US007287583B2

(12) United States Patent
Iwanami et al.

(10) **Patent No.: US 7,287,583 B2**

(45) **Date of Patent: Oct. 30, 2007**

(54) **AIR-CONDITIONING APPARATUS INCLUDING MOTOR-DRIVEN COMPRESSOR FOR IDLE STOPPING VEHICLES**

(75) Inventors: **Shigeki Iwanami**, Okazaki (JP); **Yasushi Suzuki**, Chiryu (JP); **Yasushi Yamanaka**, Nakashima-gun (JP); **Yukio Ogawa**, Kariya (JP); **Keiichi Uno**, Kariya (JP)

(73) Assignee: **DENSO CORPORATION**, Kariya (JP)

(*) Notice: Subject to any disclaimer, the term of this patent is extended or adjusted under 35 U.S.C. 154(b) by 431 days.

(21) Appl. No.: **11/029,404**

(22) Filed: **Jan. 6, 2005**

(65) **Prior Publication Data**

US 2005/0109499 A1 May 26, 2005

Related U.S. Application Data

(62) Division of application No. 10/132,764, filed on Apr. 26, 2002, now Pat. No. 6,981,544.

(30) **Foreign Application Priority Data**

Apr. 27, 2001	(JP)	....	2001-131605
May 30, 2001	(JP)	....	2001-161921
Jul. 6, 2001	(JP)	....	2001-206890
Oct. 19, 2001	(JP)	....	2001-322607
Nov. 9, 2001	(JP)	....	2001-345038
Jan. 31, 2002	(JP)	....	2002-22723

(51) **Int. Cl.**
B60H 1/00 (2006.01)

(52) **U.S. Cl.** **165/202**; 165/271; 165/244; 62/133; 62/134; 62/244

(58) **Field of Classification Search** 165/202, 165/244, 253, 254, 271, 278, 299, 42, 43, 165/44; 62/133, 134, 229, 239, 244
See application file for complete search history.

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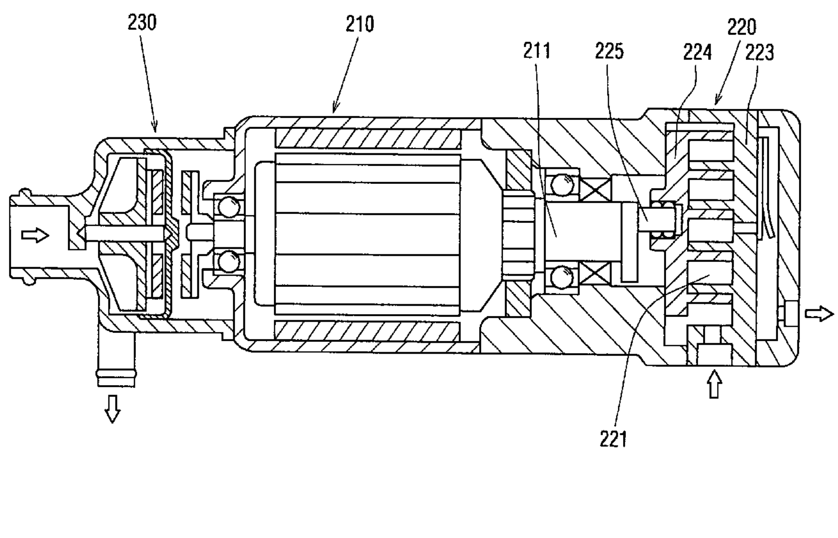
Primary Examiner—Ljiljana Ciric
(74) *Attorney, Agent, or Firm*—Posz Law Group, PLC

(57) **ABSTRACT**

A low-cost vehicle air-conditioning apparatus for idle stopping vehicles is capable of performing both cooling and heating operations throughout the year. The air-conditioning apparatus includes an engine-driven compressor and engine-driven pump for a heating unit. The air-conditioning apparatus includes a motor-driven compressor and pump. A control unit drives the motor such that the motor-driven compressor is operated when there is a need for cooling and the motor-driven pump is operated when there is a need for heating when the engine is stopped. Battery power is conserved through various methods.

