction losses. A flexure diaphragm with proper design can also achieve high operating service hours.

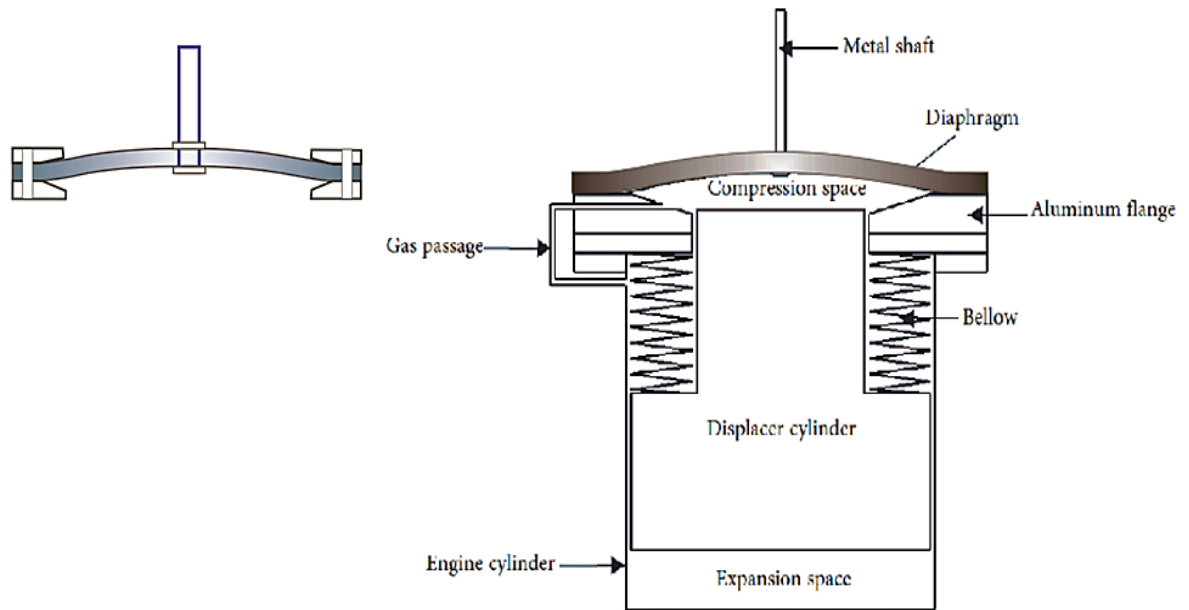


Figure 6.13: Diaphragm power piston configuration

6.7 The Proof-of-Concept Prototype Engine

From the two FPSE configurations presented in the previous section, the diaphragm type was retained for manufacturing and testing. This offers a simpler mechanical arrangement of an FPSE, cheap components and assembly process. The design integrates the novel use of bellows for displacer support and elastomeric material for the diaphragm power piston. This arrangement also makes sealing the working fluid within the working space hermetic. The final design configuration of the proposed proof-of-concept diaphragm free piston Stirling engine is shown in Figure 6.14.

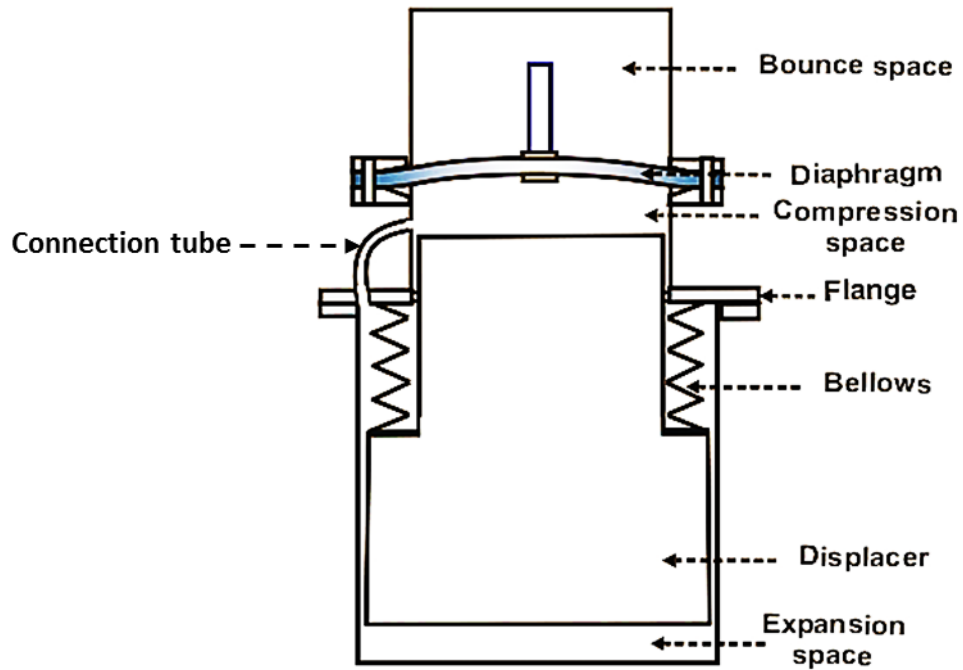


Figure 6.14: Diaphragm-Bellow FPSE

6.7.1 Prototype Components Design and Manufacturing

A proof-of-concept diaphragm FPSE was constructed. The design combined the use of thermodynamic and dynamic analysis of the engine. The thermodynamic analysis model (presented in *Chapter 4*) was used to estimate the thermal performance (heat input, heat rejection, power output, etc.) from the approximate dimensions and operating parameters (frequency, swept volume, charge pressure, temperatures, etc.) of the engine. The dynamic analysis was concerned with the oscillation of the dynamic components. Using manufacturers' data sheet (e.g., bellows, stainless steel cylinders, rubber diaphragm, etc.), the engine dimensions were adopted so that component machining was kept to a minimum and off-the-shelf parts

could be integrated into the design assembly. The manufacturing process of the components is outlined below.

a. Displacer Assembly Manufacturing

The displacer is one of the important part of the engine, as it needs to be light in weight (thin wall cylinder), use a low thermal conductivity and high operating temperature material, and have a high surface finish. In addition, the displacer in this design is mounted on an edge-welded bellows, where high mechanical alignment is required. These conflicting criteria make the manufacturing of the displacer not a straightforward process.

The main dimensions of the displacer cylinder specification are given in Table 6.3.

Table 6.3: The displacer cylinder design specification

Hot end displacer section:	
Outer diameter (OD)	95.2 mm
Inner diameter (ID)	94 mm
Height	50 mm
Wall thickness	0.5 mm

Cold end displacer section:

Outer diameter (OD)	76 mm
Inner diameter (ID)	75 mm
Height	30 mm

Threaded connector:

Outer diameter (OD)	75 mm
Inner diameter (ID)	73 mm
height	25 mm

The edge-welded bellows were also made from stainless steel material for its high temperature operation and low thermal conductivity. The specification of the bellows were obtained from the manufacturer standard sizes and specification data sheet.

The parameters to be selected, include the bellows inner and outer diameter, the number of convolution (height), stiffness of the material and deflection. The number of convolutions in particular is essential in the design of bellows as it influences the spring rate, the allowed stroke, the natural resonance frequency etc. The bellows specifications are given in Table 6.4.

Table 6.4: The edge welded bellows design specification

Materials and type	Stainless steel (316L) Edge-welded bellow
Density	7800 kg/m ³
Dimensions	
Outside diameter (OD)	95.3 mm
Inside diameter (ID)	76.2 mm
Effective area	57.75 cm ²
Wall thickness	0.13 mm
Number of convolutions	15
Nominal free length	24 mm
Axial stroke	±5 mm
Operating parameters	
Type of movement	Axial
Stiffness constant	10 N/mm
Operating pressure	1 bar
Operating temperature	450 °C
Life cycle	1,000,000 axial movement

It is shown in Table 6.3 and 6.4 that the displacer hot end cylinder and the mounting bellow were made from stainless steel 316L and have

the same outside diameter of 95.3mm. The cylinder wall was machine to less than 0.5 mm thick to reduce its overall mass. Figure 6.15 shows the unassembled displacer which is made of four parts, namely the displacer hot and cold end, the flanged bellow and the threaded connector onto which the displacer hot and cold parts screw in with the bellow sandwiched in between.

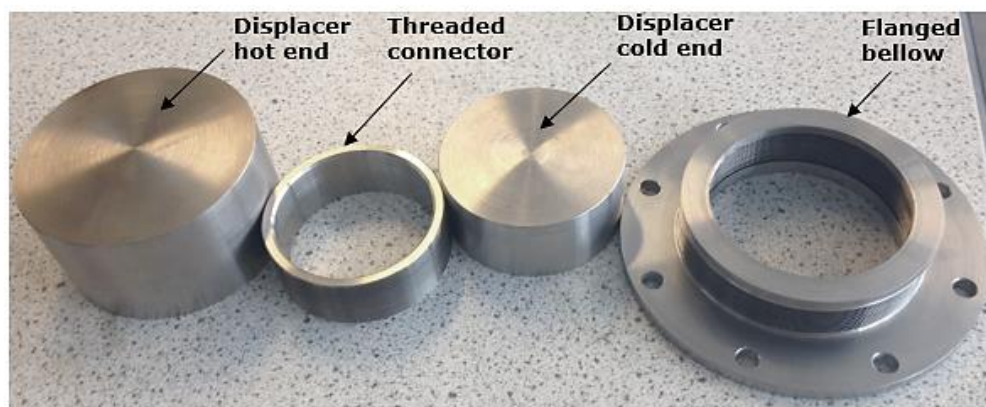


Figure 6.15: Unassembled displacer parts

Figure 6.16 shows the way the hot and cold end parts of the cylinder thread together onto each other through the threaded connector while the bellow is still not sandwiched in between.

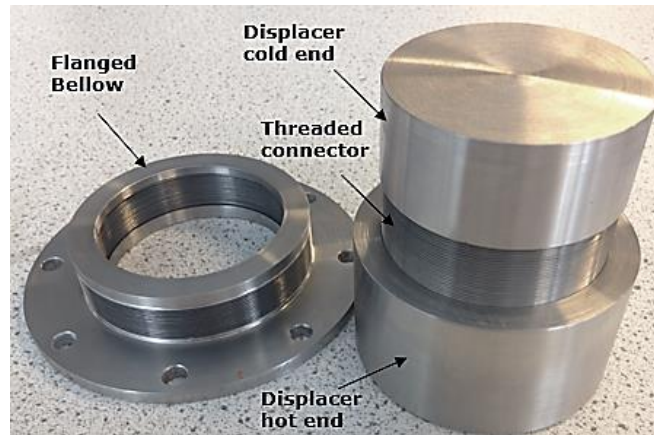


Figure 6.16: Partially assembled displacer

The full assembly of the displacer is shown in Figure 6.17 where all the parts are inserted in the correct order. The treaded connector part is clearly visible and the displacer hot end is only partially threaded in.

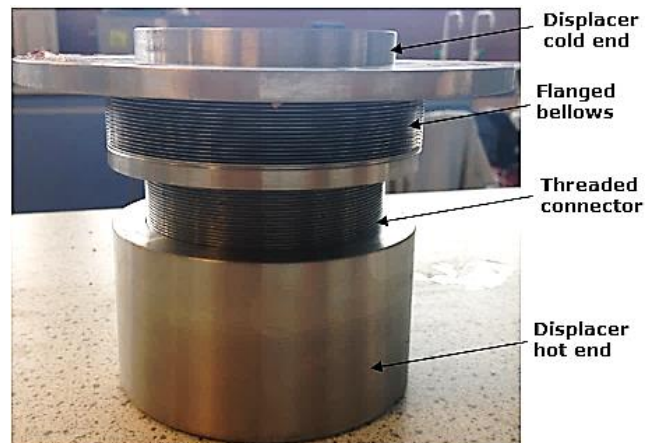


Figure 6.17: Displacer assembly (parts not fully tightened)

A fully assembled displacer block and ready to be fitted into the engine cylinder is shown in Figure 6.18. It is still important to ensure that the cold end part of the displacer is free to move inside the bellows and there is not contact between the two parts. It was also found that the

fully assembled displacer block weighs about 530 g, which is within the required mass range for a resonance frequency of 15 Hz.



Figure 6.18: Assembled displacer block

b. Diaphragm power piston manufacturing

To demonstrate the proof-of-concept prototype, a flat disk rubber diaphragm was used as the power piston. The flexure diaphragm is made from a suitable rubber hardness to give the required mechanical rigidity, stroke and minimise internal losses. The diaphragm is clamped at its circumference between two flanges through which bolts are used for tight fastening. Figure 6.19 (a, b, c) show pictures of the diaphragm shape, through holes for bolts and edge clamping flanges. It also shown that a shaft load is attached to the diaphragm at its centre to transmit the oscillation of the diaphragm directly to drive a liner load.

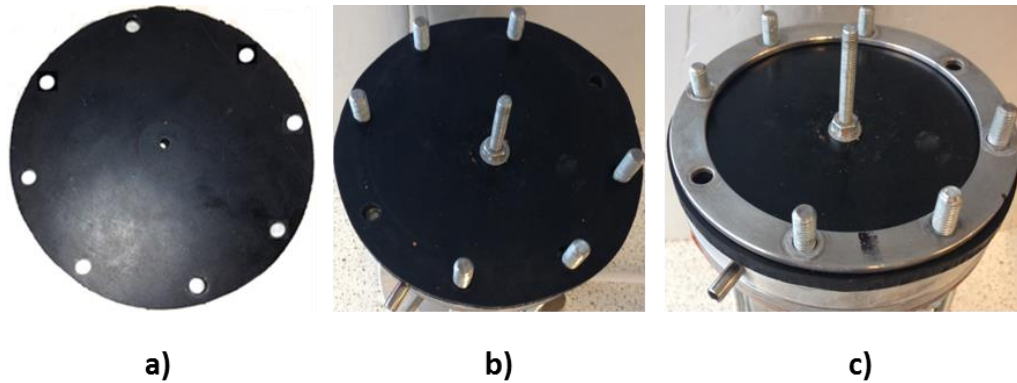


Figure 6.19: Diaphragm configuration (a) diaphragm structure (b) clamping holes (c) clamping flanges

The main physical properties of the diaphragm material are given in Table 6.5.

Table 6.5: physical properties of the diaphragm

Material	Natural rubber (NR)
Diameter	120 mm
Thickness	6 mm
Density	920 – 930 kg/m ³
Rubber Hardness	50 and 60
Operating frequency	15 Hz
Central deflection	±5 mm

In a clamped position, the diaphragm is sandwiched between two flanges. To minimise stress forces on the diaphragm material at the clamped edge, the clamping flange was designed with a sloping edge of approximately 45° as shown in Figure 6.20.



Figure 6.20: Sloped diaphragm clamping flange

c. Engine cylinder manufacturing

The engine cylinder houses the moving parts of the engine as well as allowing heat to be supplied to and rejected from the working fluid at the expansion space and compression space respectively. The cylinder is made from a stainless steel pipe machined down to a wall thickness of 1 mm. and the cylinder was made to fit around the displacer and power piston. The air clearance between the cylinder the displacer is crucial to the operation of the engine, as a large clearance will increase dead space while a tighter fit will cause mechanical friction. Therefore, the clearance between the displacer the cylinder was about 0.5 mm and the gap also plays the role of a regenerator. Figure 6.21, shows a photograph of the cylinder with a flange that sits on the flange of the bellow on which the displacer hot part is mounted. It can also be seen that a shallow groove was made into the cylinder flange

to accommodate a seal as it is clamped onto the displacer bellow flange.

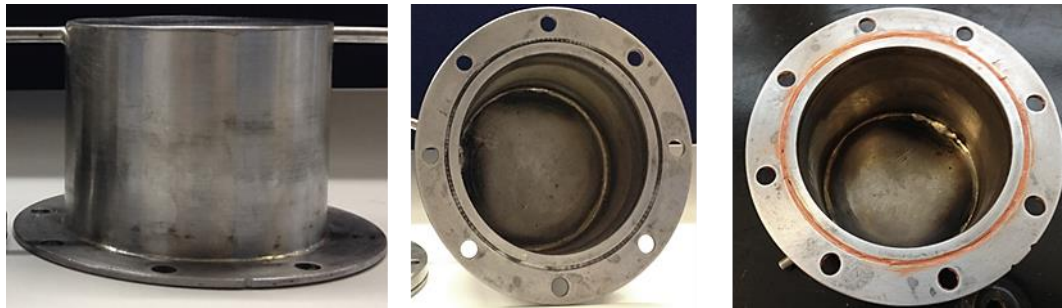


Figure 6.21: The engine cylinder

d. Heat exchangers

In this engine configuration, there is no direct communication between the expansion and the compression space, as the displacer bellow forms a physical separation between the two spaces. An external connecting pipe was used for the working fluid to shuttle between the hot and cold end, as shown in Figure 6.22.

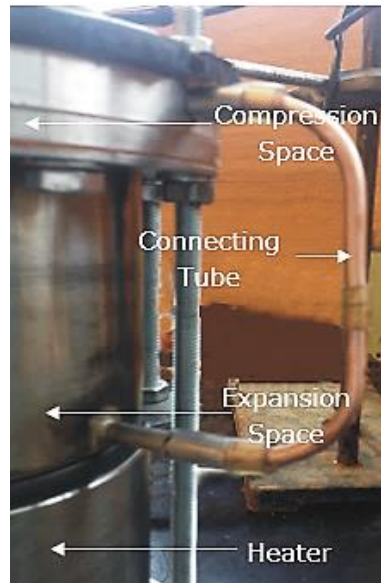


Figure 6.22: Compression and expansion space connecting tube

The heat is supplied to the working fluid in the expansion space through the cylinder's flat end plate by conduction. Equally, the heat is rejected from the compression space by natural convection through the engine cold region part. The diaphragm sloped flange described previously was machined from aluminium and offered good heat sink for heat transfer between the compression space and ambient. The expansion and compression space are made of 5 mm high cavities. The engine is intended to operate from low temperature heat sources (maximum of 300°C). Initially an electric ring heater was fitted to the hot end of the cylinder, however later it was replaced by a simple gas camping canister burner to dispense heat engine at a faster rate. An example of the electric heater and gas burner used in this work are shown in Figure 6.23.



Figure 6.23: Heat sources (a) electric ring heater (b) gas burner

Though the regenerator in a Stirling engine is an essential part to improve thermal performance, in this design the working fluid is displaced between the expansion and compression space through the air gap separating the displacer and the cylinder. The incorporation of bellows offered an extend surface area for heat storage and release as the working fluids flowed over its surface.

e. Engine Supporting Frame

The engine body was mounted on a support frame, which was made of aluminium base plate and threaded steel rod so that the height of the frame could be adjusted. Figure 6.24 shows the way the engine body is suspended using four threaded rods and a base plate, which in turn is sprung to ground.



Figure 6.24: Engine support frame

6.7.2 Prototype Assembly Procedure

The engine parts were fabricated in house with the exception of the edge-welded bellows, which were ordered from a specialist engineering company. The design, fabrication and assembly processes were kept simple. This is important for a proof-of-concept prototype as assembling and dismantling the engine parts quickly will be required for modification and test settings.

The engine assembly is not a complicated process rather a process that needs careful attention and precision in balancing the dynamic components of the engine. The step-by-step assembly process is detailed below.

a) Step 1: Displacer assembly fitting in the cylinder

The first step is to introduce the displacer-bellow block into the engine cylinder after spreading a thin sealant layer onto the cylinder flange.

This step requires careful precision to centre the two components and inserts into the cylinder-displacer clearance gap can be used to ensure uniform clearance all-round the displacer. While in this position the cylinder and displacer bellow flange are secured together with through bolts, as shown in Figure 6.25. It is essential to check that the displacer can oscillate freely inside the cylinder.



Figure 6.25: Displacer-cylinder assembly

b) Step 2: Fitting the sloped diaphragm flange

A sloped flange that adjusts the height of the cavity space between the displacer and the diaphragm is inserted next on the displacer-bellows flange through the protruding bolts. The thin layer of sealant also needs to be spread over the surface of the flange. Figure 6.26 (a) shows the sloped flange before inserting into position. Figure 6.26 (b) shows the sloped flange in position. Again, it is important to check that there is no contact between the flange and the displacer and no sealant being squeezed out into the displacer cylinder gap.

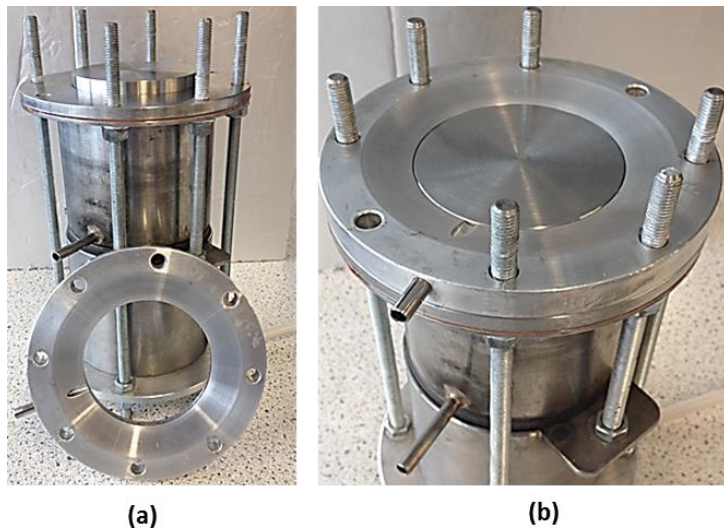


Figure 6.26: Sloped flange insertion

c) Step 3 diaphragm clamping

The diaphragm with the shaft attached to its centre is inserted through the threaded rods and pressed against the sloped flange. A second flange is then introduced on top of the diaphragm through the same rods and the whole assembly is tightened using washers and nuts on each threaded rods, as shown in Figure 6.27 (a, b).



Figure 6.27: Sloped flange insertion

Once the diaphragm is secured into position, the compression-expansion space connecting tube can be fitted. At this stage, checks for working fluid leaks can be made by pressurising the engine through the connecting tube.

The other adjustment that can be made is the distance between the engine and the base mounting plate to set the correct height for the burner from the hot end of the engine. Figure 6.28 shows a completely

assembled engine with thermocouples added to monitor the temperature at the expansion and compression space.

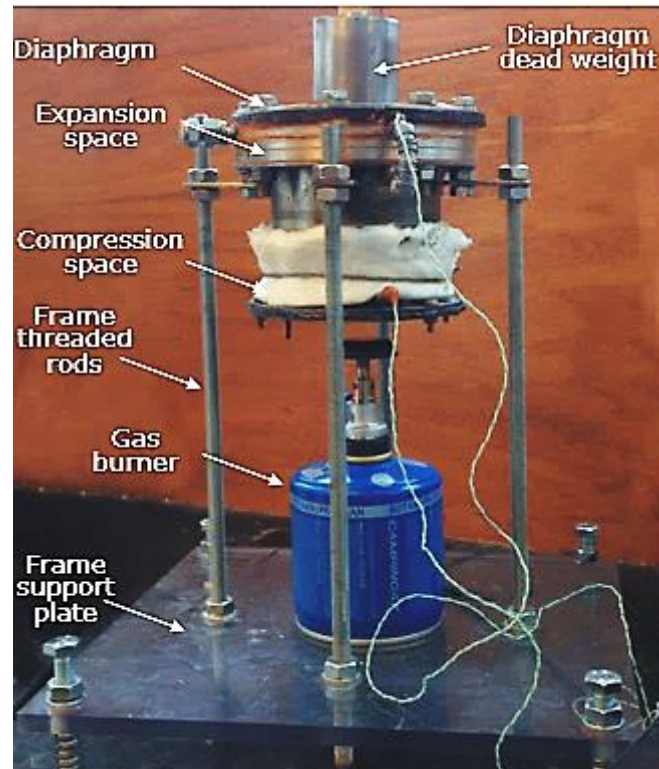


Figure 6.28: Fully assembled proof-of-concept engine

The diaphragm operating frequency can be fine-tuned by fastening additional dead weights to the diaphragm shaft. Other transducers for pressure and oscillation amplitude using a pressure transducer and a linear variable differential transformer (LVDT) respectively can also be fitted prior to testing.