13 Claims, 45 Drawing Sheets



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JP	A-2000-127753	5/2000

* cited by examiner

FIG. 1

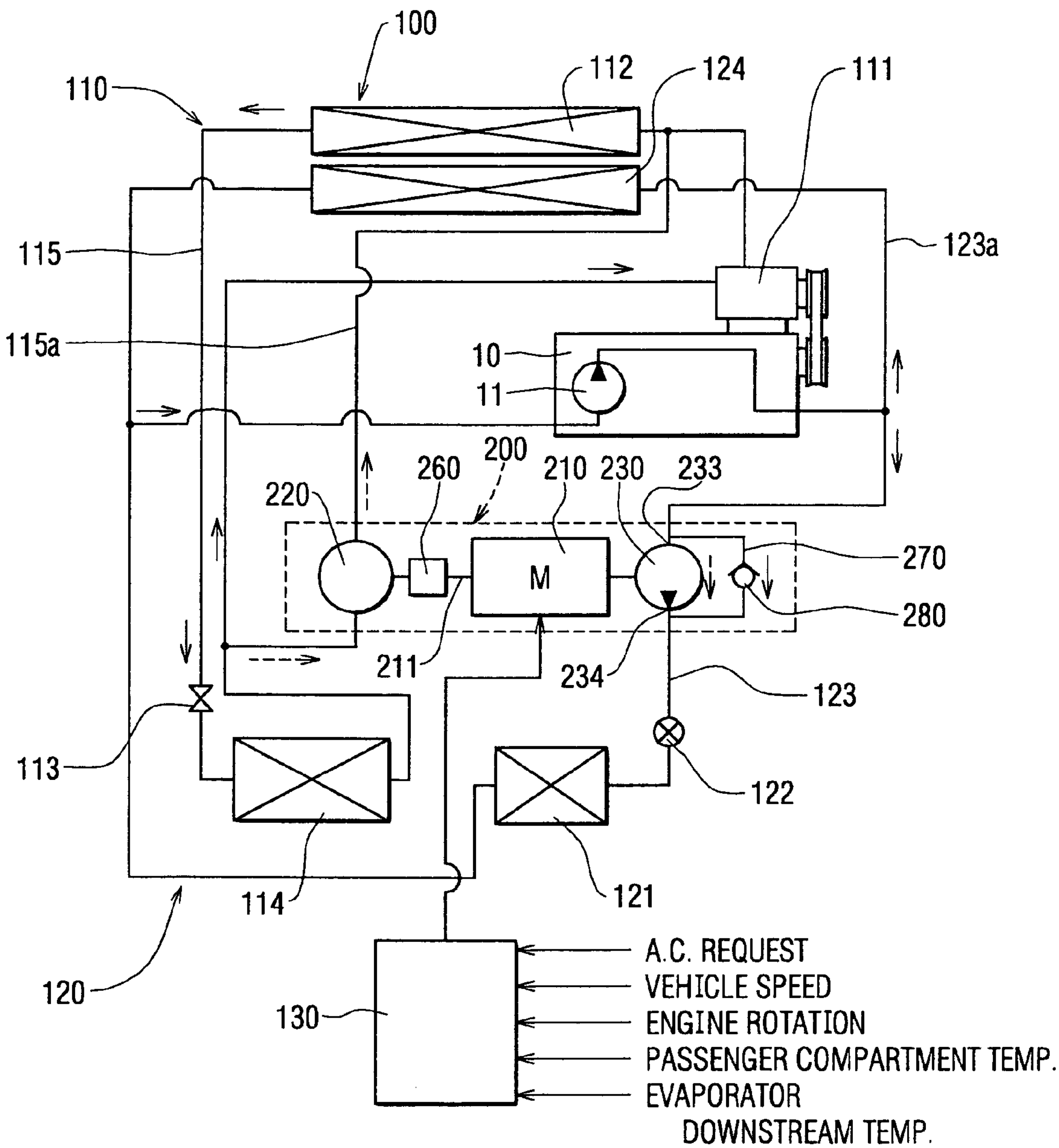

100
110
112
124
111
115
123a
115a
10
11
200
260
210
230
233
220
270
280
211
234
123
113
122
114
121
120
130
M
A.C. REQUEST
VEHICLE SPEED
ENGINE ROTATION
PASSENGER COMPARTMENT TEMP.
EVAPORATOR
DOWNSTREAM TEMP.