6.8 Summary

In this chapter, the design and manufacturing of the proposed proof-of-concept diaphragm FPSE using edge-welded bellows and a rubber

diaphragm power piston was given in detail. The configuration, selection, and sizing of the engine parts were described in full. The selection of the engine configuration and the step-by-step manufacturing and assembling of the engine components into a full unit ready for testing was discussed and described.

CHAPTER 6

EXPERIMENTAL TESTING OF THE DIAPHRAGM FPSE

7.1 Description Of the Test Rig Setup and Experimental Procedures

In this chapter, the test rig, experimental testing procedures, modifications and tuning made to the engine design are presented in detail. The measured operating parameters of the engine are also presented. In building the test rig, stringent University guidelines on Health and Safety and testing procedures were adhered to with the necessary supporting documentation prior to commissioning the lab work.

The test rig was built and equipped with measuring instruments as shown in Figure 7.1.

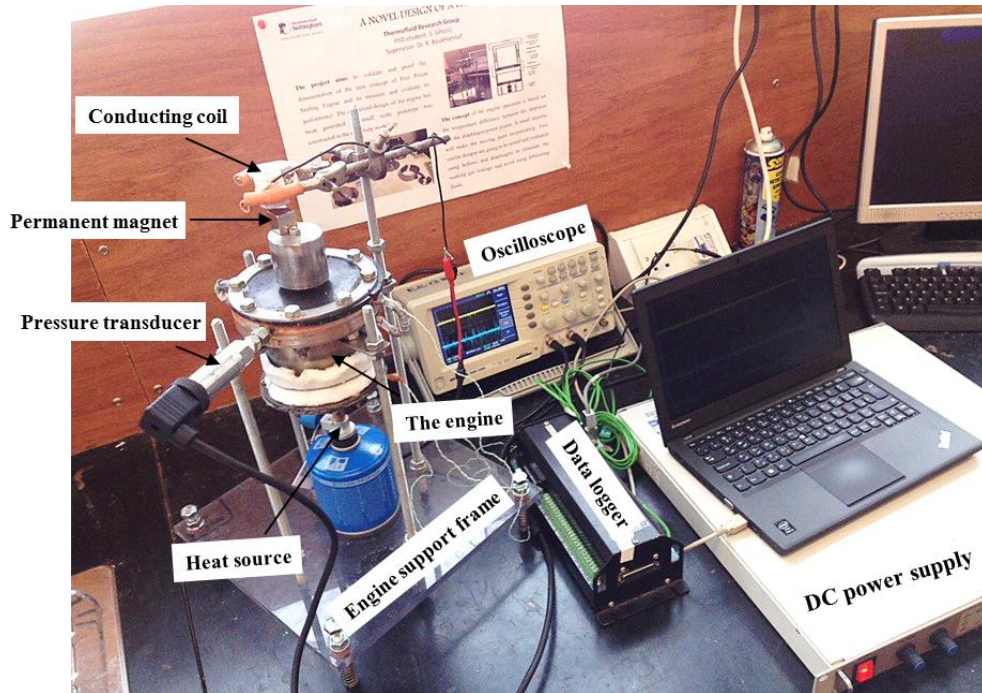


Figure 7.1: Laboratory test rig

To record the operating parameters and condition of the engine, the test rig is fully instrumented with a 2-channel oscilloscope, a pressure transducer, temperature thermocouples, a simple linear variable differential transformer (LVDT), data acquisition and a DC power supply.

7.1.1 Thermocouples

Thermocouples type (k) with temperature range of - 250 to 1250 °C were inserted into the compression and expansion space to monitor the heat source and heat sink temperature. This also served to control the heat input into the engine.

7.1.2 Pressure Transducer

A pressure transducer was connected to the compression space to measure pressure variation of the working fluid in the engine. The transducer (zero-resistive thick film ceramic sensor) is made of a Stainless steel body with a pressure rating of 0-1bar (g), accuracy of $\pm 0.25\%$, excitation voltage of 10-30V and output current signal of 4-20mA. Figure 7.2 shows an example of this pressure transducer type.



Figure 7.2: Pressure transducer

7.1.3 Liner Variable Differential Transformer (LVDT)

The oscillation amplitude of the diaphragm power piston were measured using a simple magnet/coil transducer. The transducer was calibrated manually so that the output voltage was proportional to the stroke. Knowing the effective area of the diaphragm, the swept volume amplitude and phase angle compared to pressure variation were determined.

6.1.4 Data Logger

The measured operating temperature in the compression and expansion spaces was acquired using DataTaker DT500 is shown in Figure 7.3. The data was recorded as excel files for analysis.



Figure 7.3: DT 500 Data logger

7.1.5 DC Power Supply

A DC power supply (0-40V) and current (0-30A) was used to supply regulated DC voltage to the pressure transducer.

7.1.6 Oscilloscope

The working fluid pressure variation and the oscillation amplitude of the diaphragm were recorded using a 2-channel oscilloscope. The oscilloscope allowed visualising the amplitudes of oscillation and pressure, frequency of oscillation and phase shift. The use of two channels also allowed visualising the PV diagram of the engine as a way of measuring the power output.

7.2 Engine Start up Procedure

When assembling the engine it is vital that the reciprocating components move freely in the cylinder. The initial tests were conducted with the engine cold (heat source off) to check for pressure leaks, the displacer and diaphragm oscillation and working fluid pressure. After being satisfied that all the components were assembled correctly and no potential working fluid leaks when the engine is cold, heat was applied gradually to the hot end (expansion space of the engine). Two types of heat sources were explored: an electric ring heater and a gas burner.

The electric ring heater was fastened and in direct contact with the hot end plate of the engine with adequate electrical insulation. Using an electric heater and variable voltage supply provide an accurate method of measuring heat input into the engine. The temperature of the heater was increased to 200°C and the initial tests were mainly related to investigating the effect of increasing the temperature of components such as the engine cylinder, displacer, sealants, and the diaphragm. This testing exercise revealed working fluids leaks at diaphragm periphery and around the sloped diaphragm flange. At this point, attempts to start the engine were also initiated by imparting a sudden force to the engine body or diaphragm shaft but to no avail.

7.3 Modifications and Refinement

At this stage of the project, all efforts were concentrated on re-checking the engine's operating parameters and that all the calculations were correct as well as the method of assembling of the components. The tests on the engine continued for some time and during which assembling, heating, cooling and disassembling the engine was a daily routine. Each of these was aimed at understanding and eliminating any obvious failures in design and assembling of the engine parts.

Difficulties in the alignment of the reciprocating components is one of the most common difficulties in the engines assembly, particularly with its tight tolerances. Hence, some minor improvement to ensure good alignment of the displacer in the cylinder to keep mechanical friction low and sealing between flanges using high temperature sealant or gasket were made. Figure 7.4 shows some of the modification made to the cylinder hot end cap in which a groove in the flange was machined and a high temperature gasket was used.

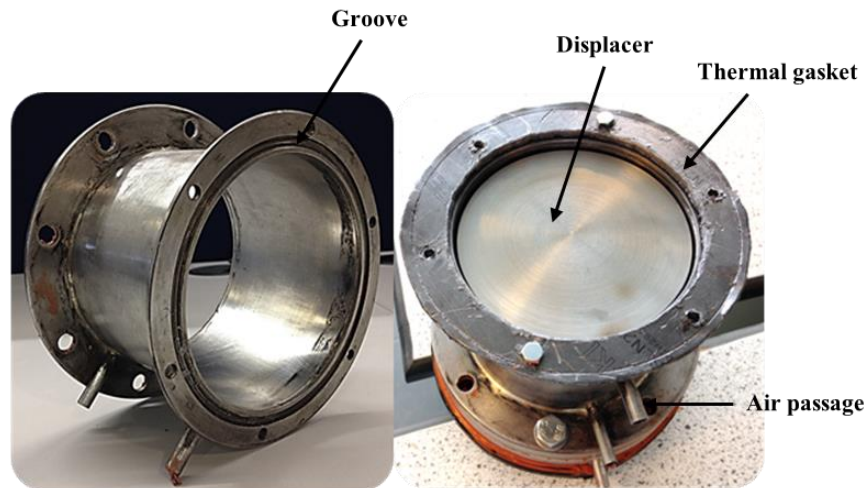
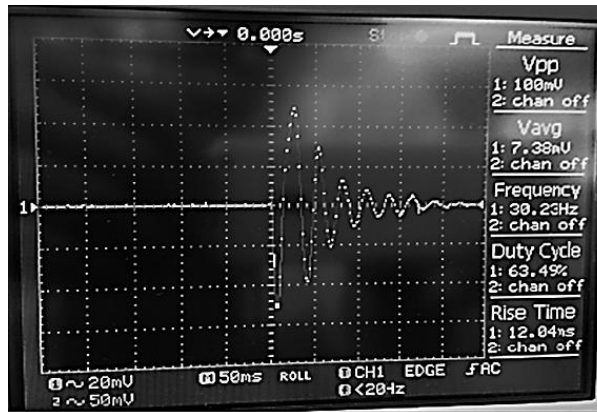
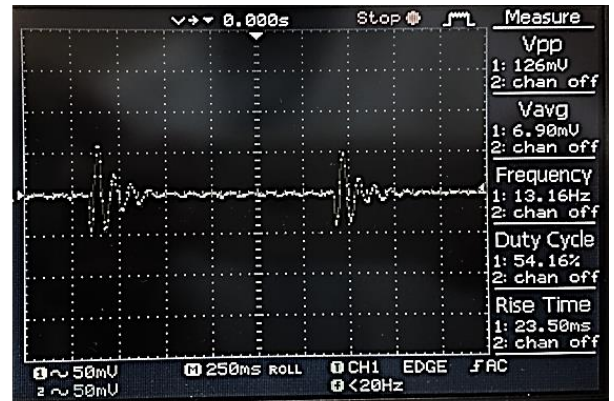


Figure 7.4: Engine cylinder hot end

With these changes, the engine parts were assembled again and heat applied using the electric ring heater. The engine hot end was heated up to 250 °C. Attempts to start up the engine were initiated but with limited success. It was however, noticeable that there are encouraging sign that the modification made were appropriate in that the engine oscillation could be seen building up before decaying after a few oscillation. An oscilloscope was used to capture the diaphragm oscillations shown in Figure 7.5, after the initial improvements to the engine. In Figure 7.5 (a) the engine started to show signs of oscillation build up at a temperature of 230°C but lasted for only a short time (just over 25 ms). Figure 7.5 (b), the temperature of the hot end was increased to 250°C and the build-up of oscillation increased in amplitude as well as in time (400 ms).



(a)



(b)

Figure 7.5: Diaphragm power piston oscillation (a) $T_h=230^\circ\text{C}$ (b) $T_h=250^\circ\text{C}$

As one of the objectives of the design is to operate the engine at low temperature, below 300°C , the electric ring used to supply heat reached its heat capacity limited at about 250°C . However, it was also found that the electric heater has a high thermal constant, in that, it takes longer to heat up the engine. This contributed to dissipating heat throughout the engine body and reducing the temperature gradient between the expansion and compression space at start of the engine. This has led to further ideas in terms of modifying the working fluid connection tubes between the compression and expansion space. Figure 7.6 shows the main mechanical modification that was made to the original design was to move the working fluid communication port between the compression and expansion space.

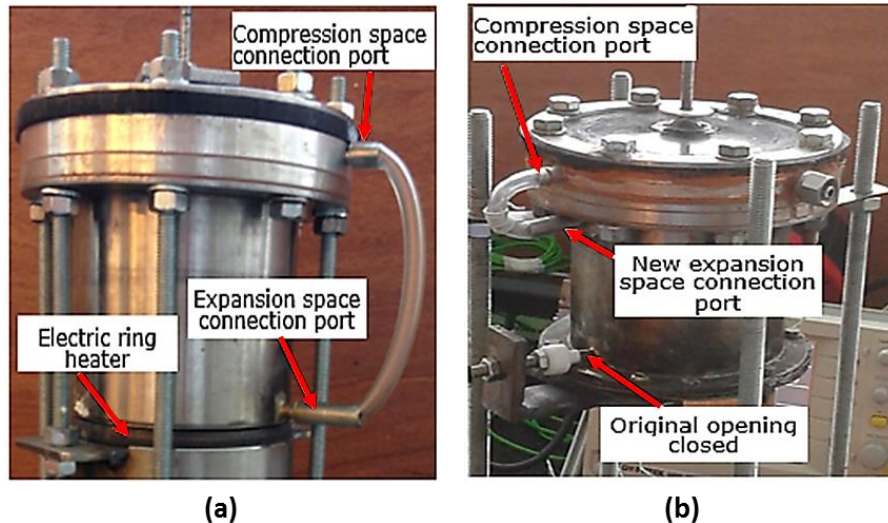


Figure 7.6: Modified working fluid connection tube location (a) expansion space (b) compression space

In Figure 7.6 (a) the working fluid is shuttled between the expansion and compression space externally using a connecting tube whereas in Figure 7.6 (b) the connection is located at the cold end. This change in design allows the working fluid to flow through the cylinder-displacer annulus and then into the connecting tube and vice versa. This has the effect of advantage of using the annulus regeneration effect as well as reduce the dead volume in the engine.

Further tests were carried out to attempt to operate the engine. The electric heater was at full power rating ensuring a maximum temperature of 250°C was applied at the expansion space. The changes brought some success in that the diaphragm oscillations were sustained for longer (3000 ms) albeit eventually die out, as shown in Figure 7.7 (a, b).

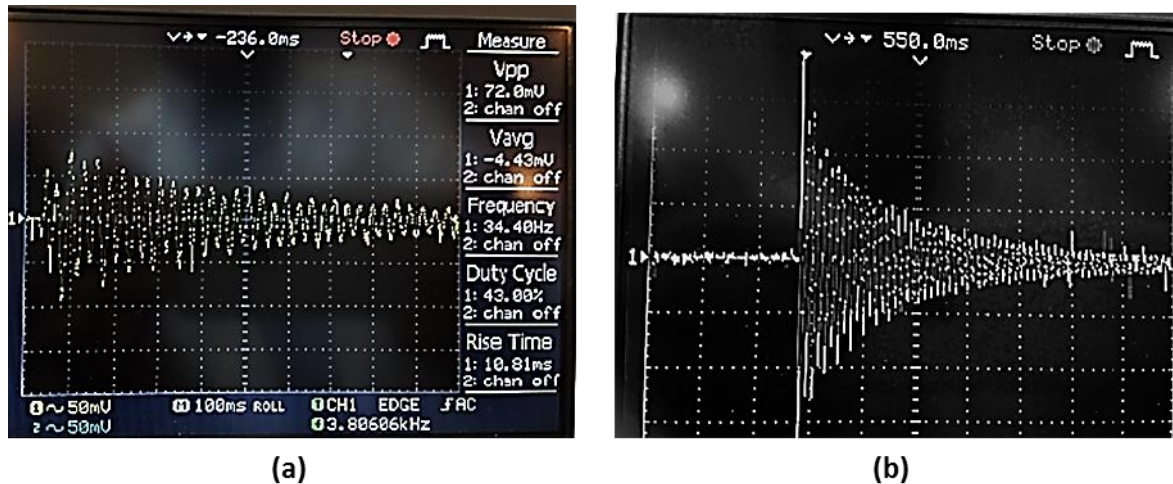


Figure 7.7: Diaphragm oscillation after modifications (a) $T_h=230^{\circ}\text{C}$ b) $T_h=250^{\circ}\text{C}$

It was, however, immediately noticed that the compression space temperature increased rapidly and exceeding 50°C in less than 10 min, which decreases the temperature gradient between the expansion and the compression space. Even so, according to the simulation results, the temperature difference is sufficient to start the engine.

To remedy the slow heat transfer from the electric heater to the engine working fluid, a small gas burner was tried instead. This allowed faster heat transfer and the effect was immediate in that the engine sustained its oscillation until it was stopped. The experiment was also repeated many times. Figure 7.8 shows the initial sustained movement and pressure oscillations of the engine. Both signals, stroke signal (yellow line) and the pressure variation (red line) are of weak amplitude and the engine barely able to operate continuously.

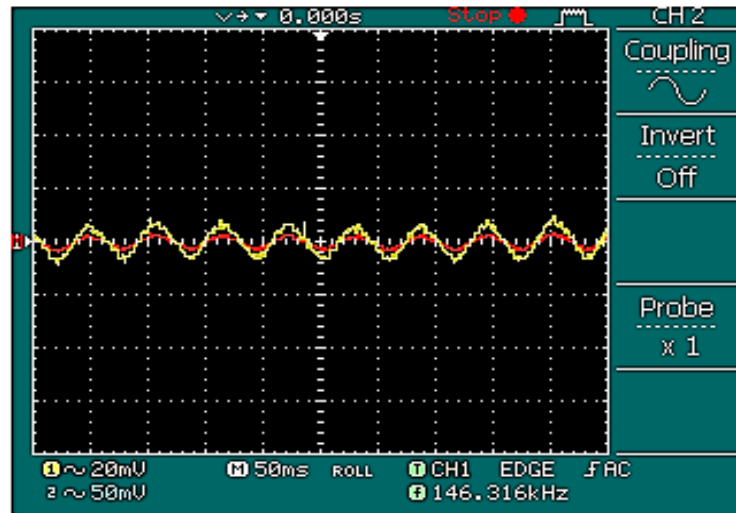


Figure 7.8: Initial sustained diaphragm and pressure oscillation (red line: stroke, yellow line: pressure, expansion space temperature $T_h=250^\circ\text{C}$)

7.4 Diaphragm Operating Frequency Tuning

Once a working base has been established that the engine can sustain its oscillation, the following steps are to enhance the performance of the engine by analysing the operating parameters of each component and overall settings. The main is to reflect the design parameters set out in the simulation model. One of the first parameters to tune was the operating frequency of the diaphragm. It is documented [52] that the optimum operation frequency of a free piston Stirling engines is obtained when the oscillating components (i.e., the displacer and diaphragm) have equal resonance frequencies. Since the displacer resonance frequency cannot be changed without altering the mass of the displacer or the stiffness of the bellows, the diaphragm resonance frequency, on the other hand, can be easily tuned. This is mainly

performed by attaching lump weights to the diaphragm centre shaft, as shown in Figure 7.9.

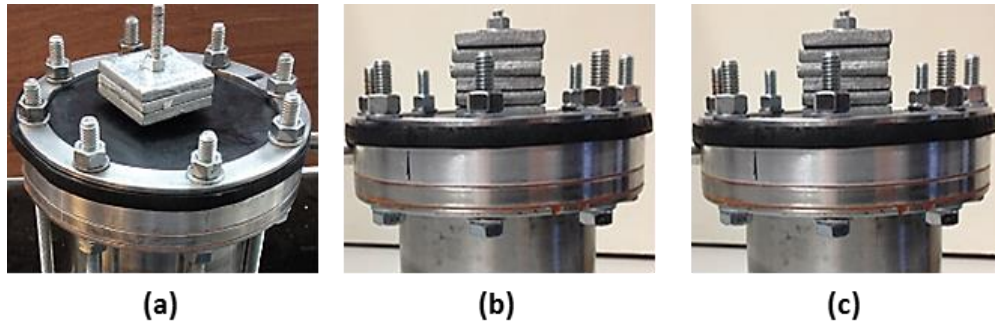


Figure 7.9: Diaphragm resonance frequency adjustment (a) $m=100$ g (b) $m=400$ g (c) 500g

The tuning of the diaphragm resulted in excellent start-up of the engine and increase in oscillation amplitude and pressure variation. Different weight were tested and it was found that a mass of 450g gave the closest resonance frequency, which is 15.6 HZ, to design frequency of 15 Hz. With the hot end temperature maintained at 270 °C and the compression space temperature at 35 °C, the diaphragm oscillation amplitude reached ± 1.25 mm as shown in Figure 7.10. The effective swept volume is calculated and found approximately of ≈ 32 cc.

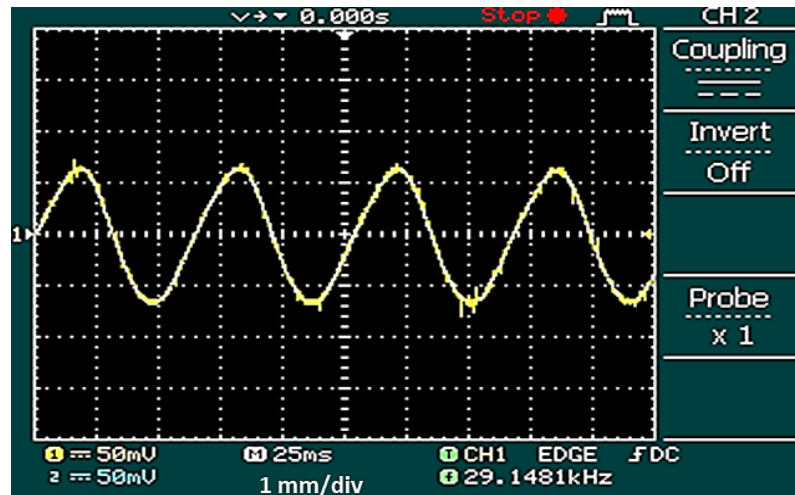


Figure 7.10: Diaphragm oscillations (freq.=15 Hz, $T_h=270\text{ }^{\circ}\text{C}$ and $T_c=35^{\circ}\text{C}$)

One further tuning that was considered is the engine mounting frame supporting springs. The support frame was mounted on helical compression springs where the stiffness of the springs can be changed. As the displacer is solely sprung to the engine body, the vibration of the engine frame has a direct mechanical feedback on the displacer oscillation amplitude and the frequency of operation increased from 15.6 HZ to 16Hz. Figure 7.11 shows the engine's base frame support structure.

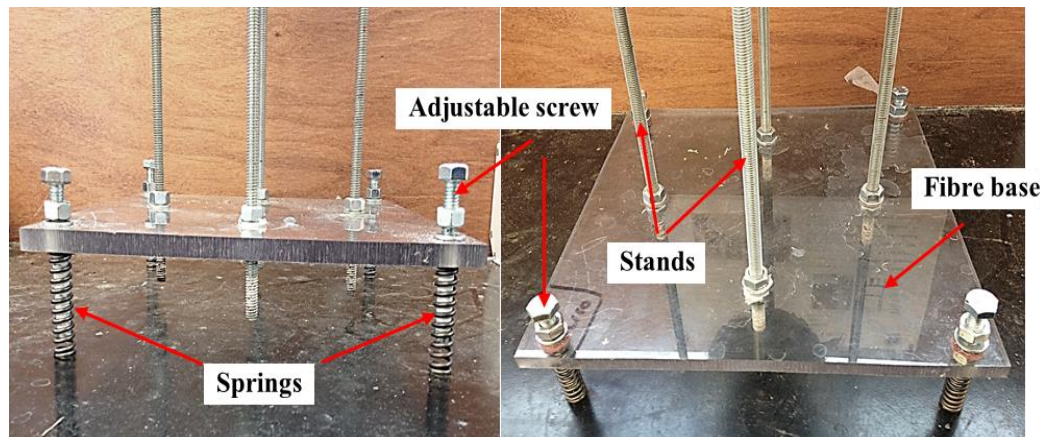


Figure 7.11: Engine supporting frame

7.5 Steady State Test Results

With all the fine-tuning complete, further measurements of the engine operating parameters were conducted. In addition, the use of a gas burner allowed heating the engine hot end to higher temperature up to 350 °C. The diaphragm oscillation amplitude grew even higher to $\pm 2\text{mm}$. Figure 7.12 shows the recorded new oscillation amplitude.