FIG. 2

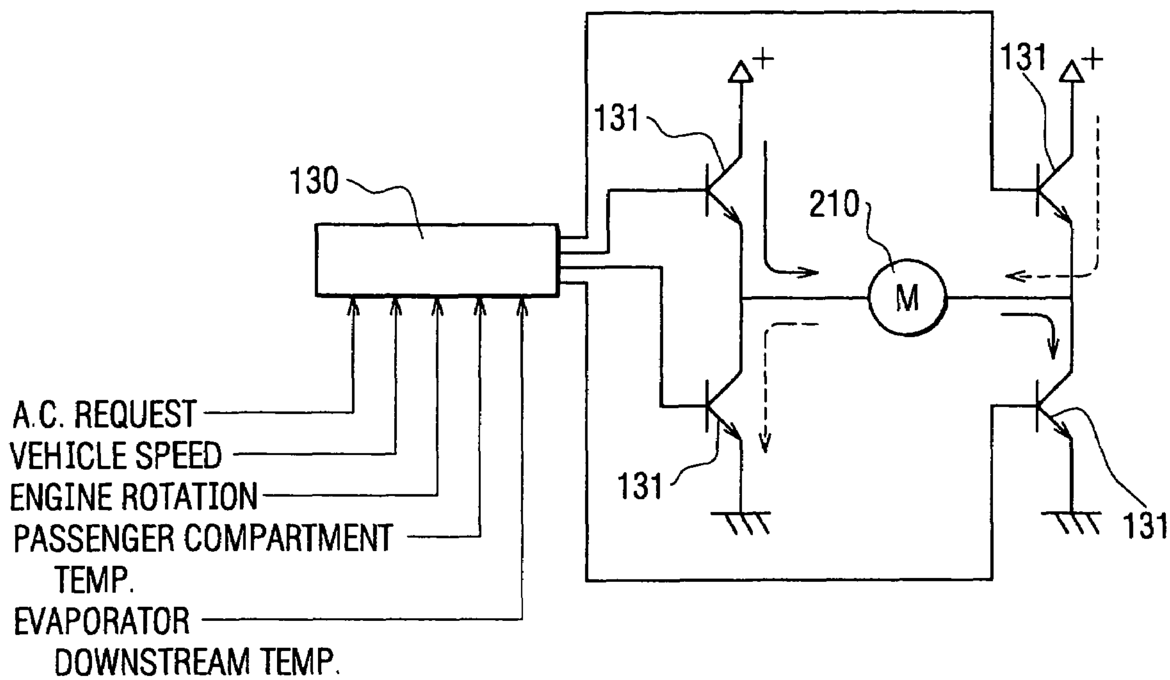

130
131
131
131
131
210
M
+
+
A.C. REQUEST
VEHICLE SPEED
ENGINE ROTATION
PASSENGER COMPARTMENT TEMP.
EVAPORATOR DOWNSTREAM TEMP.

FIG. 3

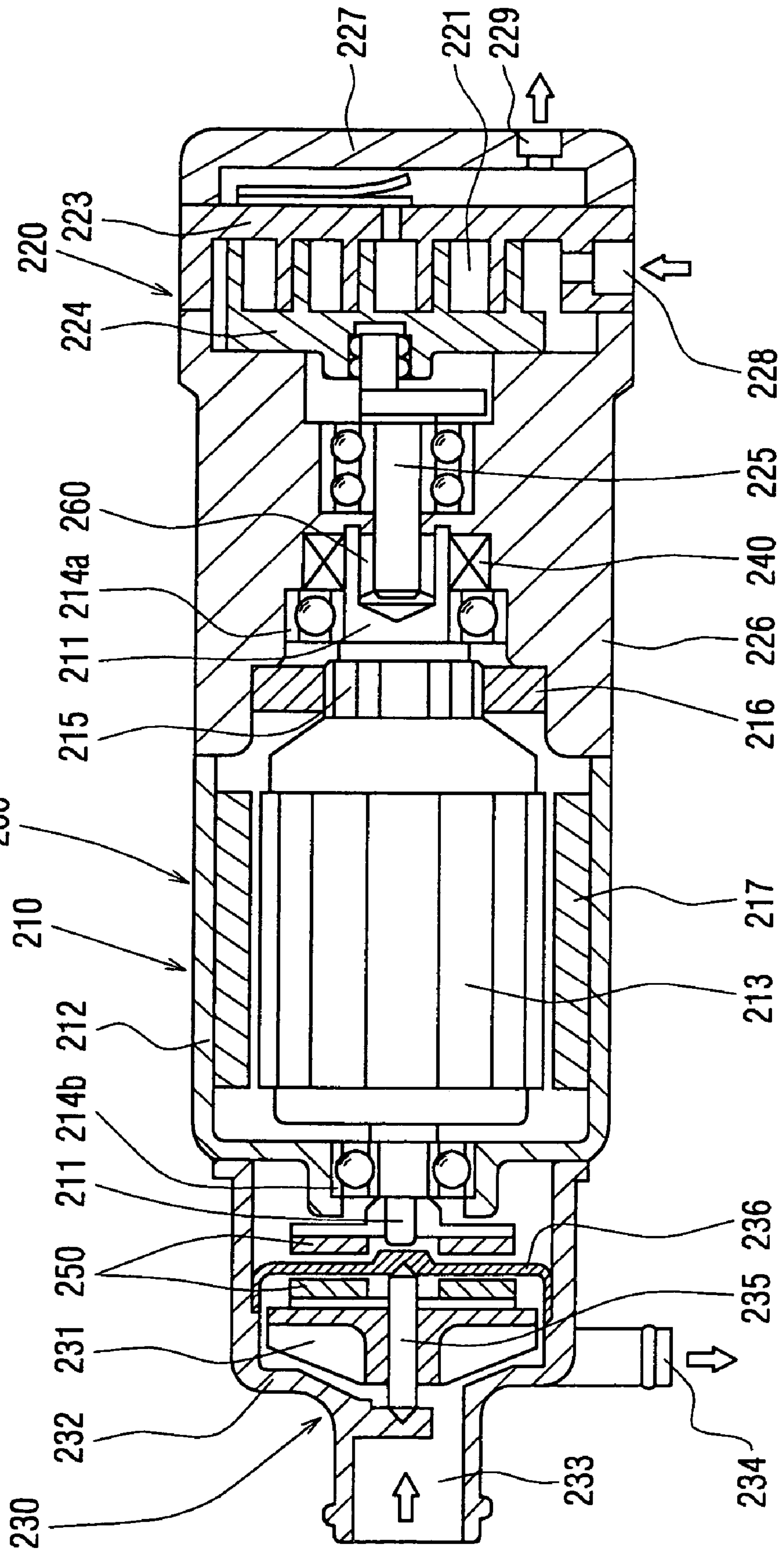

227
221
229
220
223
224
228
225
260
214a
240
211
226
215
216
200
217
210
213
212
214b
211
236
250
235
231
232
234
233
230

FIG. 4

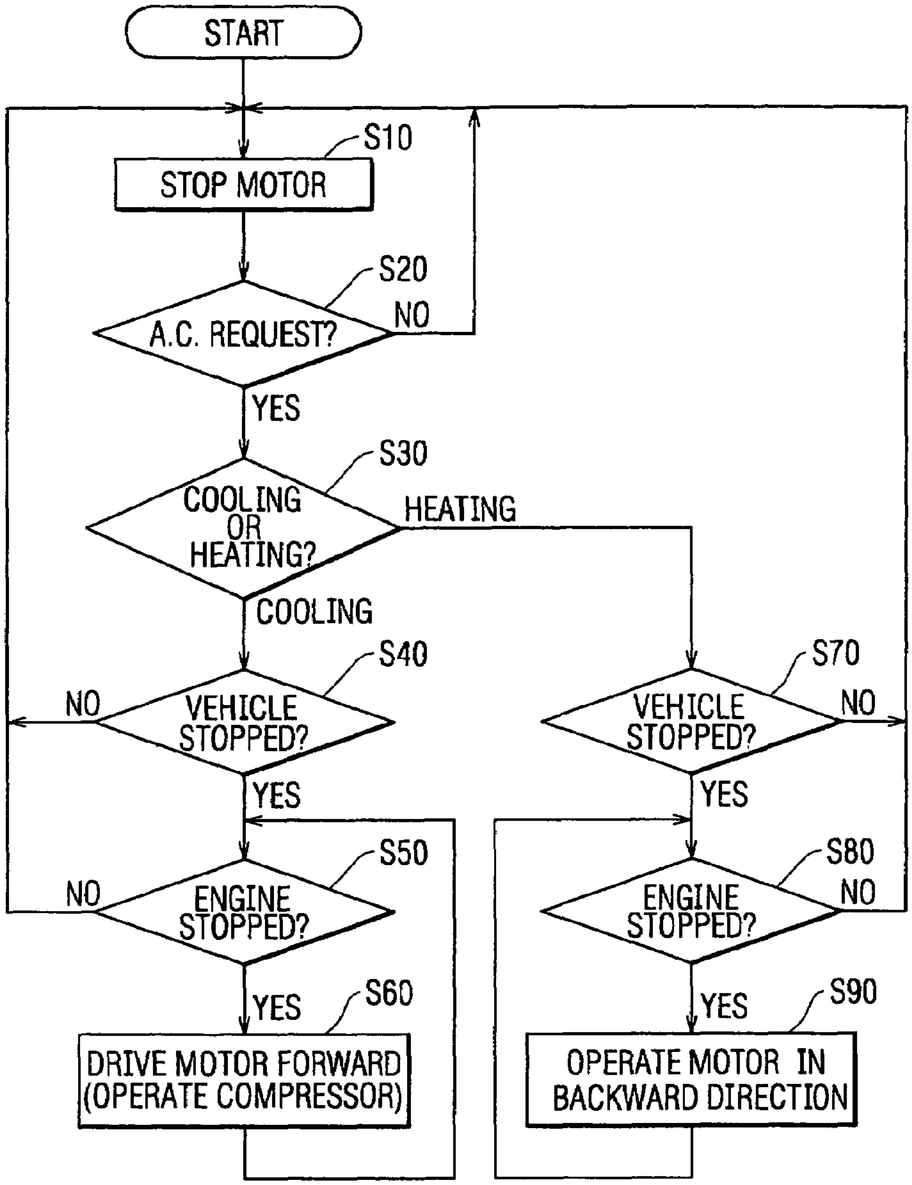

START
S10
STOP MOTOR
S20
A.C. REQUEST?
NO
YES
S30
COOLING OR HEATING?
HEATING
COOLING
S40
VEHICLE STOPPED?
NO
YES
S50
ENGINE STOPPED?
NO
YES
S60
DRIVE MOTOR FORWARD (OPERATE COMPRESSOR)
S70
VEHICLE STOPPED?
NO
YES
S80
ENGINE STOPPED?
NO
YES
S90
OPERATE MOTOR IN BACKWARD DIRECTION