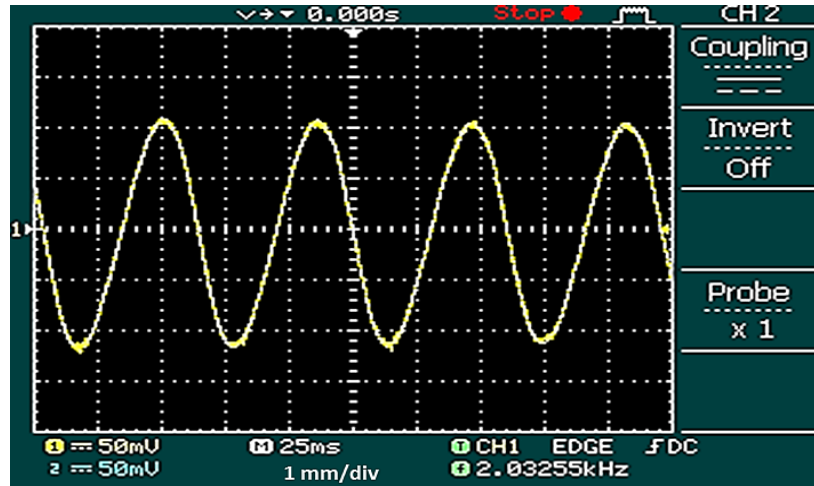


Figure 7.12: Diaphragm oscillation (freq.=16 Hz, $T_h = 300\text{ }^{\circ}\text{C}$)

A further test was conducted at a higher hot end temperature, $T_h = 350\text{ }^{\circ}\text{C}$, and a marked improvement in the engine oscillation amplitude was recorded, as it is shown in Figure 7.13. As it can be observed that the total diaphragm stroke of 7.5 mm was recorded, corresponding to a calculated swept volume of $\approx 95\text{ cc}$.

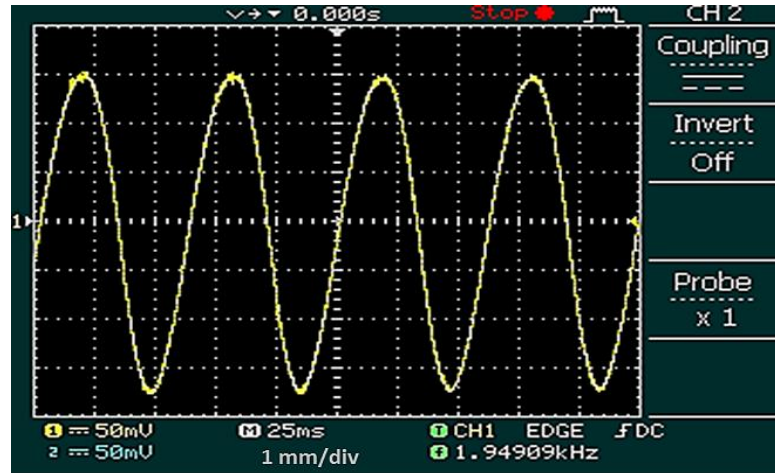


Figure 7.13: Diaphragm oscillation (freq.=16 Hz, $T_h = 350\text{ }^{\circ}\text{C}$)

7.5.1 Swept Volume and Pressure Variation

In the following measurements, the working conditions of the engine were held constant at the values presented in table 7.1.

Table 7.1: The working conditions of the engine during the test

Operating parameter	Value	Unit
Hot end temperature T_h	300	$^{\circ}\text{C}$
Compression space T_c	35	$^{\circ}\text{C}$
Engine pressure P	1	bar
Operating frequency	16	Hz

The stroke and hence the volume swept of the diaphragm and pressure variation of the working fluid in the engine were recorded as shown in Figure 7.14. It is shown that the diaphragm oscillation amplitude is about $\pm 6\text{mm}$ (12 mm stroke), the pressure amplitude

± 1.5 kPa, and the phase shift between the diaphragm motion leading the pressure by approximately 80° .

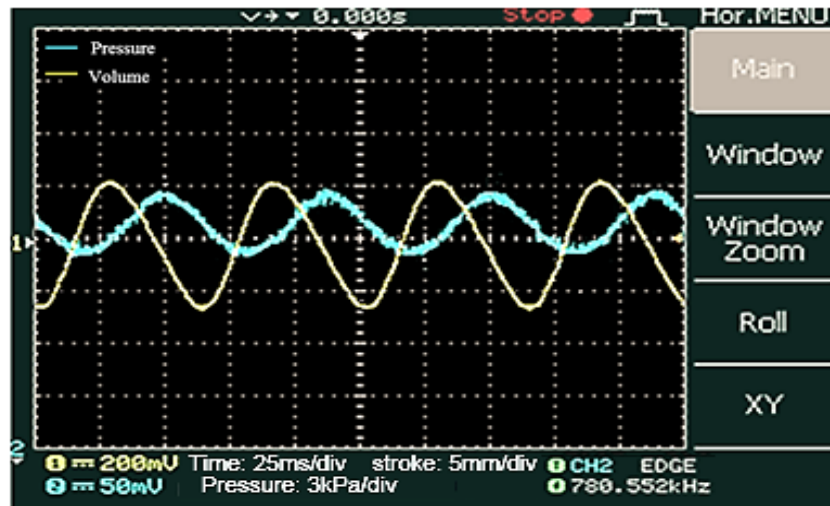


Figure 7.14: Pressure and volume variation

7.5.2 PV Diagram

The power output of the engine is approximated by measuring the area of the established a PV diagram. Using a 2-channel oscilloscope the pressure-volume diagram can be directly visualised as it's shown in Figure 7.15. The enclosed area of the diagram has the unit of work developed by the engine. From these preliminary results, it was shown that the engine power output is in the order of 0.76 W.

In this work, the tests have been undertaken on unloaded engine, however, the shaft attached to the diaphragm lifted up to 700g weight. In order to obtain realistic and more accurate results, the engine can be loaded by a mechanical or an electric load. However, due to the

time constraints, integrating a load to the engine will need to be addressed to further work.

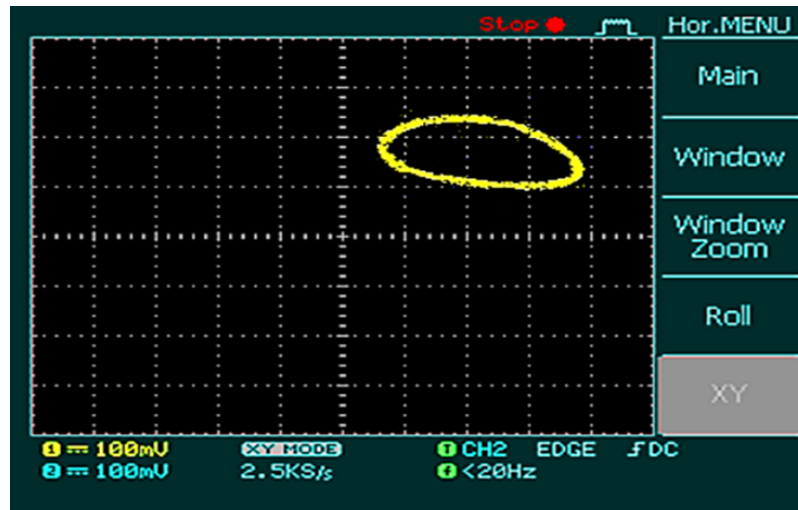


Figure 7.14: Engine PV diagram

7.6 Summary

In this chapter, a detailed account of the procedure of testing, modifying and tuning the engine components was presented. The work included overcoming initial difficulties to making the engine start and the improvements carried out to increase reliability of starting the engine and the oscillation amplitude and pressure of the working fluid. In addition, the testing methodology and instruments used was explained.

Chapter 8

CONCLUSION AND RECOMMENDATIONS FOR FURTHER WORK

8.1 Introduction

The existing literature shows that Stirling engine could be a viable technology for the conversion of thermal energy to mechanical power in remote sites particularly for the small scale power applications such as power generation and water pumping etc. This work mainly concentrates on Free Piston Stirling engine type, for its mechanical simplicity. The recent research is devoted on improving the thermal efficiency of the FPSE in order to enhance its output and hence its performance, therefore, different designs were constructed and developed. However, obtaining adequate performance at high deference temperature and high working pressure is achievable. The main merits of this research work was to design, construct and test a new FPSE mechanical configuration using low temperature heat sources. A combination of flexible metal bellows and rubber diaphragm were used for the displacer and the power piston assemblies respectively. Incorporating the flexible metal bellows and the

diaphragm provided the design simplicity by eliminating both the need for mechanical seals and working fluid leakage

In order to understand the thermal process of the Stirling cycle, approximating physical considerations of the small-scale proof of concept and roughly predict the thermal performance of the new design, a second order thermodynamic mathematical model was developed using Matlab software. The formulation of the model considers an unsteady state of energy, momentum and mass conservation equations in a series of engine zones occupied by the working fluid. The model predicts a moderate thermal performance over the range of temperatures considered (up to 300 °C) with efficiency lower than 10%. A small-scale proof-of-concept engine was then designed and manufactured as detailed in chapter 5 and 6.

A laboratory test rig was used to evaluate the prototype. After the initial tuning of the different parts of the engine, it was demonstrated to operate successfully and self-start when a temperature gradient of about 270°C was established between the expansion and compression space. The engine operating parameters such as the displacer and power piston stroke, operating frequency, pressure and temperature variation, power output using the PV diagram, etc.

8.2 Novelty Of the Research and Contribution To Advancing FPSE Design

The novelty and the contributions of this research are summarised as follows:

- Introducing a novel component of purpose designed edge welded bellows for the displacer instead of using conventional mechanical springs or/and gas spring.
- Incorporating the elastomer diaphragm for the power piston and considering the combination of edge welded bellow and rubber diaphragm to form new engine configuration.
- Improving and modifying the new design for assembly simplification and to demonstrate the operating parameters.
- Building a proof-of-concept Free-piston Stirling engine prototype working at atmospheric pressure and low-temperature difference and low frequency.
- Testing the new configuration and demonstrating its viability as a thermomechanical converter for power generation at low temperature heat sources and low frequencies operating at atmospheric pressure.
- Validate the initial design assumptions and hypothesis.
- Measuring the performance of the prototype, establishing conditions of stable operation, and self-starting. The latter

particularly proved challenging as tuning the engine moving masses is an essential process for reliable self-starting (eliminating dry friction and tuning the dynamics of the engine frequency).

- The engine operation frequency was determined and found to best tuned at around 15 Hz with compression space pressure fluctuation of about 120 Pa, a phase shift of 80° and swept volume of the power piston of 4.2 cm^3 .
- Measuring instruments using oscilloscope was used to evaluate the power output from the PV diagram of the working fluid and it was approximately 7.7 mW.

8.3 Future work

This results reported in this thesis will constitute a good basis to build on for future research to improve the design performance. This work was initiated as speculative research to investigate the viability of small-scale thermal power generators for remote areas in developing countries. However, there is a wide scope to progressing the results of this research of both the computer analysis model and the design and tuning of the engine components. These could include the following:

- i) Integrating a dynamic analysis model of the engine to take into account the non-linearities of the pressure losses in different parts of the heat exchangers;

- ii) Incorporating better performance heat exchange elements (heater, regenerator and cooler);
- iii) Redesign of the diaphragm to increase the volume swept or could be replaced by a small bellow;
- iv) Consider increasing the working pressure to improve power output or increasing the swept volume;
- v) Design and incorporate a linear mechanical load (water pump, fan, generator, etc.) and test the engine under different load conditions. Figure 7.1 shows an example of the engine connected to a linear electric generator and with the load fully enclosed within a pressurised chamber.

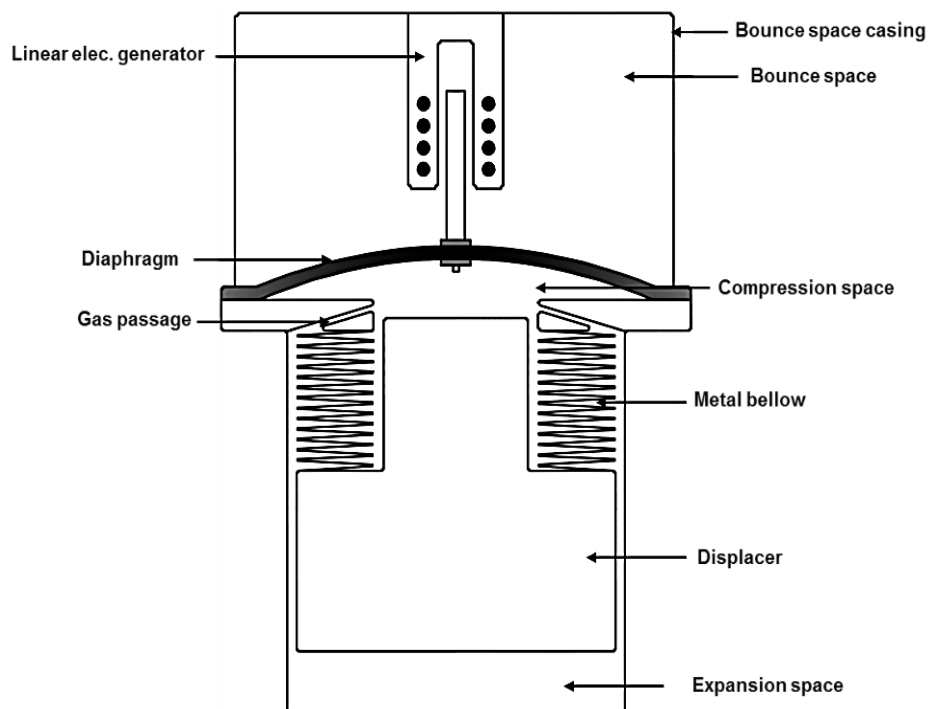


Figure 8.1: The prototype engine adopted to linear electric generator

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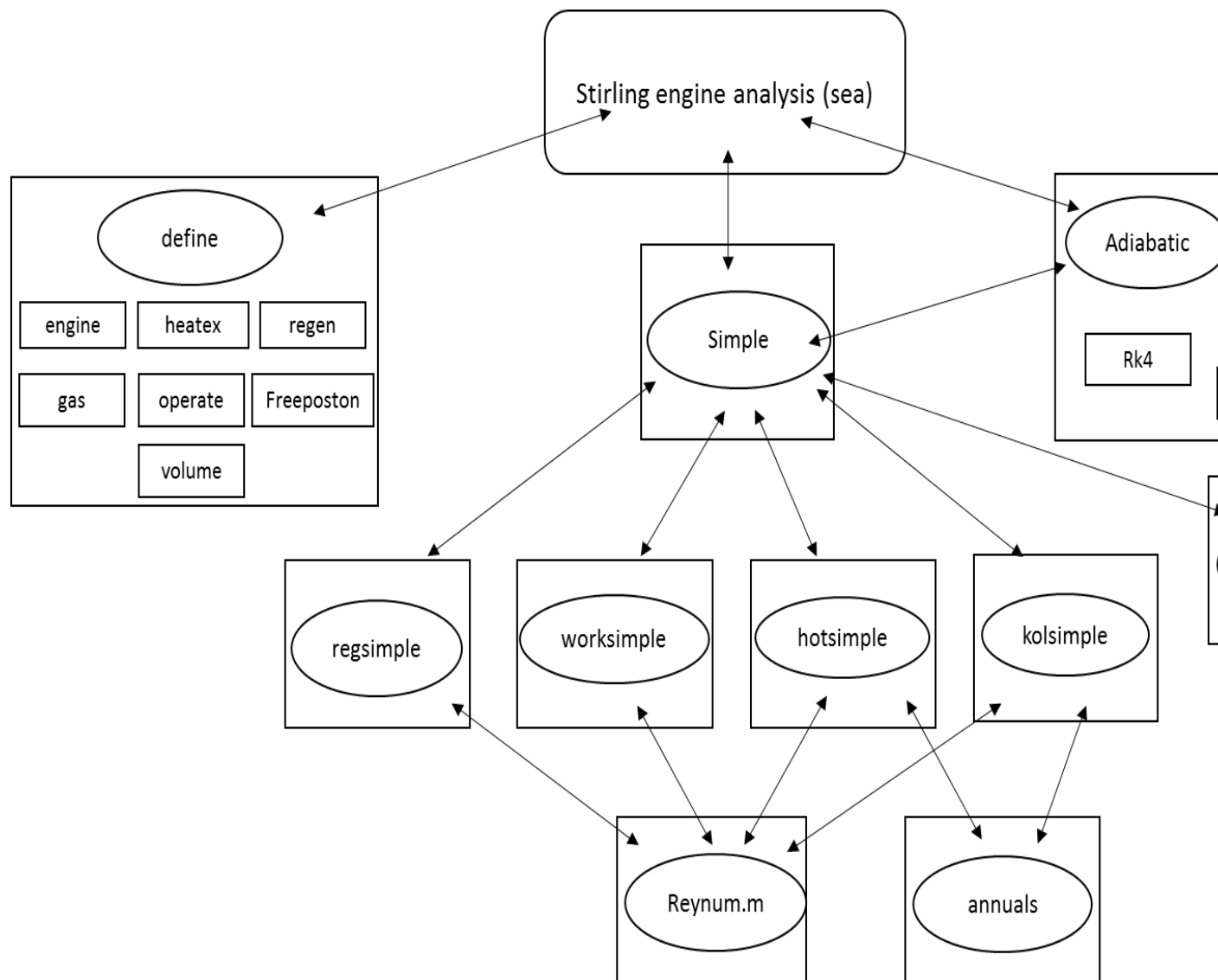
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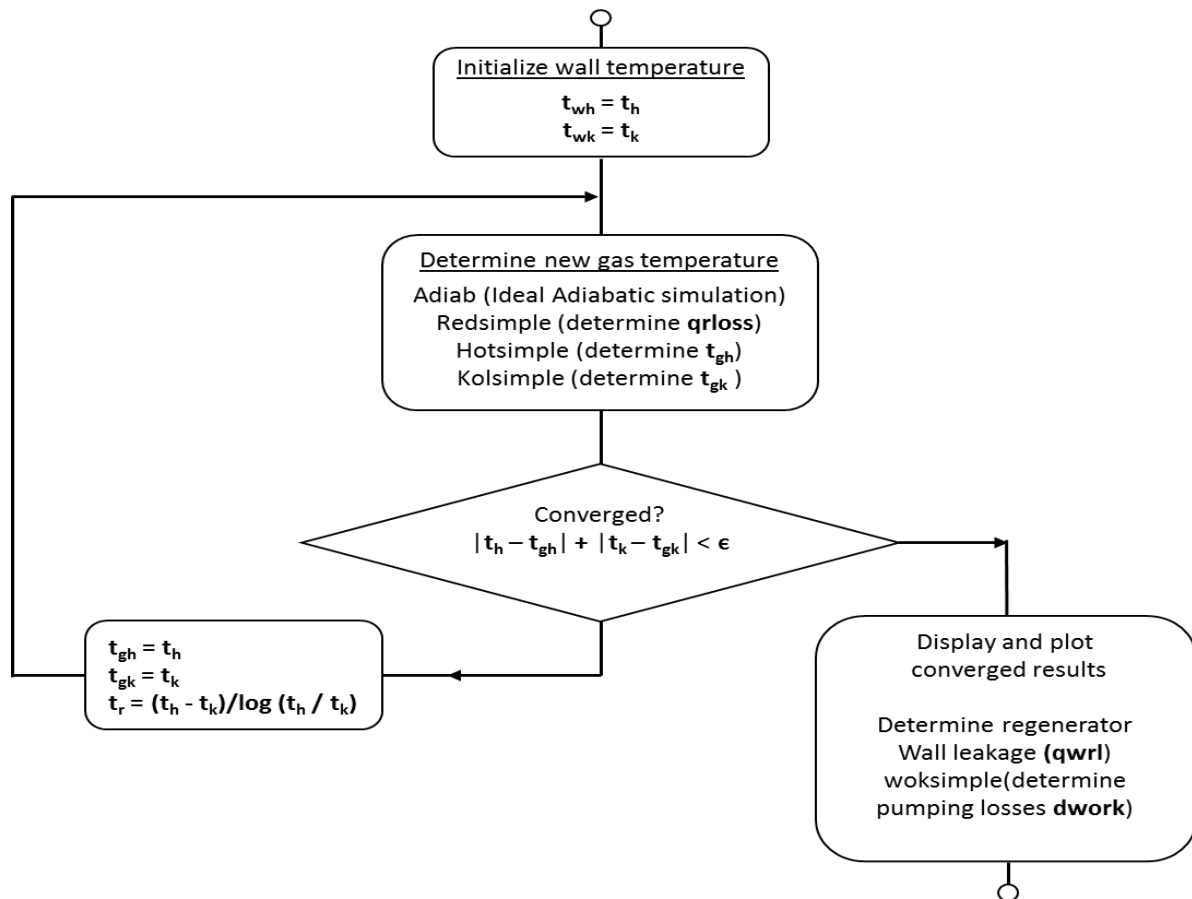
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Appendix A: Matlab computer program



The program functions



The simple algorithm

```

% sea (stirling engine analysis) - main program

% Row indices of the var, dvar arrays:

TC = 1; % Compression space temperature (K)
TE = 2; % Expansion space temperature (K)
QK = 3; % Heat transferred to the cooler (J)
QR = 4; % Heat transferred to the regenerator (J)
QH = 5; % Heat transferred to the heater (J)
WC = 6; % Work done by the compression space (J)
WE = 7; % Work done by the expansion space (J)
W = 8; % Total work done (WC + WE) (J)
  
```

```

P = 9; % Pressure (Pa)
VC = 10; % Compression space volume (m^3)
VE = 11; % Expansion space volume (m^3)
MC = 12; % Mass of gas in the compression space (kg)
MK = 13; % Mass of gas in the cooler (kg)
MR = 14; % Mass of gas in the regenerator (kg)
MH = 15; % Mass of gas in the heater (kg)
ME = 16; % Mass of gas in the expansion space (kg)
TCK = 17; % Conditional temperature compression space / cooler (K)
THE = 18; % Conditional temperature heater / expansion space (K)
GACK = 19; % Conditional mass flow compression space / cooler (kg/rad)
GAKR = 20; % Conditional mass flow cooler / regenerator (kg/rad)
GARH = 21; % Conditional mass flow regenerator / heater (kg/rad)
GAHE = 22; % Conditional mass flow heater / expansion space (kg/rad)
V = 23; % Total volume (m^3)
% Size of var(ROWV,COL), dvar(ROWD,COL)
ROWV = 23; % number of rows in the var matrix
ROWD = 16; % number of rows in the dvar matrix
COL = 37; % number of columns in the matrices (every 10 degrees)
%=====
global tk % cooler temperatures [K]
global tr % regen temperatures [K]
global th % heater temperatures [K]
global vk % cooler void volume [m^3]
global vr % regen void volume [m^3]
global vh % heater void volume [m^3]

```

```

global freq % operating frequency [Hz]

define;

% choice = 'x';

% while(~strncmp(choice,'q',1))

% fprintf('Choose simulation:\n');

% choice = input('a)diabatic, s)imple q)uit: ','s');

% if(strncmp(choice,'a',1))

% [var,dvar] = diabatic;

% else if(strncmp(choice,'s',1))

[var,dvar] = simple;

fprintf('quitting simulation...\n');

```

```

unction engine

% Define the free piston engine configuration geometric parameters.

% global engine type % free piston type)

global new fid % new data file

freepiston;