FIG. 5

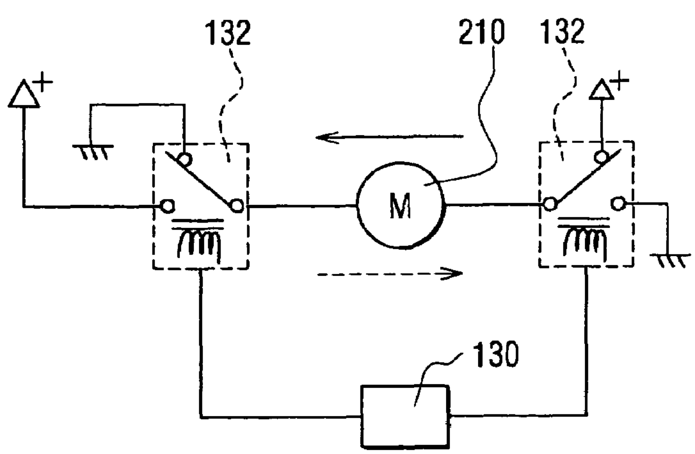

132
210
132
M
130

FIG. 6

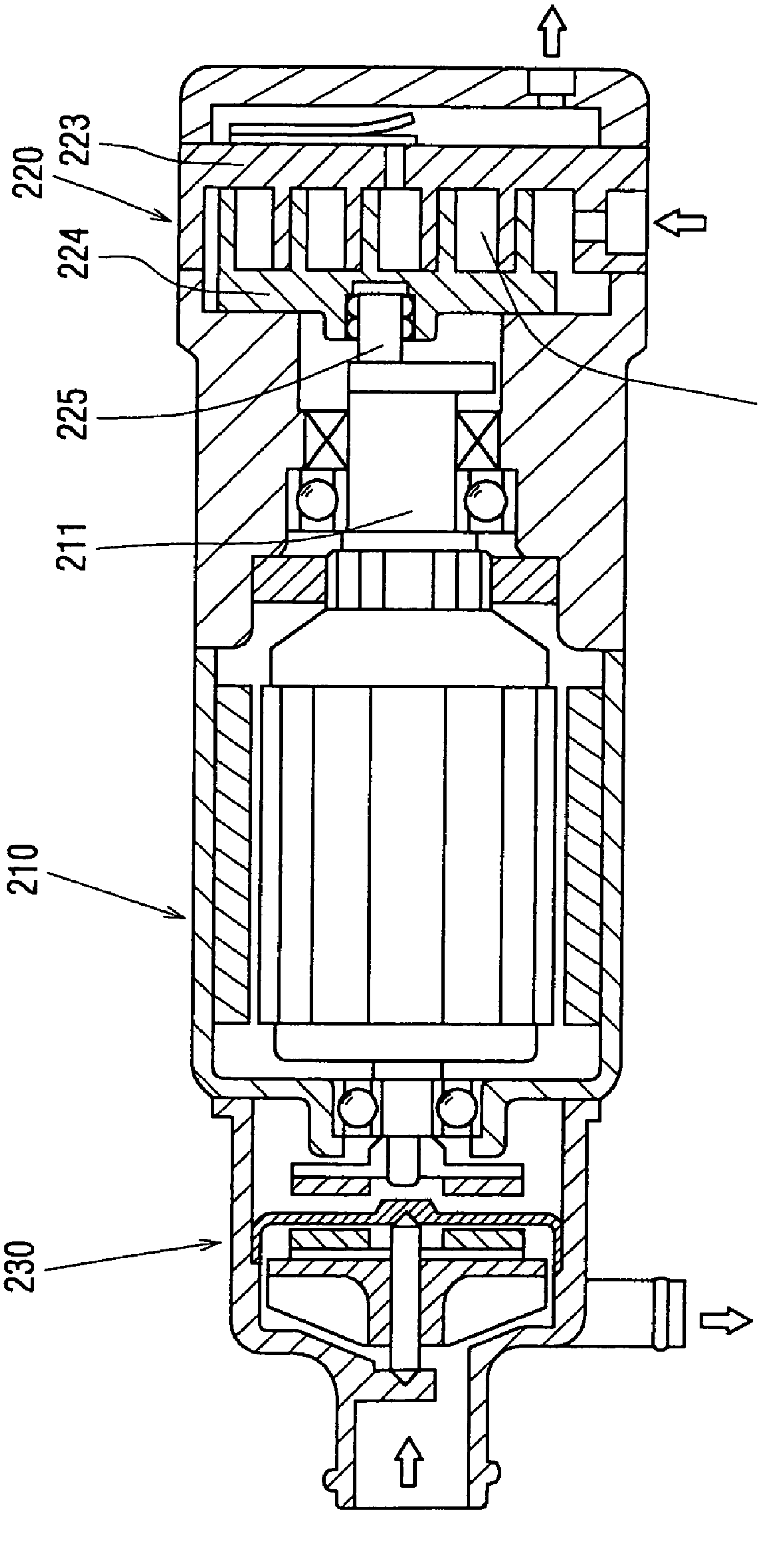


FIG. 7A

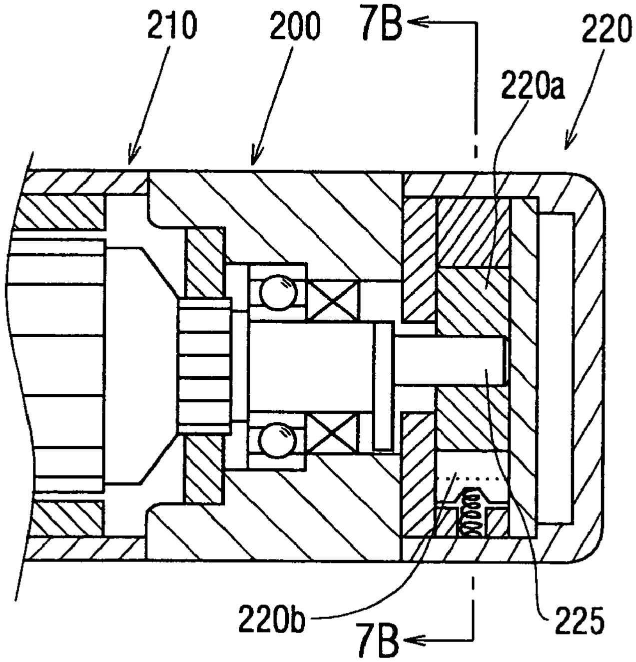


FIG. 7B

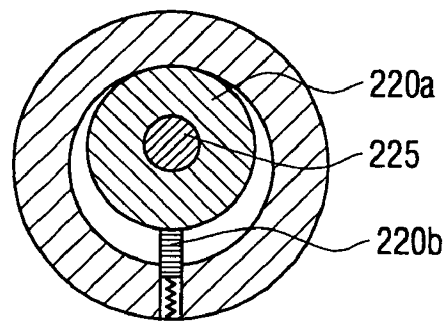


FIG. 8A

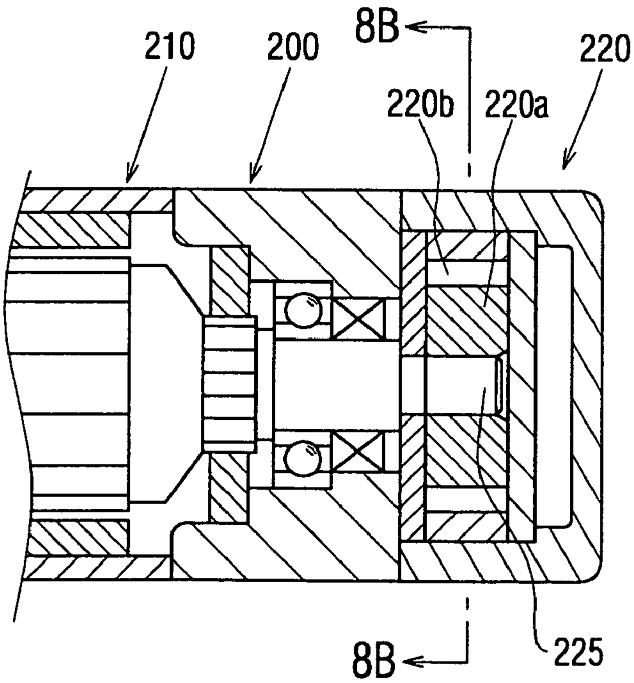


FIG. 8B

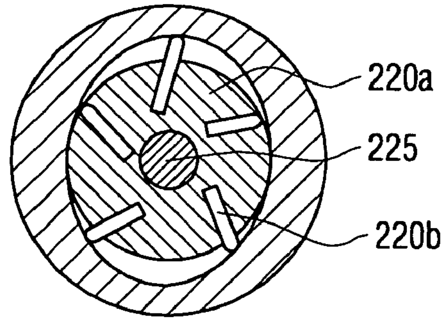


FIG. 9

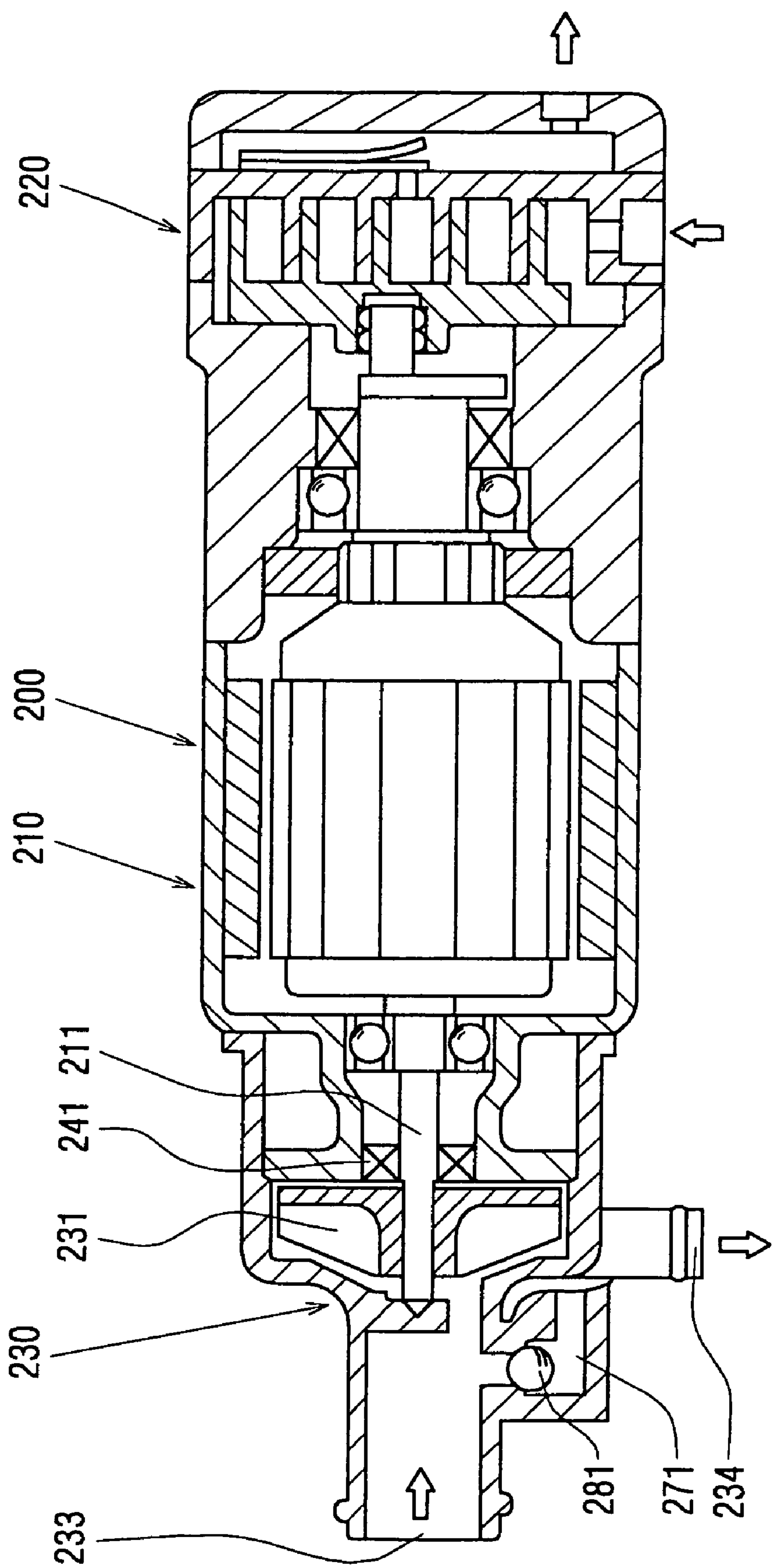


FIG. 10A

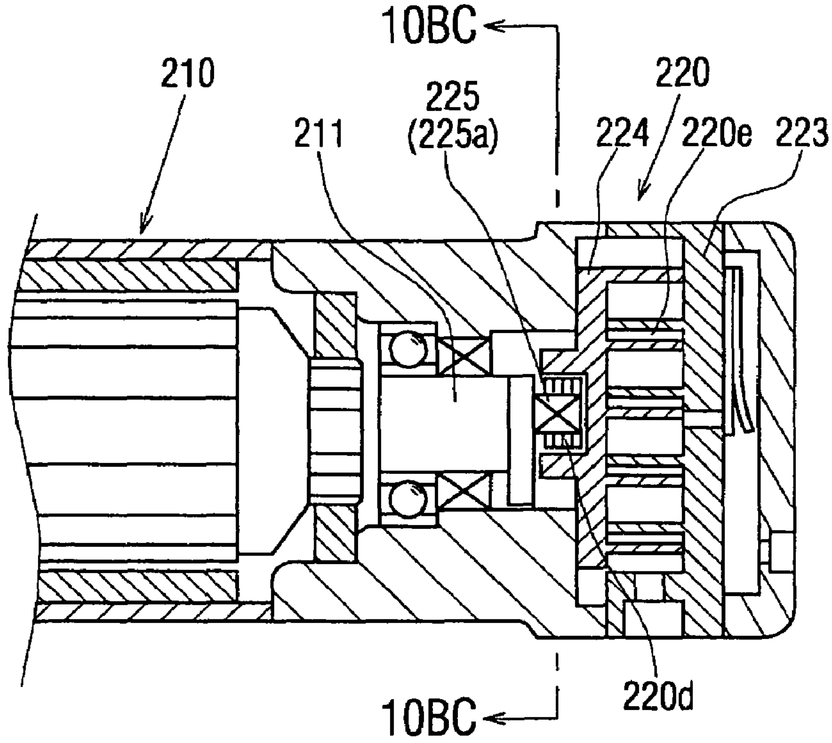