```

```

function freepiston

% New Free-Piston Stirling Engine design (FPSE)

% Ghozzi 13/02/2014

global vclc % compression clearance vols [m^3]

global vcle % expansion clearance vols [m^3]

global vswc % compression swept volumes [m^3]

```

```

global vswe % expansion swept volumes [m^3]
global alpha % phase angle advance of expansion space [radians]
% global dcomp dexp % diameter of compression/expansion pistons [m]
% global acomp aexp % area of compression/expansion pistons [m^2]
% global yminc % minimum piston vertical displacement [m]
% global ymaxc % maximum piston vertical displacement [m]
% global ymine % minimum displacer vertical displacement [m]
% global ymaxe % maximum displacer vertical displacement [m]
% global new fid % new data file
fprintf('==== FPSE Data configuration =====\n \n');
if(strncmp(new,'y',1))
vclc = input('enter compression space clearance volume (vclc) [m^3]: ');
vswc = input('enter compression space swept volume (vswc) [m^3]: ');
vcle = input('enter expansion space clearance volume (vcle) [m^3]: ');
vswe = input('enter expansion space swept volume (vswe) [m^3]: ');
phase = input('enter expansion phase angle advance (alpha) [degrees]: ');
alpha = phase * pi/180;
fprintf(fid, '% .3e \n', vclc);
fprintf(fid, '% .3e \n', vswc);
fprintf(fid, '% .3e \n', vcle);
fprintf(fid, '% .3e \n', vswe);
fprintf(fid, '% .3f \n', alpha);
else
vclc = fscanf(fid,'% e',1);
vswc = fscanf(fid,'% e',1);
vcle = fscanf(fid,'% e',1);

```

```

vswe = fscanf(fid,'% e',1);
alpha = fscanf(fid, '% f',1);
end

fprintf('==== free piston stirling engine data summary =====\n');
fprintf(' compression clearence volume: vclc = % .1f (cc)\n', vclc*1e6);
fprintf(' compression swept volume: vswc = % .1f (cc)\n', vswc*1e6);
fprintf(' expansion clearence volume: vcle = % .1f (cc)\n', vcle*1e6);
fprintf(' expansion swept volume: vswe = % .1f (cc)\n', vswe*1e6);
fprintf(' expansion phase angle advance: alpha = % .1f (rad)\n \n',
alpha);
fprintf('*****\n \n');
end
end

```

```

function define

% define the stirling engine geometric and operational parameters

% The set of global variables defined are:

% engine % Free Piston Stirling Engine (FPSE)

global vclc % compression clearence vols [m^3]
global vcle % expansion clearence vols [m^3]
global vswc % compression swept volumes [m^3]
global vswe % expansion swept volumes [m^3]
global alpha % phase angle advance of expansion space [radians]
% global dpp % diameter of power pistons [m]
% global ddcomp % diameter of displacer at compression side [m]

```

```

% global ddexp % diameter of displacer at expansion side [m]
% global app % area of power pistons [m^2]
% global adcomp % area of displacer at compression side [m^2]
% global adexp % area of displacer at expansion side [m^2]
% global ypp % maximum power piston vertical displacement [m]
% global yd % maximum displacer vertical displacement [m]

% heatex/cooler
global dk % cooler hydraulic diameter [m]
global lk % cooler effective length [m]
global ak % cooler internal free flow area [m^2]
global vk % cooler void volume [m^3]
global awgk % cooler internal wetted area [m^2]

% heatex/heater
global dh % heater hydraulic diameter [m]
global lh % heater effective length [m]
global ah % heater internal free flow area [m^2]
global vh % heater void volume [m^3]
global awgh % heater internal wetted area [m^2]

```

```

% heatex/regenerator
global dr % regen hydraulic diameter [m]
global lr % regenerator effective length [m]
global ar % regen internal free flow area [m^2]
global vr % regen void volume [m^3]
global awgr % regen internal wetted area [m^2]
global cqwr % regenerator housing thermal conductance [W/K]

```

```
global awgr0 % no matrix regenerator wetted area [m^2]
```

```
% gas
```

```
global rgas % gas constant [J/kg.K]
```

```
global cp % specific heat capacity at constant pressure [J/kg.K]
```

```
global cv % specific heat capacity at constant volume [J/kg.K]
```

```
global gama % ratio: cp/cv
```

```
global mu0 % dynamic viscosity at reference temp t0 [kg.m/s]
```

```
global t0 % reference temp. [K]
```

```
global t_suth % Sutherland constant [K]
```

```
global prandtl % Prandtl number
```

```
% operat
```

```
global tk % cooler temperatures [K]
```

```
global tr % regenerator temperatures [K]
```

```
global th % heater temperatures [K]
```

```
global pmean % mean (charge) pressure [Pa]
```

```
global freq % cycle frequency [Hz]
```

```
global omega % cycle frequency [rads/s]
```

```
global mgas % total mass of gas in engine [kg]
```

```
% new data file
```

```
global new fid
```

```
new = input('Create a new data file? (y/n)','s');
```

```
if strcmp(new,'y',1)
```

```
filename = input('enter new filename: ','s');
```

```
fid = fopen(filename,'w');
```

```

else
fid = 0;
while fid < 1
filename = input('open filename: ','s');
[fid, message] = fopen(filename,'r');
if fid == -1
display(message)
display('press ^C to exit')
end
end
end
engine;
heatex;
gas;
operat;

fclose(fid);
end

```

```

function [var,dvar] = simple
% Simple Analysis- including heat transfer and pressure drop effects
% Returned values:
% var(23,37) array of variable values every 10 degrees (0 - 360)
% dvar(16,37) array of derivatives every 10 degrees (0 - 360)
% Row indices of the var, dvar arrays:
TC = 1; % Compression space temperature [K]

```

```

TE = 2; % Expansion space temperature [K]
QK = 3; % Heat transferred to the cooler [J]
QR = 4; % Heat transferred to the regenerator [J]
QH = 5; % Heat transferred to the heater [J]
WC = 6; % Work done by the compression space [J]
WE = 7; % Work done by the expansion space [J]
W = 8; % Total work done (WC + WE) [J]
P = 9; % Pressure [Pa]
VC = 10; % Compression space volume [m^3]
VE = 11; % Expansion space volume [m^3]
MC = 12; % Mass of gas in the compression space [kg]
MK = 13; % Mass of gas in the cooler [kg]
MR = 14; % Mass of gas in the regenerator [kg]
MH = 15; % Mass of gas in the heater [kg]
ME = 16; % Mass of gas in the expansion space [kg]
TCK = 17; % Conditional temperature compression space / cooler [K]
THE = 18; % Conditional temperature heater / expansion space [K]
GACK = 19; % Conditional mass flow compression space / cooler [kg/rad]
GAKR = 20; % Conditional mass flow cooler / regenerator [kg/rad]
GARH = 21; % Conditional mass flow regenerator / heater [kg/rad]
GAHE = 22; % Conditional mass flow heater / expansion space [kg/rad]
V = 23; % Total volume ([m^3]
% Size of var(ROWV,COL), dvar(ROWD,COL)
ROWV = 23; % number of rows in the var matrix
ROWD = 16; % number of rows in the dvar matrix
COL = 37; % number of columns in the matrices (every 10 degrees)

```

```

%*****

global tk    % cooler temperatures [K]
global tr    % regen temperatures [K]
global th    % heater temperatures [K]
global cqwr % regenerator housing thermal conductance [W/K]
global freq % cycle frequency [Hz]

%*****

twk = tk; % Cooler wall temp - equal to initial cooler gas temp
twh = th; % Heater wall temp - equal to initial heater gas temp
epsilon = 1; % allowable temperature error bound for cyclic convergence
t_error = 10*epsilon; % Initial temperature error (to enter loop)
while (t_error > epsilon)
[var,dvar] = adiab;
qrloss = regsimple(var);
tgh = hotsimple(var,twh); % new heater gas temperature
tgc = kolsimple(var,twk); % new cooler gas temperature
t_error = abs(th - tgh) + abs(tk - tgc);
th = tgh;
tk = tgc;
tr = (th-tk)/log(th/tk);
end

fprintf('==== converged heater and cooler mean temperatures =====\n');
fprintf('heater wall temperatures: Twh = % .1f (K)\n',twh);
fprintf('heater gas temperatures: Th = % .1f (K)\n',th);
fprintf('cooler wall temperatures: Twk = % .1f (K)\n',twk);
fprintf('cooler gas temperatures: Tk = % .1f (K)\n \n',tk);

```

```

fprintf('*****\n\n');

%===== Print out Ideal Adiabatic analysis results =====

Qkpower = var(QK,COL)*freq; % Heat transferred to the cooler (W)
Qrpower = var(QR,COL)*freq; % Heat transferred to the regenerator (W)
Qhpower = var(QH,COL)*freq; % Heat transferred to the heater (W)
Wpower  = var(W,COL)*freq; % Total power output (W)
eff      = Wpower/Qhpower; % engine thermal efficiency
% eff(i)   = var (W,i)/var(QH,i); % engine thermal efficiency
% end

% eff      = var (W,COL)/var(QH,COL); % engine thermal efficiency
save('Wpower.mat','Wpower');
save('Qhpower.mat','Qhpower');

% Efficiency;

% for i = 1:1:37

% eff(i)    = Wpower (i)/Qhpower(i); % engine thermal efficiency
% end

% for i = 1:1:37

% freq = 5:5:185;

% power(i) = var(W,37)*freq(i);

% heatadd(i) = var(QH,37)*freq(i);

% eff(i) = power / heatadd;

% end

fprintf('===== Ideal Adiabatic analysis results =====\n');
fprintf('Heat transferred to the cooler: Qkpower = % .3f (W)\n \n',
Qkpower);

fprintf('*****\n\n');

```

```

fprintf('Heat transferred to the regen.: Qrpower = % .3f (W)\n \n',
Qrpower);

fprintf('*****\n\n');

fprintf('Heat transferred to the heater: Qhpower = % .3f (W)\n \n',
Qhpower);

fprintf('*****\n\n');

fprintf(' Total power output: Wpower = % .3f (W)\n \n', Wpower);

fprintf('*****\n\n');

fprintf(' Ideal Adiabatic Thermal efficiency: eff = % .3f(%%)\n \n',
eff*100);

fprintf('*****\n\n');

% save('thermal_efficiency.mat','eff');

fprintf('===== REGENERATOR ANALYSIS RESUTS =====\n \n');

% qrloss = regquasi(var);

fprintf('**** Regenerator Net Enthalpy Losses (qrloss) ****\n');

fprintf('    qrloss = % .3f (W)\n \n', qrloss*freq);

fprintf('*****\n \n');

fprintf('    **** Regenerator Wall Heat Leakage (qwr1)    ****\n');

% for i = 1:1:37

% freq = 5:5:185;

qwr1 = cqwr*(twh - twk)/freq;

% end

fprintf('    qwr1 = % .3f (W)\n \n', qwr1*freq);

fprintf('*****\n \n');

fprintf('===== pressure drop Simple analysis =====\n');

dwork = worksimple(var,dvar);

```

```

%save('dwork.mat','dwork');

% for i = 1:1:37

% freq = 5:5:185;

% Dwork = dwork*freq;

% end

fprintf(' *** Pressure Drop Available Work Loss (dwork) *****\n \n');

fprintf('          dwork = % .4f (W)\n \n',dwork*freq);

fprintf('*****\n \n');

% for i =1:1:37

% for j =1:1:37

% freq =5:5:37;

% Outputpower = Wpower - dwork*freq;

Outputpower = Wpower - dwork*freq;

Heatpowerin = Qhpower + qrloss*freq + qwr1*freq;

% acteff = actWpower (i)/actQhpower(i);

% end

% end

Effeciency = Outputpower/Heatpowerin;

save('Effeciency.mat', 'Effeciency');

save('Heatpowerin.mat','Heatpowerin');

save('Outputpower.mat','Outputpower');

fprintf('= Output Work, Heat and Effeciency for Simple Analysis ==\n \n');

fprintf('      ***** Total Power Output From Simple Analysis *****\n');

fprintf('          Outputpower = % .3f (W)\n \n', Outputpower);

fprintf('*****\n\n');

fprintf('** Heat power Added to the Heater From Simple Analysis **\n');

```

```

fprintf('    Heatpowerin  = % .3f (W)\n \n', Heatpowerin );

fprintf('*****\n \n');

fprintf('***** Effecieny From Simple Analysis  *****\n');

fprintf(' Efficiency = % .3f (%%)\n \n', Efficiency*100);

fprintf('*****\n \n');

% Temperature plotting of the Simple simulation results

figure

x = 0:10:360;

Tcomp = var(TC,:);

Texp = var(TE,:);

plot(x,Tcomp,'b-*',x,Texp,'r-*','LineWidth',1);

xlim([0 360]);

hold on

x = [0,360];

y = [twk,twk];

ylim([200 1100]);

plot(x,y,'b--','LineWidth',1)

y = [tk,tk];

plot(x,y,'b','LineWidth',1)

y = [tr,tr];

plot(x,y,'g','LineWidth',1)

y = [th,th];

plot(x,y,'r','LineWidth',1)

y = [twh,twh];

plot(x,y,'r--','LineWidth',1)

hleg = legend('Tcomp','Texp','twk','tk','tr','th','twh');

```

```

hold off

grid on

xlabel('Crank Angle (theta) [degree]');
ylabel('Temperature [K]');

%title('Simple analysis simulation - Wall and Gas Temps vs Crank Angle');

% various plots of the quasi steady flow analysis

plotsimple(var,dvar);

end

function [var,dvar] = adiab

% ideal adiabatic model simulation

% Returned values:

% var(23,37) array of variable values every 10 degrees (0 - 360)
% dvar(16,37) array of derivatives every 10 degrees (0 - 360)

% Row indices of the var, dvar matrices, and the y,dy variable vectors:

TC = 1; % Compression space temperature (K)
TE = 2; % Expansion space temperature (K)
QK = 3; % Heat transferred to the cooler (J)
QR = 4; % Heat transferred to the regenerator (J)
QH = 5; % Heat transferred to the heater (J)
WC = 6; % Work done by the compression space (J)
WE = 7; % Work done by the expansion space (J)
W = 8; % Total work done (WC + WE) (J)
P = 9; % Pressure (Pa)
VC = 10; % Compression space volume (m^3)
VE = 11; % Expansion space volume (m^3)
MC = 12; % Mass of gas in the compression space (kg)

```

```

MK = 13; % Mass of gas in the cooler (kg)
MR = 14; % Mass of gas in the regenerator (kg)
MH = 15; % Mass of gas in the heater (kg)
ME = 16; % Mass of gas in the expansion space (kg)
TCK = 17; % Conditional temperature compression space / cooler (K)
THE = 18; % Conditional temperature heater / expansion space (K)
GACK = 19; % Conditional mass flow compression space / cooler (kg/rad)
GAKR = 20; % Conditional mass flow cooler / regenerator (kg/rad)
GARH = 21; % Conditional mass flow regenerator / heater (kg/rad)
GAHE = 22; % Conditional mass flow heater / expansion space (kg/rad)
V = 23; % Total volume (m^3)
% Size of var(ROWV,COL), y(ROWV), dvar(ROWD,COL), dy(ROWD)
ROWV = 23; % number of rows in the var matrix
ROWD = 16; % number of rows in the dvar matrix
COL = 37; % number of columns in the matrices (every 10 degrees)
global tk % cooler temperatures [K]
global th % heater temperatures [K]
%*****

fprintf('===== The Ideal Analysis =====\n')
fprintf('Cooler temperature: Tk = % .1f (K)\n', tk);
fprintf('Heater temperature: Th = % .1f (K)\n \n', th);
fprintf('*****\n\n');

epsilon = 0.1; % Allowable error in temperature (K)
max_iteration = 20; % Maximum number of iterations to convergence
ninc = 360; % number of integration increments (every degree)
step = ninc/36; % for saving values in var, dvar matrices

```

```

dtheta = 2.0*pi/ninc; % integration increment (radians)

% Initial conditions:
y(TH) = th;
y(TK) = tk;
y(TE) = th;
y(TC) = tk;
iter = 0;
t_error = 10*epsilon; % Initial error to enter the loop

% Iteration loop to cyclic convergence
while ((t_error >= epsilon)&(iter < max_iteration))

% cyclic initial conditions
tc0 = y(TC);
te0 = y(TE);
theta = 0;
y(QK) = 0;
y(QR) = 0;
y(QH) = 0;
y(WC) = 0;
y(WE) = 0;
y(W) = 0;

fprintf('iteration % d:   y(TC) = % .1f(K)   &   y(TE) = % .1f(K)\n
\n',iter,y(TC),y(TE));

for i = 1:1:ninc

[theta,y,dy] = rk4('dadiab',7,theta,dtheta,y);

end

t_error = abs(tc0 - y(TC)) + abs(te0 - y(TE));

```

```

iter = iter + 1;

end

if (iter >= max_iteration)
fprintf('No convergence within % d iteration \n',max_iteration);
end

% Initial var and dvar matrix
var = zeros(23,37);
dvar = zeros(16,37);

% a final cycle, to fill the var, dvar matrices
theta = 0;
y(QK) = 0;
y(QR) = 0;
y(QH) = 0;
y(WC) = 0;
y(WE) = 0;
y(W) = 0;

[var,dvar] = Filmatrix(1,y,dy,var,dvar);
for i = 2:1:COL
for j = 1:1:step
[theta,y,dy] = rk4('dadiab',7,theta,dtheta,y);
end
[var,dvar] = Filmatrix(i,y,dy,var,dvar);
end
end

function [y,dy] = dadiab(theta,y)

% Evaluate ideal adiabatic model derivatives

```

```

% Arguments: theta - current cycle angle [radians]

% y(23) - vector of current variable values

% Returned values:

% y(22) - updated vector of current variables

% dy(16) vector of current derivatives

% Function invoked : volume.m

% Indices of the y, dy vectors:

TC = 1; % Compression space temperature [K]
TE = 2; % Expansion space temperature [K]
QK = 3; % Heat transferred to the cooler [J]
QR = 4; % Heat transferred to the regenerator [J]
QH = 5; % Heat transferred to the heater [J]
WC = 6; % Work done by the compression space [J]
WE = 7; % Work done by the expansion space [J]
W = 8; % Total work done (WC + WE) [J]
P = 9; % Pressure [Pa]
VC = 10; % Compression space volume [m^3]
VE = 11; % Expansion space volume [m^3]
MC = 12; % Mass of gas in the compression space [kg]
MK = 13; % Mass of gas in the cooler [kg]
MR = 14; % Mass of gas in the regenerator [kg]
MH = 15; % Mass of gas in the heater [kg]
ME = 16; % Mass of gas in the expansion space [kg]
TCK = 17; % Conditional temperature compression space / cooler [K]
THE = 18; % Conditional temperature heater / expansion space [K]
GACK = 19; % Conditional mass flow compression space / cooler [kg/rad]

```

```

GAKR = 20; % Conditional mass flow cooler / regenerator [kg/rad]
GARH = 21; % Conditional mass flow regenerator / heater [kg/rad]
GAHE = 22; % Conditional mass flow heater / expansion space [kg/rad]
V = 23; % Total volume [m^3]

%=====

% global variables used from "define" functions

global vk % cooler void volume [m^3]
global vr % regen void volume [m^3]
global vh % heater void volume [m^3]
global tk % cooler temperatures [K]
global tr % regen temperatures [K]
global th % heater temperatures [K]
global cp % specific heat capacity at constant pressure [J/kg.K]
global cv % specific heat capacity at constant volume [J/kg.K]
global gama % ratio: cp/cv
global mgas % total mass of gas in engine [kg]
global rgas % gas constant [J/kg.K]

%=====

% Volume and volume derivatives:
[y(VC),y(VE),dy(VC),dy(VE),y(V)] = volume(theta);

% Pressure and pressure derivatives:
vot = vk/tk + vr/tr + vh/th;
y(P) = (mgas*rgas/(y (VC)/y(TC) + vot + y (VE)/y(TE)));
top = -y(P)*(dy (VC)/y(TCK) + dy (VE)/y(TE));
bottom = (y (VC)/(y(TCK)*gama) + vot + y (VE)/(y(TE)*gama));
dy(P) = top/bottom;

```

```

% Mass accumulations and derivatives:

y(MC) = y(P)*y (VC)/(rgas*y(TC));
y(MK) = y(P)*vk/(rgas*tk);
y(MR) = y(P)*vr/(rgas*tr);
y(MH) = y(P)*vh/(rgas*th);
y(ME) = y(P)*y (VE)/(rgas*y(TE));

dy(MC) = (y(P)*dy(VC) + y(VC)*dy (P)/gama)/(rgas*y(TCK));
dy(ME) = (y(P)*dy(VE) + y(VE)*dy (P)/gama)/(rgas*y(TH));

dpop  = dy (P)/y(P);

dy(MK) = y(MK)*dpop;
dy(MR) = y(MR)*dpop;
dy(MH) = y(MH)*dpop;

% Mass flow between cells:

y(GACK) = -dy(MC);
y(GAKR) = y(GACK) - dy(MK);
y(GAHE) = dy(ME);
y(GARH) = y(GAHE) + dy(MH);

% Conditional temperatures between cells:

y(TCK) = tk;
if(y(GACK)>0)
y(TCK) = y(TC);
end
y(TH) = y(TE);
if(y(GAHE)>0)
y(TH) = th;
end

```

```

% 7 derivatives to be integrated by rk4:

% Working space temperatures:

dy(TC) = y(TC)*(dpop + dy (VC)/y(VC) - dy (MC)/y(MC));
dy(TE) = y(TE)*(dpop + dy (VE)/y(VE) - dy (ME)/y(ME));

% save('TC.mat','dy (TC)');
% save('TE.mat','TE');

% Energy:

dy(QK) = vk*dy(P)*cv/rgas - cp*(y(TCK)*y(GACK) - tk*y(GAKR));
dy(QR) = vr*dy(P)*cv/rgas - cp*(tk*y(GAKR) - th*y(GARH));
dy(QH) = vh*dy(P)*cv/rgas - cp*(th*y(GARH) - y(TH)*y(GAHE));
dy(WC) = y(P)*dy(VC);
dy(WE) = y(P)*dy(VE);

% dy(Q) = dy(QK) + dy(QR) + dy(QH);

% Net work done:

dy(W) = dy(WC) + dy(WE);
y(W) = y(WC) + y(WE);

end

function heatex

% Specify heat exchanger geometric parameters

cooler;

heater;

regen;

%*****

function cooler

% Specify cooler geometric parameters

global vk % cooler void volume [m^3]

```

```

global ak % cooler internal free flow area [m^2]
global awgk % cooler internal wetted area [m^2]
global dk % cooler hydraulic diameter [m]
global lk % cooler effective length [m]
global new fid % new data file

fprintf('==== Available cooler type is smooth annulus =====\n \n');
[vk,ak,awgk,dk,lk] = annulus;

fprintf('===== cooler data summary =====\n');
fprintf(' cooler void volume: vk = % .2f (cc)\n', vk*1e6);
fprintf(' cooler free flow area: ak = % .2f (cm^2)\n', ak*1e4);
fprintf(' cooler wetted area: awgk = % .2f (cm^2)\n', awgk*1e4);
fprintf(' cooler hydraulic diameter: dk = % .2f (mm) \n', dk*1e3);
fprintf(' cooler length: lk = % .2f (cm)\n \n', lk*1e2);
fprintf('*****\n \n');

end

%*****

function heater

% Specify heater geometric parameters

global vh % heater void volume [m^3]
global ah % heater internal free flow area [m^2]
global awgh % heater internal wetted area [m^2]
global dh % heater hydraulic diameter [m]
global lh % heater effective length [m]
global new fid % new data file

fprintf('==== Available heater type is smooth annulus =====\n \n');
[vh,ah,awgh,dh,lh] = annulus;