FIG. 10B

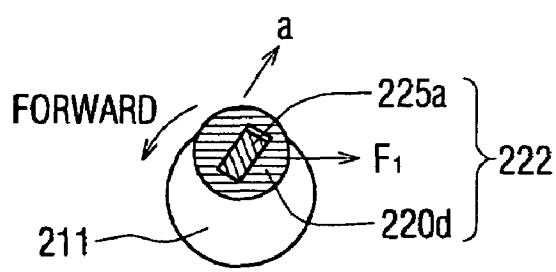


FIG. 10C

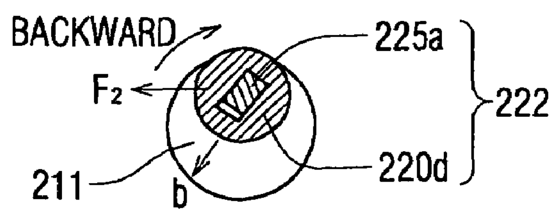


FIG. 11

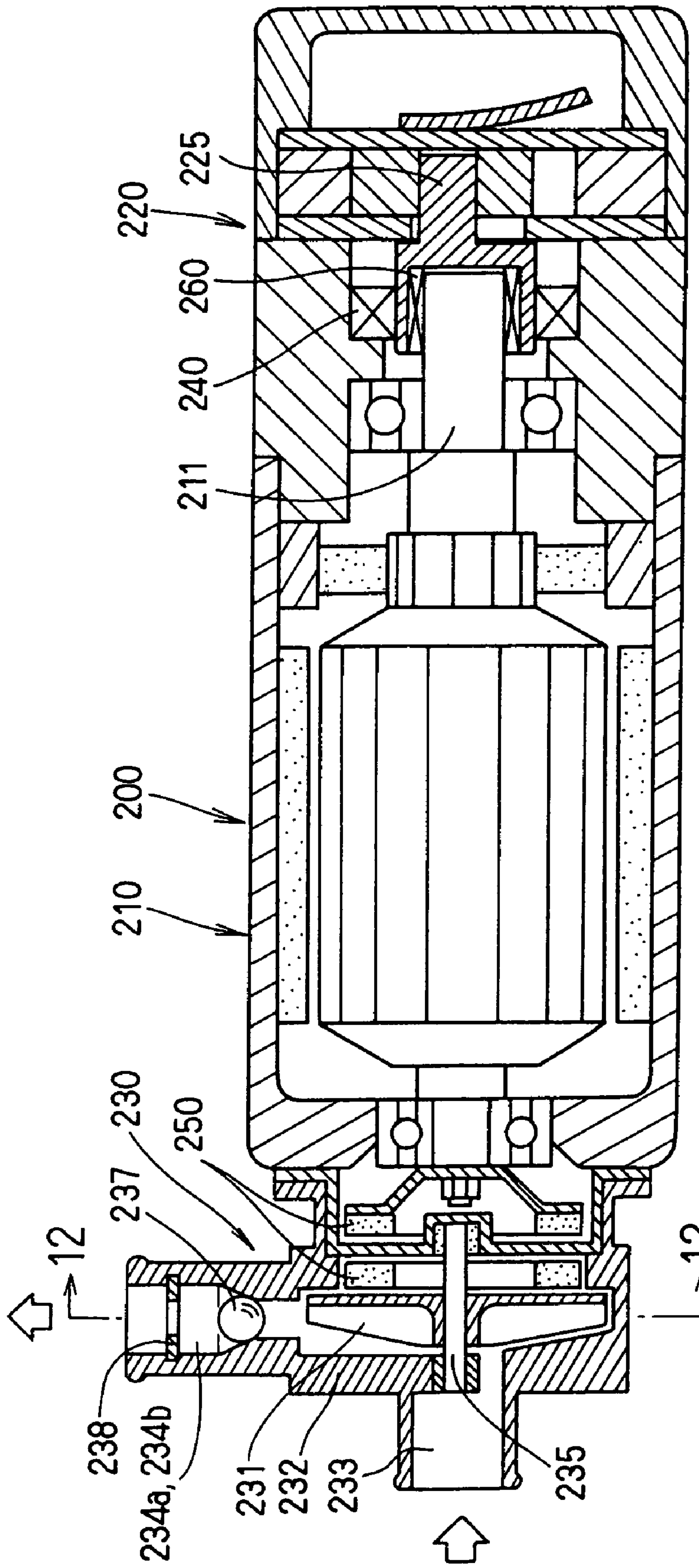

225
220
260
240
211
200
210
230
250
237
12
12