```

```

fprintf('===== heater data summary =====\n');
fprintf(' heater void volume: vh = % .2f (cc)\n', vh*1e6);
fprintf(' heater free flow area: ah = % .2f (cm^2)\n', ah*1e4);
fprintf(' heater wetted area: awgh = % .2f (cm^2)\n', awgh*1e4);
fprintf(' heater hydraulic diameter: dh = % .2f (mm)\n', dh*1e3);
fprintf(' heater length:lh = % .2f (cm)\n \n', lh*1e2);
fprintf('*****\n \n');
end

%*****

function [v,a,awg,d,len] = annulus

% annular gap heat exchanger

global new fid % new data file

% fprintf(' annular gap heat exchanger \n');

if(strncmp(new,'y',1))

dout = input('enter annular gap outer diameter (dout) [m] : ');
din  = input('enter annular gap inner diameter (din) [m] : ');
len  = input('enter heat exchanger length (len) [m] : ');

fprintf(fid, '% .3e \n', dout);
fprintf(fid, '% .3e \n', din);
fprintf(fid, '% .3e \n', len);

else

dout = fscanf(fid,'% e',1);
din  = fscanf(fid,'% e',1);
len  = fscanf(fid,'% e',1);

end

a = pi*(dout*dout - din*din)/4;

```

```

v = a*len;

awg = pi*dout*len;

d = dout - din;

end

%*****

% function [v,a,awg,d,len] = pipes

% % homogeneous smooth pipes heat exchanger

% global new fid % new data file

% fprintf('INTERCAMBIADOR DE TUBOS \n')

% if(strncmp(new,'y',1))

% d = input('ingrese el diametro del tubo [m] : ');

% l = input('ingrese el diametro promedio del intercambiador [m] : ');

% num = input('ingrese el numero de vueltas : ');

% fprintf(fid, '% .3e \n', d);

% fprintf(fid, '% .3e \n', l);

% fprintf(fid, '% d \n', num);

% else

% d = fscanf(fid,'% e',1);

% l = fscanf(fid,'% e',1);

% num = fscanf(fid,'% d',1);

% end

% dout = 0.65 + d/2;

% din = 0.65 - d/2;

% len=pi*l;

% a = pi*(0.63*0.63 - 0.6*0.6)/4 - pi*(dout*dout - din*din)/4; % area de
paso es el area total menos la

```

```

% cantidad de tubos en la primera fila, de 3.

% v = num*pi*d*d/4*len;

% awg = num*pi*d*len;

% end

%*****

% function [v,a,awg,d,len] = slots

% % slots heat exchanger

% global new fid % new data file

% fprintf(' slots heat exchanger \n')

% if(strncmp(new,'y',1))

% w = input('enter width of slot [m] : ');

% h = input('enter height of slot [m] : ');

% len = input('enter heat exchanger length [m] : ');

% num = input('enter number of slots : ');

% fprintf(fid, '% .3e \n', w);

% fprintf(fid, '% .3e \n', h);

% fprintf(fid, '% .3e \n', len);

% fprintf(fid, '% d \n', num);

% else

% w = fscanf(fid,'% e',1);

% h = fscanf(fid,'% e',1);

% len = fscanf(fid,'% e',1);

% num = fscanf(fid,'% d',1);

% end

% a = num*w*h;

% v = a*len;

```

```

% awg = num*2*(w + h)*len;

% d = 4*v/awg;

% end

end

function tgc = kolsimple(var,twk)

% evaluate cooler average heat transfer performance

% Arguments:

% var(23,37) array of variable values every 10 degrees (0 - 360)

% twk - cooler wall temperature [K]

% Returned values:

% tgc - cooler average gas temperature [K]

% Row indices of the var array:

TC = 1; % Compression space temperature [K]

TE = 2; % Expansion space temperature [K]

QK = 3; % Heat transferred to the cooler [J]

QR = 4; % Heat transferred to the regenerator [J]

QH = 5; % Heat transferred to the heater [J]

WC = 6; % Work done by the compression space [J]

WE = 7; % Work done by the expansion space [J]

W = 8; % Total work done (WC + WE) [J]

P = 9; % Pressure [Pa]

VC = 10; % Compression space volume [m^3]

VE = 11; % Expansion space volume [m^3]

MC = 12; % Mass of gas in the compression space [kg]

MK = 13; % Mass of gas in the cooler [kg]

MR = 14; % Mass of gas in the regenerator [kg]

```

```

MH = 15; % Mass of gas in the heater [kg]
ME = 16; % Mass of gas in the expansion space [kg]
TCK = 17; % Conditional temperature compression space / cooler [K]
THE = 18; % Conditional temperature heater / expansion space [K]
GACK = 19; % Conditional mass flow compression space / cooler [kg/rad]
GAKR = 20; % Conditional mass flow cooler / regenerator [kg/rad]
GARH = 21; % Conditional mass flow regenerator / heater [kg/rad]
GAHE = 22; % Conditional mass flow heater / expansion space [kg/rad]
V = 23; % Total volume ([m^3]

*****

global tk % cooler temperature [K]
global freq % cycle frequency [Hz]
global omega % cycle frequency [rads/s]
global ak % cooler internal free flow area [m^2]
global awgk % cooler internal wetted area [m^2]
global dk % cooler hydraulic diameter [m]
global qrloss % Regenerator net enthalpy loss
% Calculating the Reynolds number over the cycle
for i = 1:1:37
% freq = 5:5:185;
% omega = 2*pi*freq(i);
gak(i) = (var(GACK,i) + var(GAKR,i))*omega/2;
gk = gak (i)/ak;
[mu,kgas,re(i)] = reynum(tk,gk,dk);
end
% Average and maximum Reynolds number

```

```

sumre=0;
remax=re(1);
for i = 1:1:36
sumre=sumre + re(i);
if(re(i) > remax)
remax = re(i);
end
end
reavg = sumre/36;
[ht,fr] = pipefr(dk,mu,reavg); % Heat transfer coefficient
tgk = twk - (var(QK,37))*freq/(ht*awgk); % Cooler gas temperature [K]
qk = var(QK,37);
% tgk = twk - (var(QK,37)-qrloss)*freq/(ht*awgk); % Cooler gas temperature
[K]
fprintf('===== Cooler Simple analysis =====\n')
fprintf(' Average Reynolds number: reavg = % .1f \n',reavg)
fprintf(' Maximum Reynolds number: remax = % .1f \n',remax)
fprintf(' Heat transfer coefficient: ht = % .2f (W/m^2*K)\n',ht)
fprintf('Heat transferred from the cooler: qk = % .2f (J)\n',qk);
fprintf('cooler wall temperatures: Twk = % .1f (K)\n',twk);
fprintf('cooler gas temperatures: Tk = % .1f (K)\n \n',tgk);
fprintf('*****\n \n');
end
function tgh = hotsimple(var,twh)
% evaluate heater average heat transfer performance
% Arguments:

```

```

% var(23,37) array of variable values every 10 degrees (0 - 360)

% twh - heater wall temperature [K]

% Returned values:

% tgh - heater average gas temperature [K]

% Row indices of the var array:

TC = 1; % Compression space temperature [K]

TE = 2; % Expansion space temperature [K]

QK = 3; % Heat transferred to the cooler [J]

QR = 4; % Heat transferred to the regenerator [J]

QH = 5; % Heat transferred to the heater [J]

WC = 6; % Work done by the compression space [J]

WE = 7; % Work done by the expansion space [J]

W = 8; % Total work done (WC + WE) [J]

P = 9; % Pressure [Pa]

VC = 10; % Compression space volume [m^3]

VE = 11; % Expansion space volume [m^3]

MC = 12; % Mass of gas in the compression space [kg]

MK = 13; % Mass of gas in the cooler [kg]

MR = 14; % Mass of gas in the regenerator [kg]

MH = 15; % Mass of gas in the heater [kg]

ME = 16; % Mass of gas in the expansion space [kg]

TCK = 17; % Conditional temperature compression space / cooler [K]

THE = 18; % Conditional temperature heater / expansion space [K]

GACK = 19; % Conditional mass flow compression space / cooler [kg/rad]

GAKR = 20; % Conditional mass flow cooler / regenerator [kg/rad]

GARH = 21; % Conditional mass flow regenerator / heater [kg/rad]

```

```

GAHE = 22; % Conditional mass flow heater / expansion space [kg/rad]

V = 23; % Total volume [m^3]

%*****

global th    % heater temperature [K]
global dh    % heater hydraulic diameter [m]
global ah    % heater internal free flow area [m^2]
global awgh  % heater internal wetted area [m^2]
global freq  % cycle frequency [Hz]
global omega % cycle frequency [rads/s]
global qrloss % Regenerator net enthalpy loss

% Calculating the Reynolds number over the cycle
for i = 1:1:37
%   freq = 5:5:185;
%   omega = 2*pi*freq(i);
gah(i) = (var(GARH,i) + var(GAHE,i))*omega/2;
gh = gah (i)/ah;
[mu,kgas,re(i)] = reynum(th,gh,dh);
end

% Average and maximum Reynolds number
sumre=0;
remax=re(1);
for i = 1:1:36
sumre=sumre + re(i);
if(re(i) > remax)
remax = re(i);
end

```

```

end

reavg = sumre/36;

[ht,fr] = pipefr(dh,mu,reavg); % Heat transfer coefficient

tgh = twh - (var(QH,37))*freq/(ht*awgh); % Heater gas temperature [K]

qh = var(QH,37);

fprintf('===== Heater Simple analysis =====\n');

fprintf(' Average Reynolds number: reavg = % .1f \n',reavg);

fprintf(' Maximum Reynolds number: remax = % .1f \n',remax);

fprintf(' Heat transfer coefficient: ht = % .2f (W/m^2*K)\n',ht);

fprintf('Heat transferred to the heater: qh = % .2f (J)\n',qh);

fprintf('heater wall temperatures: Twh = % .1f (K)\n',twh);

fprintf('heater gas temperatures: Tgh = % .1f (K)\n \n',tgh);

fprintf('*****\n \n');

end

function [ht,fr] = pipefr(d,mu,re)

% evaluate heat transfer coefficient, Reynolds friction factor

% Arguments:

% d - hydraulic diameter [m]

% mu - gas dynamic viscosity [kg.m/s]

% re - Reynolds number

% Returned values:

% ht - heat transfer coefficient [W/m^2.K]

% fr - Reynolds friction factor ( = re*fanning friction factor)

global cp % specific heat capacity at constant pressure [J/kg.K]

global prandtl % Prandtl number

fr = 0.0791*re^0.75;

```

```

% From Reynolds simple analogy:

ht = fr*mu*cp/(2*d*prandtl);

% k = 389; % conductividad termica del cobre [W/mK]

% Nu =

0.3+0.62*re^0.5*prandtl^(1/3)/(1+(0.4/prandtl)^(2/3))^0.25*(1+(re/282)^(5/
8))^0.25);

% ht=N

end

function qrloss = regsimple(var)

% Evaluate the effectiveness and performance of the regenerator

% Arguments:

% var(22,37) array of variable values every 10 degrees (0 - 360)

% Returned value:

% qrloss - regenerator net enthalpy loss [J]

% Row indices of the var array:

TC = 1; % Compression space temperature [K]

TE = 2; % Expansion space temperature [K]

QK = 3; % Heat transferred to the cooler [J]

QR = 4; % Heat transferred to the regenerator [J]

QH = 5; % Heat transferred to the heater [J]

WC = 6; % Work done by the compression space [J]

WE = 7; % Work done by the expansion space [J]

W = 8; % Total work done (WC + WE) [J]

P = 9; % Pressure [Pa]

VC = 10; % Compression space volume [m^3]

VE = 11; % Expansion space volume [m^3]

```

```

MC = 12; % Mass of gas in the compression space [kg]
MK = 13; % Mass of gas in the cooler [kg]
MR = 14; % Mass of gas in the regenerator [kg]
MH = 15; % Mass of gas in the heater [kg]
ME = 16; % Mass of gas in the expansion space [kg]
TCK = 17; % Conditional temperature compression space / cooler [K]
THE = 18; % Conditional temperature heater / expansion space [K]
GACK = 19; % Conditional mass flow compression space / cooler [kg/rad]
GAKR = 20; % Conditional mass flow cooler / regenerator [kg/rad]
GARH = 21; % Conditional mass flow regenerator / heater [kg/rad]
GAHE = 22; % Conditional mass flow heater / expansion space [kg/rad]
V = 23; % Total volume ([m^3]

%*****

% global matrix_type % m)esh or f)oil %% No marrix
% global tr1    % regen1 temperature [K]
% global tr2    % regen2 temperature [K]
global tr    % regen temperature [K]
global dr    % regen hydraulic diameter [m]
global ar    % regen internal free flow area [m^2]
global awgr % regen internal wetted area [m^2]
global freq % cycle frequency [Hz]
global omega % cycle frequency [rads/s]
% Reynolds number over the cycle
for i = 1:1:37
gar(i) = (var(GAKR,i) + var(GARH,i))*omega/2;
gr = gar (i)/ar;

```

```

[mu,kgas,re(i)] = reynum(tr,gr,dr);
end
% average and maximum Reynolds number
sumre = 0;
rema = re(1);
for i = 1:1:36
    sumre = sumre + re(i);
    if(re(i) > rema)
        rema = re(i);
    end
end
reavg = sumre/36;
[st,fr] = matrixfr(reavg); %%% No matrix
ntu = st*awgr/(2*ar);
effect = ntu/(ntu + 1);
% Calculate qrloss
for i = 1:1:37
    qreg(i) = var(QR,i);
end
qrmin = min(qreg);
qrmax = max(qreg);
qrloss = (1 - effect)*(qrmax - qrmin);
% Regenerator Simple Analysis Results:
fprintf('===== Regenerator simple analysis =====\n');
fprintf('Average Reynolds number: reavg = % .1f \n', reavg);
fprintf('Maximum Reynolds number: rema = % .1f \n', rema);

```

```

fprintf('Stanton number(Average Re): st = % .3f \n',st);
fprintf('Number of transfer units: ntu = % .1f \n',ntu);
fprintf('Regenerator effectiveness = % .3f % \n \n',effect*100);
fprintf('*****\n \n');
end

function dwork = worksimple(var,dvar)

% Evaluate the pressure drop available work loss [J]

% Arguments:
% var(23,37) array of variable values every 10 degrees (0 - 360)
% dvar(16,37) array of derivatives every 10 degrees (0 - 360)

% Returned value:
% dwork - pressure drop available work loss [J]

% Row indices of the var, dvar arrays:
TC = 1; % Compression space temperature [K]
TE = 2; % Expansion space temperature [K]
QK = 3; % Heat transferred to the cooler [J]
QR = 4; % Heat transferred to the regenerator [J]
QH = 5; % Heat transferred to the heater [J]
WC = 6; % Work done by the compression space [J]
WE = 7; % Work done by the expansion space [J]
W = 8; % Total work done (WC + WE) [J]
P = 9; % Pressure [Pa]
VC = 10; % Compression space volume [m^3]
VE = 11; % Expansion space volume [m^3]
MC = 12; % Mass of gas in the compression space [kg]
MK = 13; % Mass of gas in the cooler [kg]

```

```

MR = 14; % Mass of gas in the regenerator [kg]
MH = 15; % Mass of gas in the heater [kg]
ME = 16; % Mass of gas in the expansion space [kg]
TCK = 17; % Conditional temperature compression space / cooler [K]
THE = 18; % Conditional temperature heater / expansion space [K]
GACK = 19; % Conditional mass flow compression space / cooler [kg/rad]
GAKR = 20; % Conditional mass flow cooler / regenerator [kg/rad]
GARH = 21; % Conditional mass flow regenerator / heater [kg/rad]
GAHE = 22; % Conditional mass flow heater / expansion space [kg/rad]
V = 23; % Total volume ([m^3]
% Size of var(ROWV,COL), dvar(ROWD,COL)
ROWV = 23; % number of rows in the var matrix
ROWD = 16; % number of rows in the dvar matrix
COL = 37; % number of columns in the matrices (every 10 degrees)
%*****
global tk    % cooler temperatures [K]
global tr    % regenerator temperatures [K]
global th    % heater temperatures [K]
global omega % cycle frequency [rads/s]
global vh    % heater void volume [m^3]
global ah    % heater internal free flow area [m^2]
global dh    % heater hydraulic diameter [m]
global lh    % heater effective length [m]
global vk    % cooler void volume [m^3]
global ak    % cooler internal free flow area [m^2]
global dk    % cooler hydraulic diameter [m]

```

```

global lk    % cooler effective length [m]
global vr    % regen void volume [m^3]
global ar    % regen internal free flow area [m^2]
global lr    % regenerator effective length [m]
global dr    % regen hydraulic diameter [m]
global qrloss % Regenerator net enthalpy loss
global kgas
dtheta = 2*pi/36;
dwork = 0; % initialise pumping work loss
for i = 1:1:37
% dwork = 0; % initialise pumping work loss
gk = (var(GACK,i) + var(GAKR,i))*omega/(2*ak);
% tk
% dk
% gk
[mu,kgas,re(i)] = reynum(tk,gk,dk);
[ht,fr] = pipefr(dk,mu,re(i));
dpkol(i) = 2*fr*mu*vk*gk*lk/(var(MK,i)*dk^2);
gr = (var(GAKR,i) + var(GARH,i))*omega/(2*ar);
[mu,kgas,re(i)] = reynum(tr,gr,dr);
[st,fr] = matrixfr(re(i));
dpreg(i) = 2*fr*mu*vr*gr*lr/(var(MR,i)*dr^2);
gh = (var(GARH,i) + var(GAHE,i))*omega/(2*ah);
[mu,kgas,re(i)] = reynum(th,gh,dh);
[ht,fr] = pipefr(dh,mu,re(i));
dphot(i) = 2*fr*mu*vh*gh*lh./(var(MH,i)*dh^2);

```

```

% dphot(i) = 18.03*gh*v*3*re (i)^(-0.316)*(Pt/dh)^(-0.927)*(Pt/Pd)^0.515;
dp(i) = dpkol(i) + dpreg(i) + dphot(i);
dwork = dwork + dtheta*dp(i)*dvar(VE,i); % pumping work [J]
pcom(i) = var(P,i);
pexp(i) = pcom(i) + dp(i);
pcombar = pcom*1e-5;
pexpbar = pexp*1e-5;
save('com_space _pressure.mat','pcombar');
save('exp_space _pressure.mat','pexpbar');
% end
end
dpkol(COL) = dpkol(1);
dpreg(COL) = dpreg(1);
dphot(COL) = dphot(1);
dp(COL)    = dp(1);
pcom(COL)  = pcom(1);
pexp(COL)  = pexp(1);
save('pressure_drop _cooler.mat', 'dpkol');
save('pressure_drop _regen.mat', 'dpreg');
save('pressure_drop _heater.mat', 'dphot');
save('pressure_drop _drop.mat', 'dp');
save('pressure_drop _comp.mat', 'pcom');
save('pressure_drop _exp.mat', 'pexp');
fprintf('quitting pressure plots...\n \n');
fprintf('*****\n \n');
end

```

```

function [x, y, dy] = rk4(deriv,n,x,dx,y)

% Classical fourth order Runge-Kutta method

% Integrates n first order differential equations

% dy(x,y) over interval x to x+dx

x0 = x;

y0 = y;

[y,dy1] = feval(deriv,x0,y);

for i = 1:n

y(i) = y0(i) + 0.5*dx*dy1(i);

end

xm = x0 + 0.5*dx;

[y,dy2] = feval(deriv,xm,y);

for i = 1:n

y(i) = y0(i) + 0.5*dx*dy2(i);

end

[y,dy3] = feval(deriv,xm,y);

for i = 1:n

y(i) = y0(i) + dx*dy3(i);

end

x = x0 + dx;

[y,dy] = feval(deriv,x,y);

for i = 1:n

dy(i) = (dy1(i) + 2*(dy2(i) + dy3(i)) + dy(i))/6;

y(i) = y0(i) + dx*dy(i);

end

end

```

```

function operat
% Determine operating parameters

global pmean % mean (charge) pressure [Pa]

global tk tr th % cooler, regenerator, heater temperatures [K]

global freq omega % cycle frequency [herz], [rads/s]

global new fid % new data file

if(strncmp(new,'y',1))

pmean = input('enter mean pressure (Pa) : ');

tk    = input('enter cold sink temperature (tk) in [K] : ');

th    = input('enter hot source temperature (th) in [K] : ');

freq  = input('enter operating frequency in (freq) in [Hz] : ');

fprintf(fid, '% .1f \n', pmean);

fprintf(fid, '% .1f \n', tk);

fprintf(fid, '% .1f \n', th);

fprintf(fid, '% .1f \n', freq);

else

pmean = fscanf(fid,'% f',1);

tk    = fscanf(fid,'% f',1);

th    = fscanf(fid,'% f',1);

freq  = fscanf(fid,'% f',1);

end

tr = (th - tk)/log(th/tk);

% for i = 1:1:37

%

% freq = 5:5:185;

omega = 2*pi*freq;

```

```

% end

% save('freq.mat','freq');

% save('omega.mat','omega');

fprintf('*****\n \n');

fprintf('===== operating parameters ===== \n');

fprintf(' mean pressure: pmaen = % .3f (kPa) \n',pmean*1e-3);

fprintf(' cold sink temperature: tk = % .1f (K) \n',tk);

fprintf(' hot source temperature: th = % .1f (K) \n',th);

fprintf(' effective regenerator temperature: tr = % .1f (K) \n',tr);

fprintf(' operating frequency: fr = % .1f (Hz)\n \n',freq);

fprintf('*****\n \n');

Schmidt;

end

%*****

function Schmidt

% Schmidt anlysis

global vclc % compression clearence volume [m^3]

global vcle % expansion clearence volume [m^3]

global vswc % compression swept volume [m^3]

global vswe % expansion swept volume [m^3]

global vk    % cooler volumes [m^3]

global vr    % regenerator volumes [m^3]

global vh    % heater volumes [m^3]

global tk    % cooler temperatures [K]

global tr    % regen temperatures [K]

global th    % heater temperatures [K]

```

```

global pmean % mean (charge) pressure [Pa]
global mgas % total mass of gas in engine [kg]
global rgas % gas constant [J/kg.K]
global freq % cycle frequency [Hz]
global omega % cycle frequency [rads/s]
global alpha % phase angle advance of expansion space [radians]
global vtot % total Schmidt volume
global p % pressure

%*****

% Schmidt analysis

c = (((vswe/th)^2 + (vswc/tk)^2 + *(vswe/th)*(vswc/tk)*cos(alpha))^0.5)/2;
s = (vswc/2 + vclc + vk)/tk + vr/tr + (vswe/2 + vcle + vh)/th;
b = c/s;
sqrtb = (1 - b^2)^0.5;
bf = (1 - 1/sqrtb);
beta = atan(vswe*sin (alpha)/th/(vswe*cos (alpha)/th + vswc/tk));
fprintf(' pressure phase angle: (beta) = % .1f(degrees)\n
\n',beta*180/pi);

% total mass of working gas in engine
mgas = pmean*s*sqrtb/rgas;
fprintf(' total mass of gas: (mgas) = % .3f(gm)\n \n',mgas*1e3);

% work output
wc = (pi*vswc*mgas*rgas*sin(beta)*bf/c);
we = (pi*vswe*mgas*rgas*sin(beta - alpha)*bf/c);
w = (wc + we);

% for i = 1:1:37

```

```

%      freq = 5:5:185;

power = w*freq;

eff = power /we; % qe = we

% end

save('effecincy.mat','eff');

% power = w*freq;

% eff = w/we; % qe = we

% figure;

% plot(freq,eff,'k-','LineWidth',2);

% grid on

% xlabel('frequency [Hz]');

% ylabel('Effecincy [%]');

% Printout Schmidt analysis results

fprintf('*****\n \n');

fprintf('===== Schmidt analysis =====\n')

fprintf(' Work: w = % .3e (joules)\n', w);

fprintf(' Power: power = % .3e (Watts)\n',power);

fprintf(' Heat removed from compression space: qc = wc = % .3e
(joules)\n',wc);

fprintf(' Heat added to the expansion space: qe = we = % .3e (joules) \n',
we);

fprintf(' Schmidt efficiency: eff = % .2f \n \n', eff);

fprintf('*****\n\n')

% Plot Schmidt analysis pv and p-theta diagrams

theta = 0:5:360;

vc = vclc + 0.5*vswc*(1 + cos(theta*pi/180));

```

```

ve = vcle + 0.5*vswe*(1 + cos(theta*pi/180 + alpha));

p = mgas*rgas./(vc/tk + vk/tk + vr/tr + vh/th + ve/th)*1e-5; % conversion
to [bar]

vtot = (vc + vk + vr + vh + ve)*1e6; % conversion to [cc]

fprintf('*****\n\n')

% Plot Alan Organ's particle mass distribution in Natural Coordinates

% fprintf('Do you want particle mass distribution plot \n');

% choice = input('y)es or n)o: ','s');

% if(strncmp(choice,'y',1))

% plotmass

% end

end

%=====

function [vc,ve,dvc,dve,v] = volume(theta)

% determine working space volume variations and derivatives

% Argument: theta - current cycle angle [radians]

% Returned values:

% vc, ve - compression, expansion space volumes [m^3]

% dvc, dve - compression, expansion space volume derivatives

[vc,ve,dvc,dve,v] = freepiston(theta);

%=====

function [vc,ve,dvc,dve,v] = freepiston(theta)

% determine drive volume variations and derivatives of FPSE

% Argument: theta - current cycle angle [radians]

% Returned values:

% vc, ve - compression, expansion space volumes [m^3]

```

```

% dvc, dve - compression, expansion space volume derivatives

global vclc % compression clearence volume [m^3]

global vcle % expansion clearence volume [m^3]

global vswc % compression swept volume [m^3]

global vswe % expansion swept volume [m^3]

global alpha % phase angle advance of expansion space [radians]

% theta = 0:5:360;

vc  = vclc + 0.5*vswc*(1 + cos(theta));

ve  = vcle + 0.5*vswe*(1 + cos(theta + alpha));

dvc = -0.5*vswc*sin(theta);

dve = -0.5*vswe*sin(theta + alpha);

v    = vc +ve;

end

function [mu,kgas,re] = reynum(t,g,d)

% evaluate dynamic viscosity, thermal conductivity, Reynolds number

% Arguments:

% t - gas temperature [K]

% g - mass flux [kg/m^2.s]

% d - hydraulic diameter [m]

% Returned values:

% mu - gas dynamic viscosity [kg.m/s]

% kgas - gas thermal conductivity [W/m.K]

% re - Reynolds number

global cp % specific heat capacity at constant pressure [J/kg.K]

global mu0 % dynamic viscosity at reference temp t0 [kg.m/s]

global t0 % reference temperature [K]

```

```

global t_suth % Sutherland constant [K]

global prandtl % Prandtl number

mu = mu0*(t0 + t_suth)/(t + t_suth)*(t/t0)^1.5;

kgas = cp*mu/prandtl;

re = abs(g)*d/mu;

if(re < 1)

re = 1;

end

end

function plotsimple(var,dvar)

% various plots of Simple simulation results

% Arguments:

% var(23,37) array of variable values every 10 degrees (0 - 360)

% dvar(16,37) array of derivatives every 10 degrees (0 - 360)

% Row indices of the var, dvar arrays:

TC = 1; % Compression space temperature [K]

TE = 2; % Expansion space temperature [K]

QK = 3; % Heat transferred to the cooler [J]

QR = 4; % Heat transferred to the regenerator [J]

QH = 5; % Heat transferred to the heater [J]

WC = 6; % Work done by the compression space [J]

WE = 7; % Work done by the expansion space [J]

W = 8; % Total work done (WC + WE) [J]

P = 9; % Pressure [Pa]

VC = 10; % Compression space volume [m^3]

VE = 11; % Expansion space volume [m^3]

```

```

MC = 12; % Mass of gas in the compression space [kg]
MK = 13; % Mass of gas in the cooler [kg]
MR = 14; % Mass of gas in the regenerator [kg]
MH = 15; % Mass of gas in the heater [kg]
ME = 16; % Mass of gas in the expansion space [kg]
TCK = 17; % Conditional temperature compression space / cooler [K]
THE = 18; % Conditional temperature heater / expansion space [K]
GACK = 19; % Conditional mass flow compression space / cooler [kg/rad]
GAKR = 20; % Conditional mass flow cooler / regenerator [kg/rad]
GARH = 21; % Conditional mass flow regenerator / heater [kg/rad]
GAHE = 22; % Conditional mass flow heater / expansion space [kg/rad]
V = 23; % Total volume ([m^3]
% Size of var(ROWV,COL), dvar(ROWD,COL)
ROWV = 23; % number of rows in the var matrix
ROWD = 16; % number of rows in the dvar matrix
COL = 37; % number of columns in the matrices (every 10 degrees)
%=====
global tk % cooler temperatures [K]
global tr % regenerator temperatures [K]
global th % heater temperatures [K]
global vk % cooler void volume [m^3]
global vr % regen void volume [m^3]
global vh % heater void volume [m^3]
global theta % phase angle advance of expansion space [radians]
global vtot % total Schmidt volume
global p % pressure

```

```

global pmean % mean pressure [Pa]

global Qkol

global Qreg

global Qhot

global Work

global Wcom

global Wexp

% Simple and Schmidt pressure volume plotting: PV diagram figure

% vol = (var(VC,:) + vk + vr + vh + var(VE,:))*1e6; % cubic centimetres
vol = (var(V,:) + vk + vr + vh )*1e6;

pres = (var(P,:))*1e-5; % bar

plot(vol,pres,'r','LineWidth',1);

grid on

hold on

plot(vtot,p,'k','LineWidth',1);

hleg = legend('Simple PV','Schmidt PV');

hold off

xlabel('Volume [cc]')

ylabel ('Pressure [bar]')

title('Simple and Schmidt P-V diagram')

% pressure vs crank angle plotting: p-theta diagram

figure;

theta = 0:5:360;

plot(theta,p,'k-','LineWidth',1);

xlim([0 360]);

% ylim([1.5 2.5]);

```

```

grid on
hold on
x = [0,360];
y = [pmean*1e-5, pmean*1e-5];
plot(x,y,'g-','LineWidth',2);
load com_space _pressure.mat
load exp_space _pressure.mat
x = linspace(0,360,37);
plot(x,pcombar,'b-',x,pexpbar,'r-','LineWidth',1);
hleg = legend('Schmidt pressure','mean pressure','comp. space
pressure','exp. space pressure');
xlim([0 360]);
% ylim([1.5 2.5]);
hold off
grid on
xlabel('Crank angle (theta) [degree]');
ylabel('Working Space Pressure [bar]');
% title('Working space pressure (P) vs Crank Angle (Theta): P-Theta
diagram');
% Energy plottings of the Simple simulation results
figure
x = 0:10:360;
Qkol = var(QK,:); % [J]
Qreg = var(QR,:); % [J]
Qhot = var(QH,:); % [J]
Work = var(W,:); % [J]

```

```

Wcom = var(WC,:); % [J]
Wexp = var(WE,:); % [J]

plot(x,Qkol,'b-', x,Qreg,'g-', x,Qhot,'r-', x,Work,'k--', x,Wcom,'b--',
x,Wexp,'r--','LineWidth',1);

hleg = legend('Qk','Qr', 'Qh', 'Work', 'Wcom', 'Wexp');

xlim([0 360]);

grid on

xlabel('Crank angle (theta) [degrees]');

ylabel('Energy [J]');

% title('Energy vs Crank Angle (theta)');

load pressure_drop _cooler.mat
load pressure_drop _regen.mat
load pressure_drop _heater.mat
load pressure_drop _drop.mat
load pressure_drop _comp.mat
load pressure_drop _exp.mat

% Heat Exchanger pressure drop vs Crank Angle

figure;

x = 0:10:360;

% plot(x,dpkol,'b-',x,dpreg,'g-',x,dphot,'r-',x,dp,'k--','LineWidth',2);

plot(x,dpkol,'b-',x,dphot,'r-','LineWidth',1);

% hleg = legend('Cooler pressure drop','Regn. pressure drop', 'Heater
pressure drop', 'total HE pressure drop');

hleg = legend('Cooler pressure drop','Heater pressure drop');

xlim([0 360]);

grid on

```

```

xlabel('Crank angle [degrees]');
ylabel('Heat exchanger pressure drop [Pa]');
% title('Heat exchanger pressure drop vs crank angle');
% Heat Exchanger pressure drop vs Crank Angle including pressure drop in
the regenerator
figure;
x = 0:10:360;
% plot(x,dpkol,'b-',x,dpreg,'g-',x,dphot,'r-',x,dp,'k--','LineWidth',2);
plot(x,dpkol,'b-',x,dphot,'r-',x,dpreg,'g-','LineWidth',1);
hleg = legend('Cooler pressure drop','Regn. pressure drop', 'Heater
pressure drop');
xlim([0 360]);
grid on
xlabel('Crank angle [degrees]');
ylabel('Heat exchanger pressure drop [Pa]');
% title('Heat exchanger pressure drop vs crank angle');
load Efficiency.mat
% load TE.mat
load Heatpowerin.mat
load Outputpower.mat
load effecincy.mat
% Expansion space Temperature vs Actual efficiency
% figure;
% plot(eff,var(TE,:), 'k-', 'LineWidth',2);
% grid on
% xlabel('Expansion space TE [K]')

```

```

% ylabel('Efficiency [%]')

% title('Expansion Space Temp vs Actual efficiency')

% Expansion space temperature vs Actual heat power

figure;

plot(Heatpowerin,Outputpower,'k-','LineWidth',1);

ylim([0 2]);

grid on

xlabel('Heat added to the engine [W]');

ylabel('Power output [W]');

% title('Actual Heat Power added to the engine vs Actual Power [W]');

% Heat transferred to the Heater vs the Total work done

% figure;

% plot(var(QH,:),var(W,:), 'k-', 'LineWidth',2);

% ylim([0 4])

% grid on

% xlabel('Heat transferred to the heater [J]');

% ylabel('Total Work done (Wc+We) [J]');

% title('Total Work vs Heat transferred to the heater')

% Heat transferred to the engine vs the crank angle

figure;

x = 0:10:360;

Q = var(QK,:) + var(QR,:) + var(QH,:);

plot(x,Q,'k--',x,var(QK,:), 'b-',x,var(QH,:), 'r-',x,var(QR,:), 'g-

','LineWidth',1);

hleg = legend('Total heat transferred (Q)', 'QK','QH', 'QR');

xlim([0 360]);

```

```

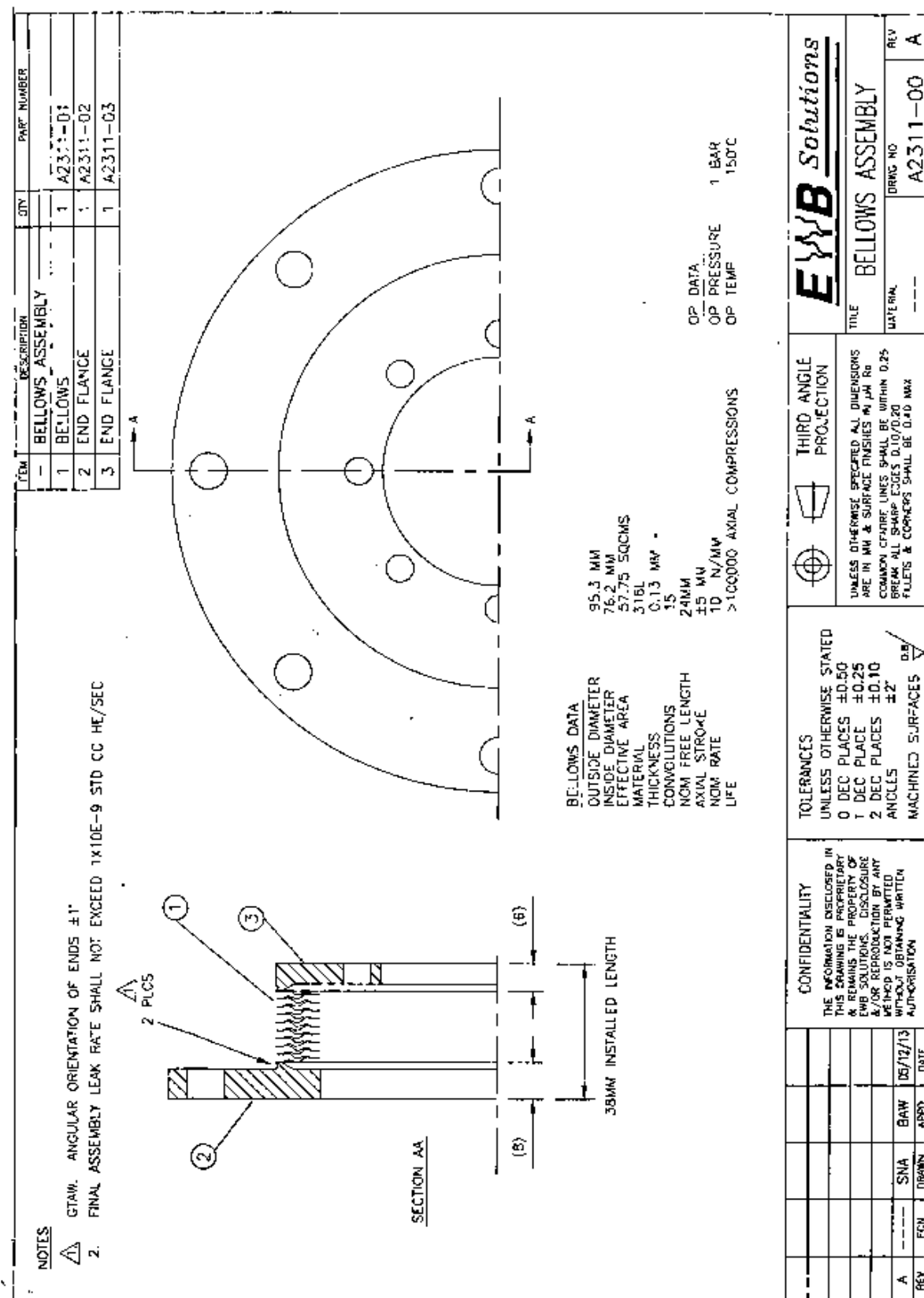
grid on
xlabel('Crank Angle (theta)[degrees]');
ylabel('Heat transferred [J]');
% title('Heat transfer vs Crank Angle (theta)');
% Gas mass flow (Kg/rad) vs the Crank angle
figure;
GA = var(GACK,:) + var(GAKR,:) + var(GARH,:) + var(GAHE,:);
x = 0:10:360;
plot(x,GA,'k--',x,var(GACK,:), 'b-',x,var(GAKR,:), 'g--',x,var(GARH,:), 'k-
',x,var(GAHE,:), 'r--', 'LineWidth',1);
hleg = legend('Total mass flow (Kg/rad)', 'GACK', 'GAKR', 'GARH', 'GAHE');
xlim([0 360]);

grid on
xlabel('Crank angle (theta) [degrees]');
ylabel('mass flow [Kg/rad]');
% title('Mass Flow vs Crank Angle');
% Gas mass vs the Crank angle
figure;
% M = var(MC,:) + var(MK,:) + var(MR,:) + var(MH,:) + var(ME,:);
x = 0:10:360;
% plot(x,M,'k--',x,var(MC,:), 'b', x,var(MK,:), 'cyan', x,var(MR,:), 'g',
x,var(MH,:), 'magenta', x,var(ME,:), 'red', 'LineWidth',2);
plot(x,var(MC,:),x,var(ME,:), 'red', 'LineWidth',1);
% hleg = legend('Total mass', 'MC', 'MK', 'MR', 'MH', 'ME');
hleg = legend('MC', 'ME');
xlim([0 360]);

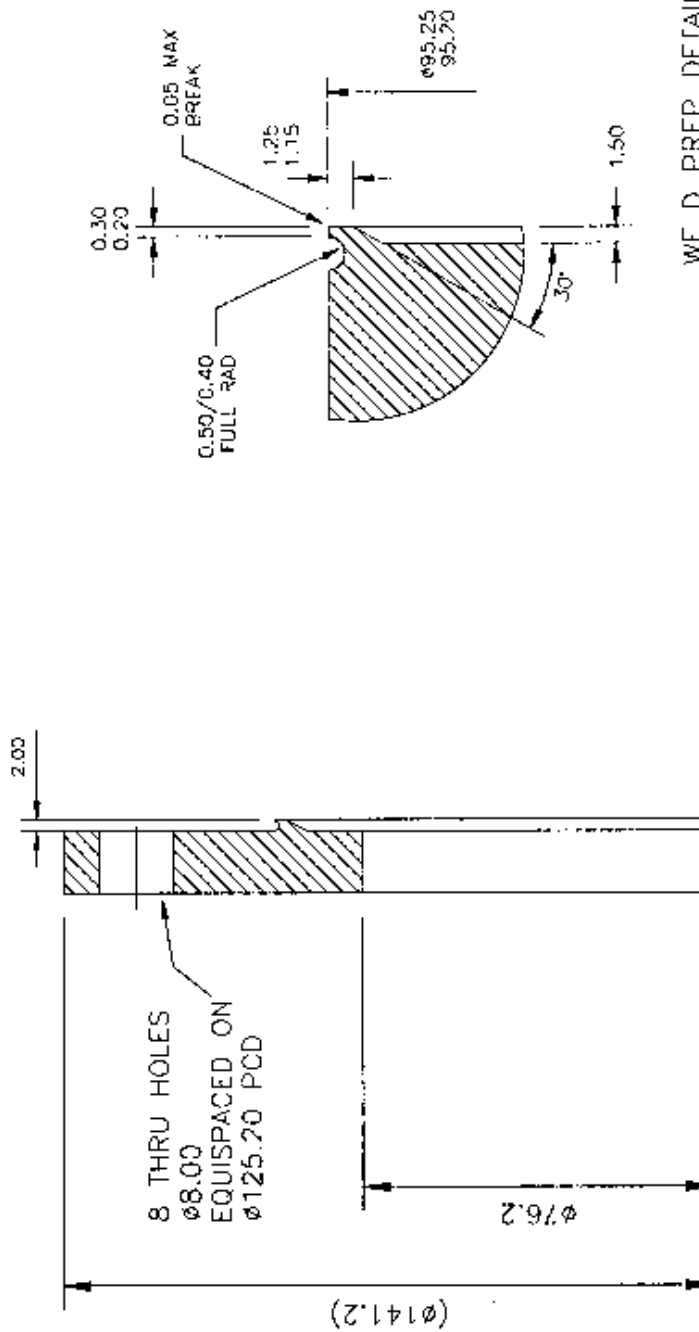
```

```
% ylim([-1*10^-4 6*10^-4]);  
grid on  
xlabel('Crank Angle (theta)[degree]')  
ylabel('Mass of gas [Kg]')  
% title('Gas Mass vs Crank angle')  
fprintf('quitting Simple plots...\n \n');  
end
```

Appendix B: Bellows' dimensions and specifications

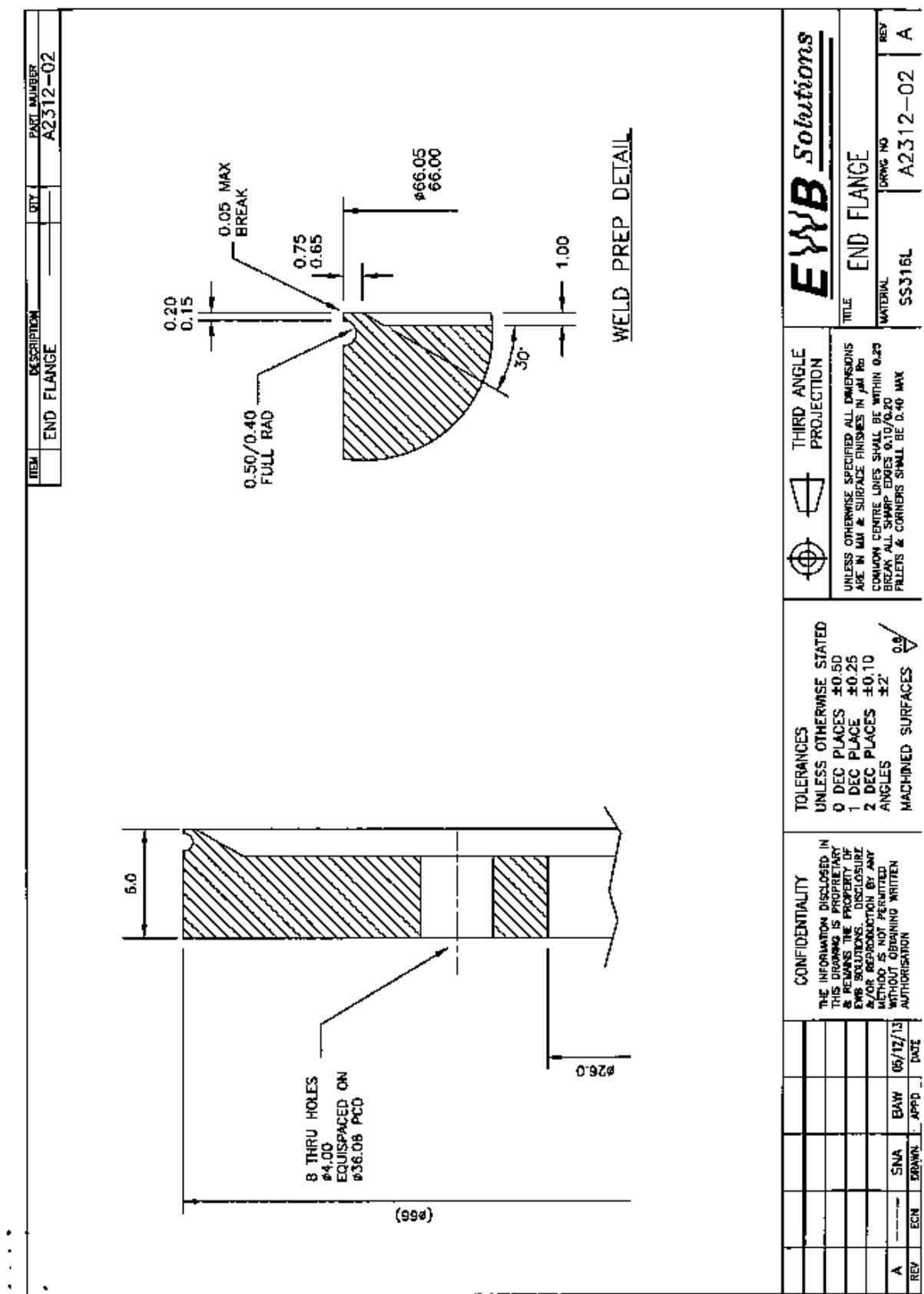


ITEM	DESCRIPTION	QTY	PART NUMBER
	END FLANGE		A2311-02



WELD PREP DETAIL

				EVB Solutions	
A		REV		A	
SNA		BAW		A2311-02	
ECN		DRWNG		SS316L	
QTY		QTY		SAUCE: NO	
				DATE	
				REV	
				A	
				A	
				A	
				A	
				A	
				A	
				A	
				A	
				A	
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Appendix C: Natural diaphragm properties

NITRILE (NBR)		
Nitrile is the most widely used elastomer in the seal industry. The popularity of nitrile is due to its excellent resistance to petroleum products and its ability to be compounded for service over a temperature range of -22°F to 212°F. Nitrile is a copolymer of butadiene and acrylonitrile. Variation in proportions of these polymers is possible to accommodate specific requirements. An increase in acrylonitrile content increases resistance to heat plus petroleum base oils and fuels but decreases low temperature flexibility. Military AN and MS O ring specifications require nitrile compounds with low acrylonitrile content to insure low temperature performance. Nitrile provides excellent compression set, tear, and abrasion resistance. The major limiting properties of nitrile are its poor ozone and weather resistance and moderate heat resistance, but in many application these are not limiting factors.	Temperature Range (dry heat)	
	low	high
	-22 °F	212 °F
	-30 °C	100 °C
	Application Advantages	
	» excellent compression set, » superior tear resistance » abrasion resistance	
	Application Disadvantages	
	» poor weather resistance » moderate heat resistance	
	Modifications	
	» acrylonitrile content (ACN) from 18% to 50% » peroxide vs. sulfur donor cure system » XNBR improved wear resistance formulation	
Primary Uses	Specialized Applications	
O-rings, rubber seals and custom molded rubber components for: » Oil resistant applications » Low temperature applications » Fuel systems, automotive, marine, and aircraft » General Industrial Use	» NBR NSF standard 61 for potable water applications » NBR WRc, KTW water applications » NBR FDA white list compounds	

Appendix D: Data taker specifications

DT500 & DT600 Range

dataTaker®

- General Purpose Low Power Data Logger
- 10-30 Sensor Channels, 7 Digital Channels
- Unique Universal Channels
- Up to 1,390,000 Data Points
- Stand Alone and Real Time Data Acquisition
- Remote Monitoring and Control
- Removable Screw Terminals
- Expandable

FREE
Software &
Technical
Support



Specifications

The dataTaker DT500 & DT600 Range

The dataTaker DT500 range of general purpose, battery powered data acquisition and data logging systems measure inputs from most sensor types. Data can be conveniently and securely stored in battery backed RAM and removable memory cards.

The dataTaker DT500 range consists of four models: DT500, DT505, DT600, DT605.

The DT500 and DT600 models have Solid State Channel Selectors. (Low Voltage)

The DT505 and DT605 models have Relay Channel Selectors. (High Voltage)

The DT600 and DT605 both have an integral display and keypad that allows users to view channel data, alarm status, and system information including time, battery status and amount of data stored. Programmable function keys allow keypad control over the unit's operation.

Applications include:

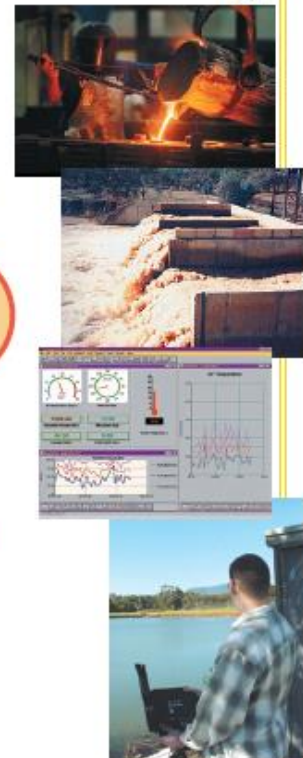
- Fault Finding
- Automotive Testing
- Temperature Profiling
- Weather Stations
- Flood Warnings
- Research & Development
- Process Monitoring
- Machine Down Time
- Load Cells
- Building Monitoring
- Strain Gauges

dataTaker software and Resource CD

The dataTaker Resource CD is provided FREE with every new logger. It contains software to enable easy setup, fast download, real time viewing of data and direct export to excel spreadsheets. The Resource CD also contains helpful training videos, manuals, application and technical notes and other valuable utilities.

DeLogger™ 4 Pro is the enhanced graphical package including additional automation, reporting, database and remote dataTaker management features.

For your nearest local DataTaker distributor visit www.dataTaker.com



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Analog Channels

Channel Number

Number of input channels depends on sensor wiring configuration. Sensor configurations may be mixed.
Two wire with one shared terminal: 30
Three and Four wire: 10
Expansion: by Channel Expansion Modules (CEM)

Fundamental Input Ranges

Full Scale	Resolution	Full Scale	Resolution
±25.00 mVdc	2.00 µV	50 Ω	25 mΩ
±250.0 mVdc	20.00 µV	500 Ω	250 mΩ
±2.50 Vdc	200.00 µV	5,000 Ω	2,500 mΩ
±100.0 Vdc*	500.00 µV	100 Hz	0.01 %
±0.25 mA	0.20 µA	10 kHz	0.01 %
±2.50 mA	1.00 µA		
±25.00 mA	10.00 µA		

*100 Vdc range of DT505 and DT605 only

Accuracy

Measurement at	25°C	-45°C to 60°C
DC Voltage	0.15%	0.25%
DC Current	0.25%	0.35%
DC Resistance	0.20%	0.30%

Multiplexer (Channel Selector)

DT500 and DT600: solid-state ±5V input range
DT505 and DT605: relay ±100V input
Input impedance: 1MΩ or >100MΩ, programmable
Common mode range: ±3.5V
DT500 and DT600: ±3.5V
DT505 and DT605: ±100V on 100V range

Sampling

Sampling for accuracy and noise rejection by integrating over 50/60Hz line period
Maximum sample speed: 25Hz (up to 70Hz without noise rejection)
Effective resolution: 15 bits
Linearity: 0.01%
Common mode rejection 25mV range: >90dB
Line (50/60Hz) series mode rejection: >35dB

Sensor Excitation

Each channel: 4.5V, 250µA or 2.5mA

DC voltage: 5V at 100mA (max.) switched

Internal Channels

Temperature (thermocouple reference junction): 1
Reference voltage channels: 1
Internal battery voltage: 1

Sensor Support

Supports a wide range of sensors types including, but not limited to the following:

Thermocouples

Types: B, C, D, E, G, J, K, N, R, S, T

Reference junction compensation accuracy:

Case temperature	2.5°C	-20 to +60°C
Accuracy	±1.0°C	±1.5°C

RTDs

Types: Pt, Ni, Cu

Resistance range: 10Ω to 2kΩ

Measurement accuracy:

4 wire: 0.15% of resistance value

3 wire: 0.25% of resistance value

Monolithic Temperature Sensors

Types supported: LM34, LM35, AD590

Thermistors

Types: YSI 400xx Series

Resistance range: <7kΩ

<20kΩ with parallel resistor

Bridge Sensors

Configurations: 4-wire and 6-wire

Bridge completion: external or internal half bridge

4-20mA Current Loops

Shunt value: 100Ω to a shared common

Accuracy: 0.25% at 25°C

Sensors - Comments

A wide range of sensor scaling and linearising facilities is provided including polynomials, expressions and functions.

Digital Channels

Number of channels

Bi-directional channels: 4

Dedicated counter channels: 3

Digital Input

Number: 4, shared with output channels

Input Type: logic level (protected and 5kΩ pull-up to 5V)

Counter Channels

Number: 4 low speed (10kHz) shared with input channels)

3 high speed (1kHz, sleep mode) with switchable internal clocking options

Size: 16 bit (65,535 counts)

Digital Output

Number: 4, shared with input channels

Output type: open-collector npn transistor

Rating: +30V, 100mA

Calculation Channels

Any expression involving variables and functions including: sin(), cos(), tan(), asin(), acos(), atan(), abs(), sqrt(), average, maximum, minimum, time of max., time of min., variance, integral, histogram

Scheduling of Data Acquisition

Number of schedules: 4 acquisition schedules

1 immediate schedule

1 alarm schedule

Scan triggers: time base or digital event

Conditional scanning: while digital input high

Time based scheduling: from seconds to months in increments of 1 second, 1 minute, 1 hour and 1 day

Maximum scheduled rate: 1 second or as fast as possible, typically 25 samples per second

Dynamic scan time base change: yes

Maximum number of channel entries: 110

Alarms

Condition: high, low, within range and outside range

Delay: optional time period for alarm response

Actions: set digital outputs, execute any dataTaker commands. Alarms can be combined in a logical fashion.

Data Storage

Internal

Type: battery backed SRAM

Capacity: 166,530 data points

PC Card

Types: SRAM up to 4MByte, Type 1

Card voltage: 5V types

Capacity: up to 1,390,000 data points

Data format: proprietary

Download Data Format

Format: ASCII floating point, fixed point or exponential formats

Compatibility: spreadsheets, word processors, graphing packages, statistical programs and SCADA software

Serial Interface (RS232)

The DT500 range are programmed and data extracted via the RS232 serial interface

Speed: 300 to 9600 baud (9600 default)

Handshake: XON and XOFF

Wake from sleep: yes

Isolation: 500V

Compatibility: computers, modems, satellite-modems, radio-modems and printers

Network Interface (Multiple dataTaker only)

Standard: RS485

Protocol: proprietary with error correction

Speed: 1200 Baud

Distance: 1000 meter maximum

System

Display and Keypad

Models: on DT600 and DT605 only

Type: LCD, 2 lines by 16 characters, back light

Display functions: channels data, alarms, battery status, data capacity

Key pad: 5 keys for scrolling, function execution

Beeper: for alarms, etc.

Indicator LED's: 3 programmable

Real Time Clock

For time stamping of data, scheduling and timers

Normal resolution: 1 second

Accuracy: 2 seconds per day (25°C)

Power Supply

Voltage range: 11 to 24Vdc or 9 to 18Vdc

Power Consumption

In normal mode: 1W (2W with battery charging)

Sleeping: 2mW (350µA from battery)

Typical low power operation: 20mW

Internal Main Battery

Chemistry: lead acid gel cell

Voltage (capacity): 6V (1.2Ah)

Temperature compensation: -10°C to +70°C

Operating time: Normal: approximately 10 hours

Low power: approximately 3 months

Internal Backup Battery

For real time clock and internal data storage backup

Type: 3V 15mA Lithium

Physical and Environment

Construction: Powder coated fabricated steel

Physical dimensions: 280 x 110 x 85mm (height 104mm with PC Card)

Weight: 2.2kg (4kg shipping)

Environment temperature range: -45°C to 70°C

Humidity: 85%RH, non-condensing

Accessories Included

Line adaptor: 110/240Vac, 500mA

Comms cable: for PC RS232 and USB adaptor

Resource CD which includes standard software

Manuals: "Getting Started with dataTaker"

"User's Manual"

Options & Accessories

Channel Expansion Module (CEMS3)

Multiplexer: relay

Number: 2 per DT500 Series unit

Channel number:

10 two wire

30 two wire shared terminals

20 digital inputs

10 digital outputs, 5 with relay contacts

Portable Carrying Case (PE500)

Capacity: 1 DT500 range unit + 1 x CEMS3

(Requires AS1072)

Environmental protection: IP66

SRAM PC Card (MC1024P, MC4096P)

Capacity: 1MByte, approximately 340,000 data points

4MByte, approximately 1,390,000 data points

DeLogger™ 4 Pro

Graphical programming and supervision software.

Supports a large network of DT500 range units connected via modem. Features include comprehensive plotting, reporting, mimics, database, web publishing and other powerful capabilities.



Your local distributor



Warranty: The dataTaker DT500 & DT600 range are covered by a 3 year warranty on workmanship and parts. For further information on the dataTaker range, or for useful downloads, visit the DataTaker web site www.dataTaker.com or contact your nearest DataTaker office or distributor.

Quality Statement: DataTaker operates a Quality Management System complying with ISO 9001:2000. It is DataTaker's policy to supply customers with products which are fit for their intended purpose, safe in use, perform reliably to published specification and are backed by a fast and efficient customer support service.

Trademarks: dataTaker is a registered trademark of DataTaker Pty Ltd.

Specifications: DataTaker Pty Ltd reserves the right to change product specifications at any time without notice.

Manufactured and designed in Australia.



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Appendix E: Oscilloscope specifications

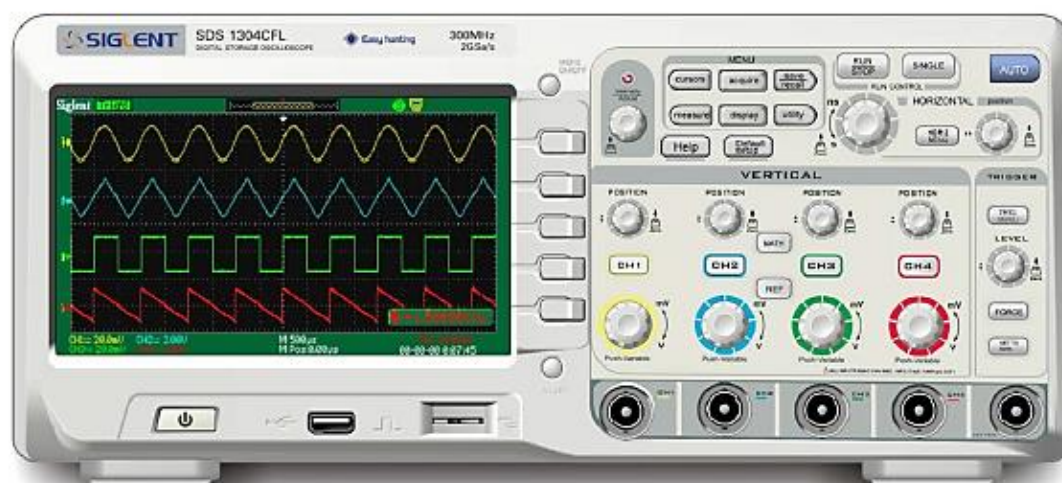


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Catalogue for Digital Storage Oscilloscope

As the biggest ODM manufacturer of DSO in the world, we have launched 4 series of digital storage oscilloscope up to now, which are SDS1000CFL, SDS1000CM, SDS1000D, SDS1000C. SDS1000CFL series is 7 inches wide screen 4CH DSO, others are 5.7" screen 2CH DSO.

SDS1000CFL Series



◆ Overview

Adhering to Siglent's goal of multi-function with high performance-to-price ratio, SDS1000CFL series oscilloscopes have been launched with international research concept, powerful functions and perfect operation. By providing up to 4 channels and one external trigger input channel, SDS1000CFL can capture and display multiple signals simultaneously, which meets many new product development and verification needs. With max 2GSa/s real time sampling rate, a 7" color TFT LCD display, SDS1000CFL is perfect for working in complex hardware design situations that need high bandwidth and fast real time sampling rate. Besides, with 24kpts memory depth, instead of the common 4Kpts, this newly-launched oscilloscope leads the oscilloscope industry. Deep memory enables engineers to detect fast, intermittent signals easier. Powerful trigger and analysis functions facilitate waveform capture and examination, which greatly promotes test efficiency. Multiple interface configurations allow optimum connections between PC and oscilloscope, meeting the needs of processing waveform data and creates fully-functioning test systems rapidly.

< Specifications

Model	SDS1304CFL /SDS1302CFL	SDS1204CFL /SDS1202CFL	SDS1104CFL /SDS1102CFL	SDS1074CFL /SDS1072CFL
Bandwidth	300MHZ	200MHZ	100MHZ	70MHZ
Channels	4CH+1EXT; 2CH+1EXT			
Real-time sample rate	1GSa/s(each channel) 2GSa/s (half channel)			
Equivalent sample rate	50GSa/s			
Depth memory	12K(each channel) 24K(half channel)			
Rise time	<1.2ns	<1.7ns	3.5ns	5.8ns
Input impedance	1MΩ 13Pf, 50Ω		1MΩ 13Pf	
Time base range	1.0ns/div-50s/div	2.5ns/div-50s/div	2.5ns/div-50s/div	5.0ns/div-50s/div
	Roll: 100ms-50s/div			
Vertical sensitivity	2mV - 5V/div(1-2-5 sequence)			
Vertical resolution	8bit			
Trigger source	CH1、CH2、CH3、CH4、Ext、Ext/5、AC Line			
Trigger types	Edge, pulse, video, slope, alternative			
Math operation	+, -, *, FFT, invert			
Digital filter	High pass, low pass, band pass, band stop			
Max input voltage	±400V PK-PK CAT 1			
Internal save	4 groups of reference waveform, 20 groups of settings, 20 groups of waveforms			
External save	Bitmap save, CSV save, waveform save, setting save			
Language	English, German, French, Japanese, Simplified Chinese, traditional Chinese, Korean, Russian, Arabic, Spanish, Italian, Portuguese			
Interface	double USB Host, USB Device, LAN, Pass/Fail interface			
Colorful display	TFT (480*234) 7" (178mm) LCD			

Size: 336mm×152mm×118mm(L×W×H)

Gross Weight: 7.5kg

Net Weight: 4.5kg



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< **Application:**

- embedded electronic circuit design and test;
- mechanical and electrical product design and analysis;
- education and research;
- product quality control;
- real-time signal display;
- production test, circuit function test;

< **Features and benefits:**

* **High performance**

- bandwidth: 300MHz, 200MHz, 100MHz, 60MHz
- four analog channels plus one external trigger channel
- 2GSa/s sample rate and 10Kpts depth memory for each channel

* **Complete connection standard**

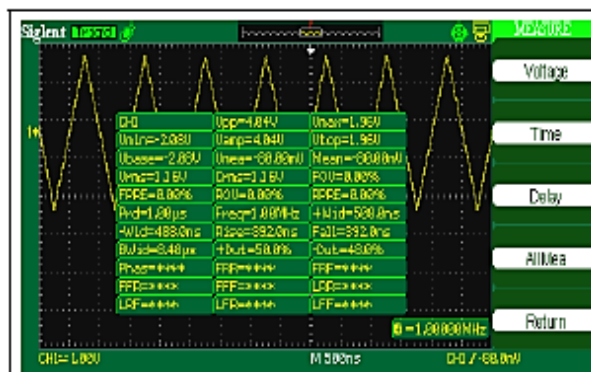
- Double USB Host: supports USB flash drive and firmware upgrades; USB Device;
- Supports PC remote control and PictBridge printing, LAN, RS-232, Pass/Fail interface;
- Compatible with LXI standard, supports SCPI programming command;

* **More signal display**

- 7", 64k color, wide viewing-angle TFT-LCD screen;
- very large screen display: 8x18 grid waveform display scope;

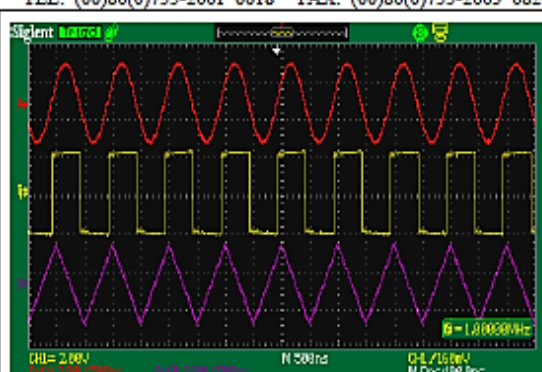
* **More efficient**

- 6 digit hardware frequency counter, real time counting display;
- Storage for: Four reference waveforms, 20 settings, 20 internally saved waveforms;
- 5 triggers: Edge, pulse, video, slope, alternative
- 4 saved types: setup data, waveform data, interface bitmap, CSV document
- unique digital filtering and data recorder function;
- real time display date and time;
- 12 languages, embedded online help, one key storing and one key printing function;



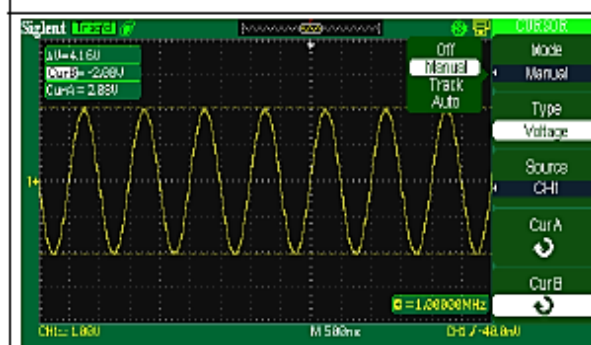
auto measure function

32 types of auto-measurements, 3 types of cursor modes, can display all the measured parameters on the screen simultaneously; plenty cursor modes meet measuring needs.



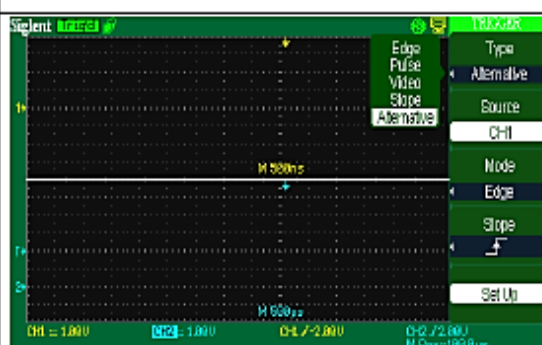
reference waveform

You can store two groups of reference waveform and display them simultaneously on the screen, allowing these two waveforms to be compared



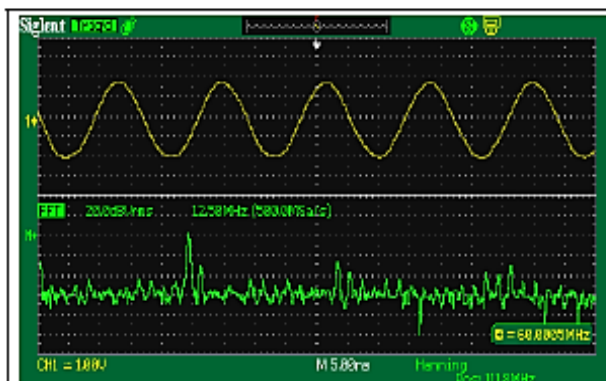
manual mode

including time, voltage test. The distance between two cursors can be adjusted manually while measuring time and voltage

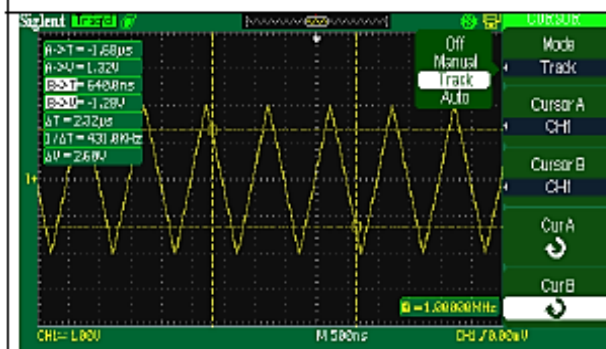


alternative trigger

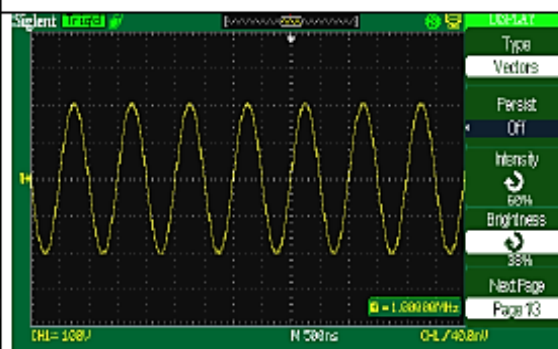
In order to stabilize two signals with different frequency you can choose different triggers for each channel, which like having an analog oscilloscope in a digital oscilloscope


FFT split display

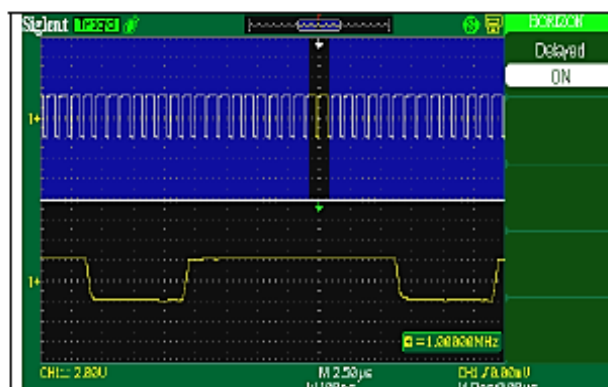
Equipped with the function of analyzing signals in frequency domain, channel waveform and FFT waveform can be displayed simultaneously on the split screen, making waveform observation more convenient.


Tracking mode

A cursor can track waveforms automatically, through rotating a universal knob to adjust cursor's horizontal position-then its vertical position will be altered accordingly.

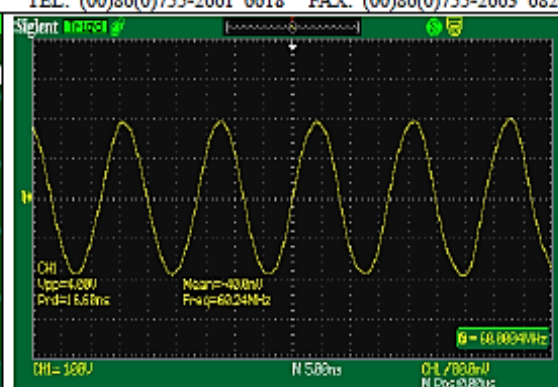

Waveform intensity is adjustable

the waveform can be observed more clearly due to the intensity of waveform can be adjusted accordingly.



Delayed scan

You can use EasyZoom, a dual-time-base system window extension technology to expand the partial waveform perfectly, and display the whole waveform as well as part of it.



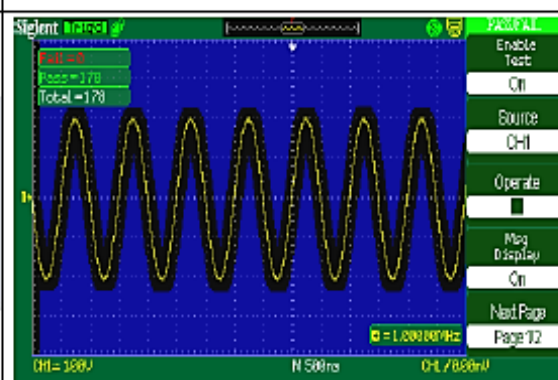
pop-up menu

pop-up menu display mode enables the menu to be hidden at any time, thus waveforms can be observed more clearly by displaying in 18grids.



sufficient interface configuration

SDS1000CFL series oscilloscopes are compatible with the LXI standard and support SCPI programming control commands, making test systems easy to create and configure, and easily integrate into test systems with display also.



PASS/FAIL

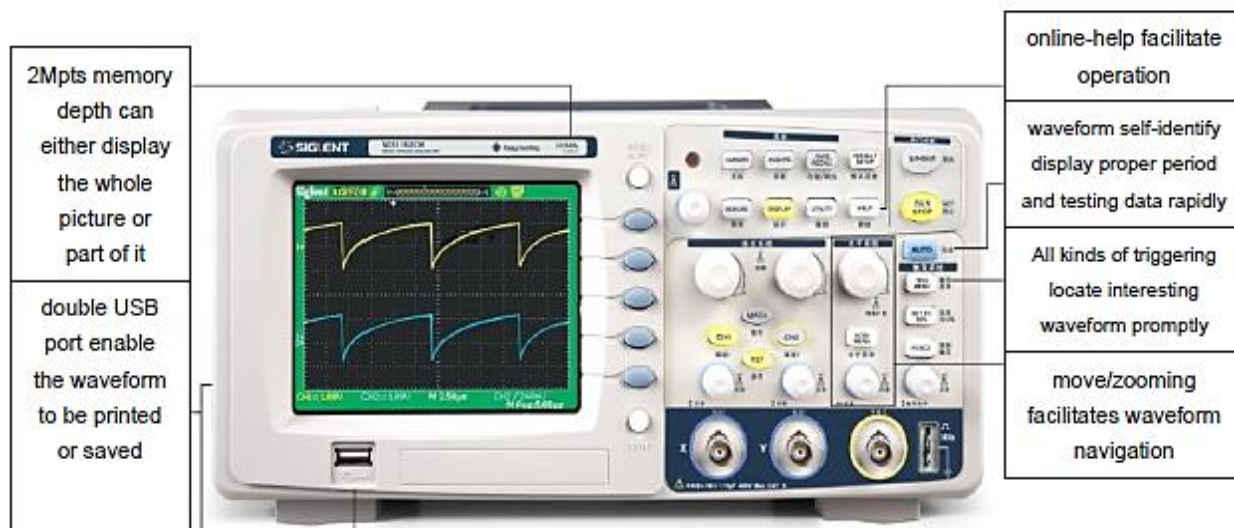
You can also make special signal templates for testing item on a production line, improving testing efficiency, with the capability of stopping production immediately using the pass/fail output when the output signal is wrong.



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SDS1000CM Series

- 1GSa/s real time sample rate, 2Mpts, the highest memory depth among same level products.
- 150MHz bandwidth



Size: 305mm*133mm*154mm Weight:2.3kg

◆ Application

- electronic circuit design and debugging
- electro circuit function test
- inspect instantaneous signal
- industrial control and measuring
- products quality control
- education and training

◆ Features and benefits

- The highest bandwidth is 150MHZ, real time sample rate is 1GSa/s, equivalent sampling rate is 50GSa/s
- 2Mpts memory depth
- Five types of triggering: edge, pulse, video, slope, alternative
- Screen-saver function (one minute to 5 hours)
- Unique digital filter and data recorder function



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- Interface configuration: USB Host, support U memory saving and firmware updating; USB Device; support PC remote control and PictBridge printing; RS-232; Pass/Fail interface

◆ Specification

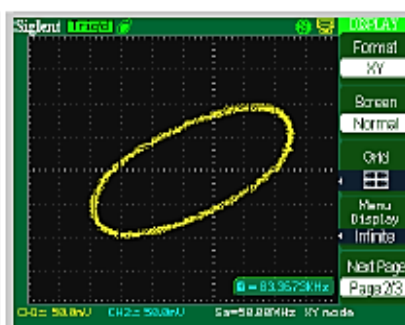
Model	SDS1152CM	SDS1102CM	SDS1062CM
Bandwidth	150MHZ	100MHZ	60MHZ
Channels	double channel plus one EXT trigger channel		
Real time sampling rate	1G Sa/s(single channel), 500M Sa/s(double channel)		
Equivalent sampling rate	50G Sa/s		
Memory depth	2Mpts		
Rise time	<2.4ns	<3.5ns	<5.8ns
Input impedance	1MΩ 17pF		
Time base range	2.5ns/div-50s/div,	2.5ns/div-50s/div,	5ns/div-50s/div
	Roll: 100ms-50s/div		
Vertical sensitivity	2mv-10v/div(1-2-5 sequence)		
Vertical resolution	8bit		
Trigger source	CH1, CH2, EXT, EXT/5, AC Line		
Trigger types	Edge, pulse, video, slope, alternative		
Math operation	+, -, *, / ,FFT		
Digital filter	High pass, low pass, band pass, band stop		
Max input voltage	±400V Pk-Pk CAT 1		
Internal save	2 groups of reference waveform, 20 groups of setting, 20 groups of waveform		
External save	Bitmap save, CSV save, waveform save, setting save		
Lasting	Turn off, 1s, 2s, 5s, infinite		
Language	English, Russian, French, Japanese, Korean, German, Arabic, Spanish, Italian, Portuguese, Simplified Chinese, Traditional Chinese		
Connection	USB Host, USB Device, RS-232, Pass/Fail out		
Display	5.7 inches color TFT-LCD		
Power	AC 100-240V, 47HZ-440HZ, 50VA Max		

◆ Function



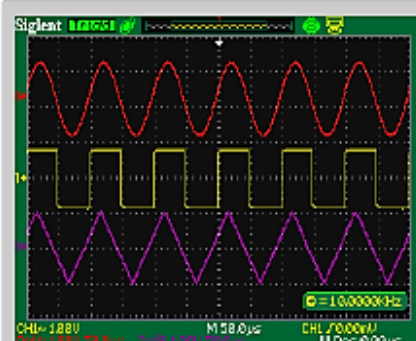
32 auto measurements

Measure 32 parameters automatically; Users can select each one separately. 32 parameters can display simultaneously.



XY mode display

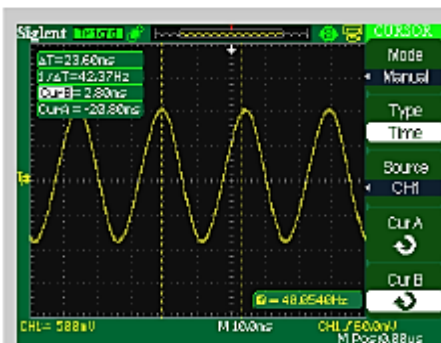
Use XY format to analyze phase shift, the picture in this mode formed by points.



Reference waveform

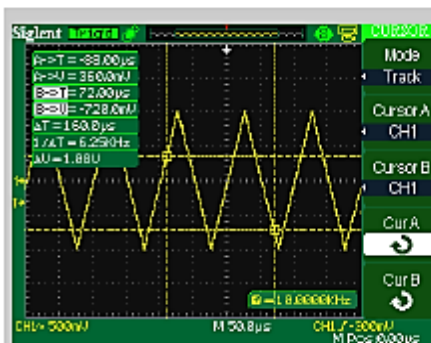
Can save two groups of reference waveform, and be able to open them at the same time, so the tested waveform and reference waveform can be compared.

◆ Cursor mode



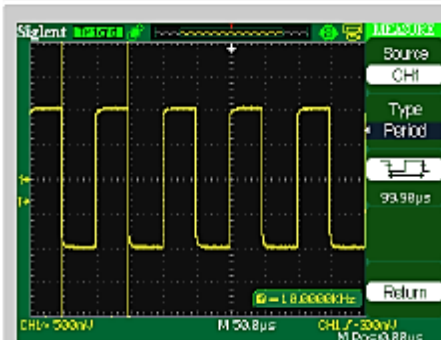
Manual mode

Including time and voltage measuring. By adjusting cursors' distance manually to measure time and voltage.



Tracking mode

The cursor can track waveform automatically. Alter cursor's horizontal position by rotating universal knob, and then the vertical position will change automatically.



Auto-measuring

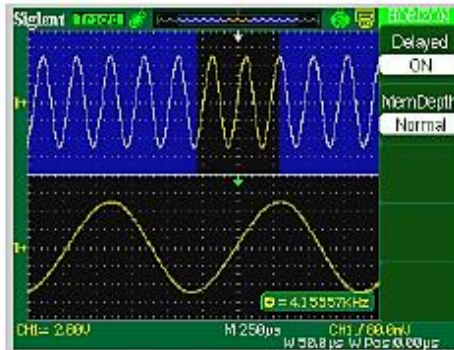
System will adjust cursor position according to the change of selective auto-measuring parameter and calculate corresponding parameters value.

◆ **Excellent design**



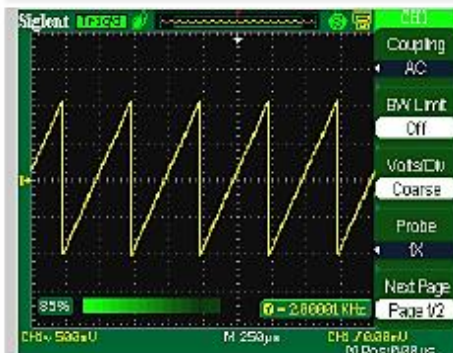
Independent channel control

Independent channel control, simple operation; clear back light design of button makes working state is leap to eye.



Delayed scan

Apply the expansion technique of double-time base EasyZoom window to expand waveform in detail, either display the complete waveform or part of it.



Adjustable waveform intensity

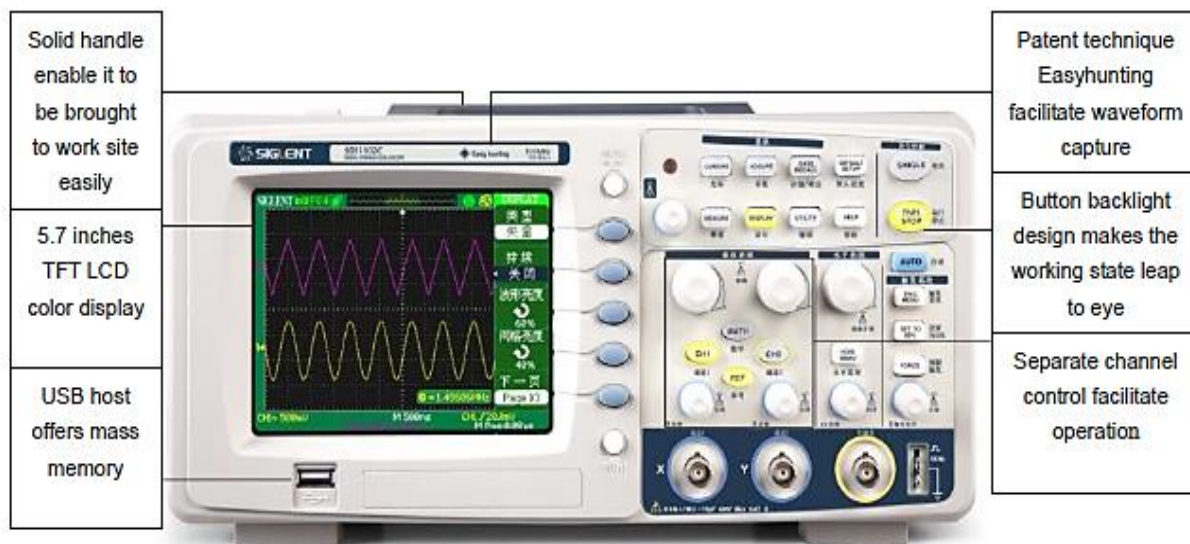
The waveform intensity can be adjusted according to different requirement.



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SDS1000C&SDS1000D Series

- ◆ Cost-effective and high performance;
- ◆ 500MSa/s real time sample rate, 25Mhz-100Mhz bandwidth;



Size: 305mm*133mm*154mm

Weight: 2.3kg

◆ Application

- electronic circuit design and modify
- industrial control and power measuring
- inspect instant signal
- production test
- high education
- service

◆ Features and benefits

- With the highest bandwidth 100MHZ, real time sampling rate 500MSa/s, equivalent sampling rate 50GSa/s
- Ultra slim, small volume, portable
- 3 types of cursor mode, 32 kinds of auto-measuring reference
- 5 triggering functions: edge, pulse, video, slope, alternative
- 2 groups of reference waveform, 20 groups of setting, 20 groups of waveform internal saving



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- Standard connection: USB Host; U memory saving and firmware updating; USB device; PC remote control and PictBridge printing; RS-232; Pass/Fail interface

◆ Specifications of SDS1000C series

Model	SDS1022C
Bandwidth	25MHZ
Channels	double channel plus one EXT trigger channel
Real time sampling rate	500MSa/s(single channel), 250M Sa/s(double channel)
Equivalent sampling rate	10G Sa/s
Memory depth	4K/CH
Rise time	<14ns
Input impedance	1MΩ 17pF
Time base range	25ns/div-50s/div Roll:100ms-50s/div
Vertical sensitivity	2mv-5v/div(1-2-5 sequence)
Vertical resolution	8bit
Trigger source	CH1, CH2, EXT, EXT/5, AC Line
Trigger types	Edge, pulse, video, slope, alternative
Math operation	+, -, *, /, FFT
Digital filter	High pass, low pass, band pass, band stop
Max input voltage	±400V Pk-Pk CAT 1
Internal save	2 groups of reference waveform, 20 groups of settings, 20 groups of waveform
External save	Bitmap save, CSV save, waveform save, setting save
Lasting	Turn off, 1s, 2s, 5s, infinite
Language	English, Russian, French, Japanese, Korean, German, Arabic, Spanish, Italian, Portuguese, Simplified Chinese, Traditional Chinese
Connection	USB Host, USB Device, RS-232, Pass/Fail out
Display	5.7 inches color TFT-LCD
Power	AC 100-240V, 47HZ-440HZ, 50VA Max

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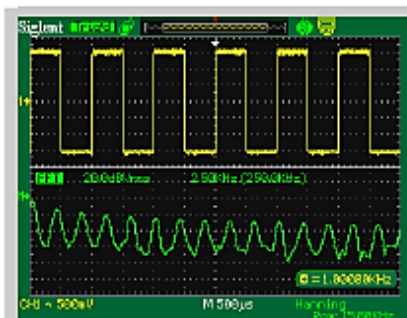
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◆ Specifications of SDS1000D series

Model	SDS1202D	SDS1102D	SDS1062D	SDS1042D	SDS1022D
Bandwidth	200MHZ	100MHZ	60MHZ	40MHZ	25MHZ
Channels	double channel plus one EXT trigger channel				
Real time sampling rate	500MSa/s(single channel), 250M Sa/s(double channel)				
Equivalent sampling rate	50G Sa/s	50G Sa/s	50G Sa/s	25G Sa/s	10G Sa/s
Memory depth	32K/CH				
Rise time	<1.7ns	<3.5ns	<5.8ns	<8.8ns	<14ns
Input impedance	1MΩ 17pF				
Time base range	2.5ns/div-50s/div	2.5ns/div-50s/div	5ns/div-50s/div	10ns/div-50s/d	25ns/div-50s/div
	v	v		iv	
	Roll:100ms-50s/div				
Vertical sensitivity	2mv-5v/div(1-2-5 sequence)				
Vertical resolution	8bit				
Trigger source	CH1, CH2, EXT, EXT/5, AC Line				
Trigger types	Edge, pulse, video, slope, alternative				
Math operation	+,-, *, /,FFT				
Digital filter	High pass,low pass,band pass, band stop				
Max input voltage	±400V Pk-Pk CAT 1				
Internal save	2 groups of reference waveform, 20 groups of settings, 20 groups of waveform				
External save	Bitmap save, CSV save, waveform save, setting save				
Lasting	Turn off, 1s, 2s, 5s, infinite				
Language	English, Russian, French, Japanese, Korean, German, Arabic, Spanish, Italian, Portuguese, Simplified Chinese, Traditional Chinese				
Connection	USB Host, USB Device, RS-232, Pass/Fail out				
Display	5.7 inches color TFT-LCD				
Power	AC 100-240V, 47HZ-440HZ, 50VA Max				

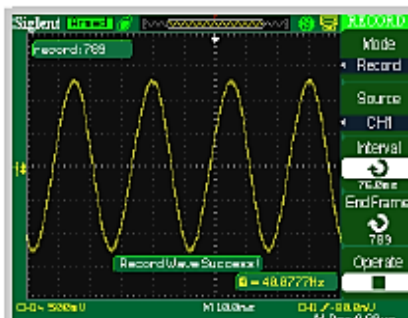
Remark: The differences between SDS1000C and SDS1000D series is that SDS1000D series has got 32K/CH depth memory, and it has data recorder function, while SDS1000C's depth memory is 4K/CH, and it doesn't have data recorder function. Most of other specifications are the same.

◆ Powerful function



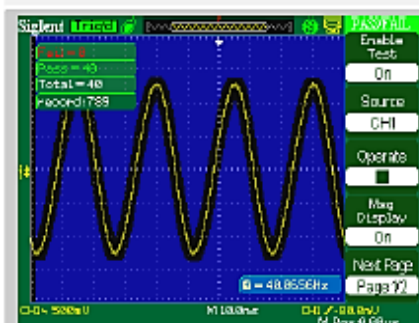
FFT split display

Can analyze signals in frequency area, channel waveform and FFT waveform display at the same time in one screen, making the waveform observed directly



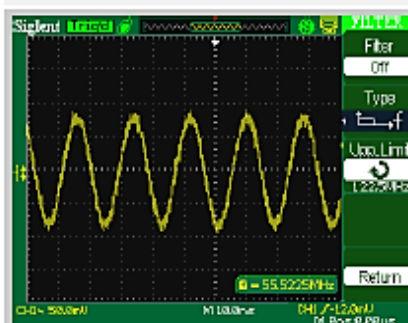
Waveform save/recall

The biggest waveform recording is up to 2500frame, channel waveform and Pass/Fail waveform can be recorded too, and capture better waveform analysis effect through recall function

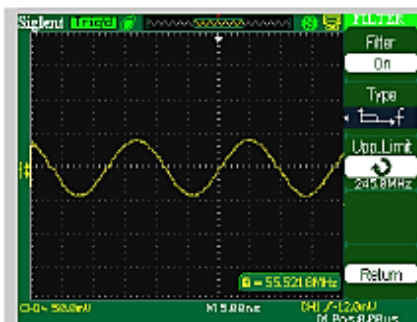


PASS/FAIL

Make signal mode for testing item of production line, improve testing efficiency, possess the control function of stopping while outputting



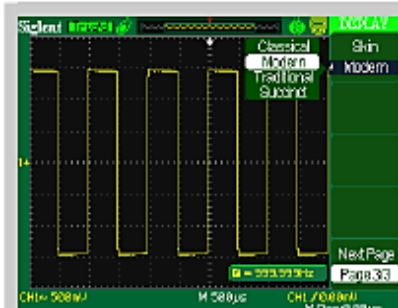
Turn off digital filter function



Turn on digital filter function

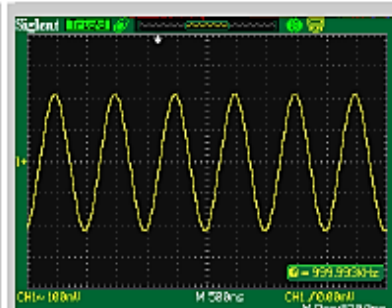
Precise digital filter function; can be set frequency range to reduce signal noise or filter error signal, enable users observe their signals more clearly, and work more efficiently.

◆ Humanized design



Multi-style display

Classic, modern, tradition, and concision four different display styles meet customers' different needs; relieve tiredness after long time working.



Pop-up menu

Pop-up menu display mode, hide menu at any time, 12 grid full-screen display, enable users observe the waveform more clearly



On-line help

One-key on-line help system works all time, facilitating the learning and application.



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Brief introduction of Siglent:

Founded in 2002, Siglent has dedicated to DSO developing and manufacturing for more than 8 years. Now it has grown to be the largest DSO manufacturer in the world, for it produces more than 50,000 units a year, does ODM for more than 20 brands in worldwide. Because of profession, we have developed four series of DSO for overseas market. Now we are developing more and more instruments to meet customer's needs, like signal generator, handheld oscilloscope, 500MHz-1G BW digital scope, etc. We are making effort to offer better and better products to our customers.

R&D capability:

Siglent's R&D outlay possesses more than 20% of turnover of each year. With more than 50 high degree and experienced engineers, and those from world-renowned companies' managers, Siglent has got plenty experience in research and management.

Production system:

In 2005, Siglent built a modernized automation factory of digital storage oscilloscope with an area of more than 2000 m², building up a standardized supplying-chain system at the same time. Since most of our suppliers are world renowned companies, our material is absolutely reliable and quality ensured. Therefore, our oscilloscopes are of high quality and high reliability.

Siglent's history:

- 2002, Siglent started to research DSO;
- October 2005, Siglent brought out SDS7000 Series with 1G sampling rate and many other original technologies such as EasyHunting, which makes our DSO 2-4 times more precise than other same level products.
- 2007, SDS7000 Series are sold in large-scale in USA and west Europe, the annual sales is more than 10000 pcs.
- January 2007, Siglent developed SDS2000 Series, applying USB flash saving technology and PicBridge printing technology in DSO with 1G sampling rate for the first time.
- March 2008, Siglent gave birth to SDS1000C Series.
- September 2008, Siglent got UL certification.
- December 2009, Siglent developed SDS1000CM series with 2M depth memory.
- December 2010, Siglent brought out 7" 4CH DSO with 2G sample rate.