238
234a, 234b
231
232
233
235

FIG. 12

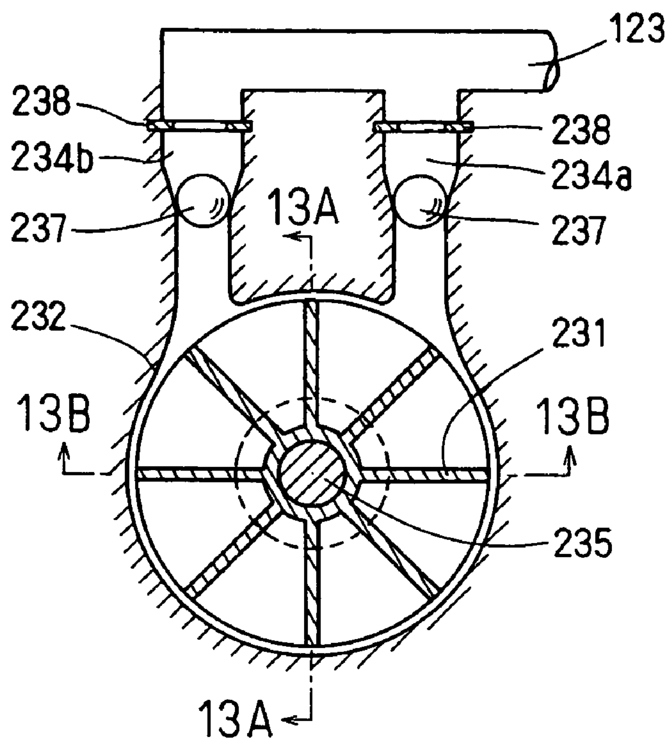


FIG. 13A

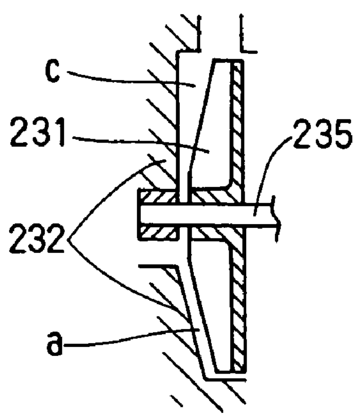


FIG. 13B

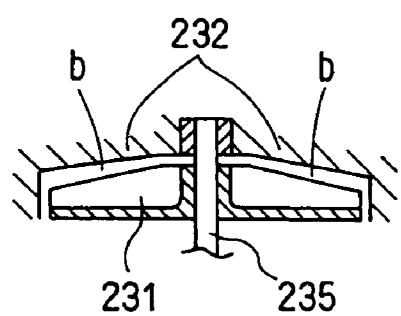


FIG. 14A

FIG. 14B

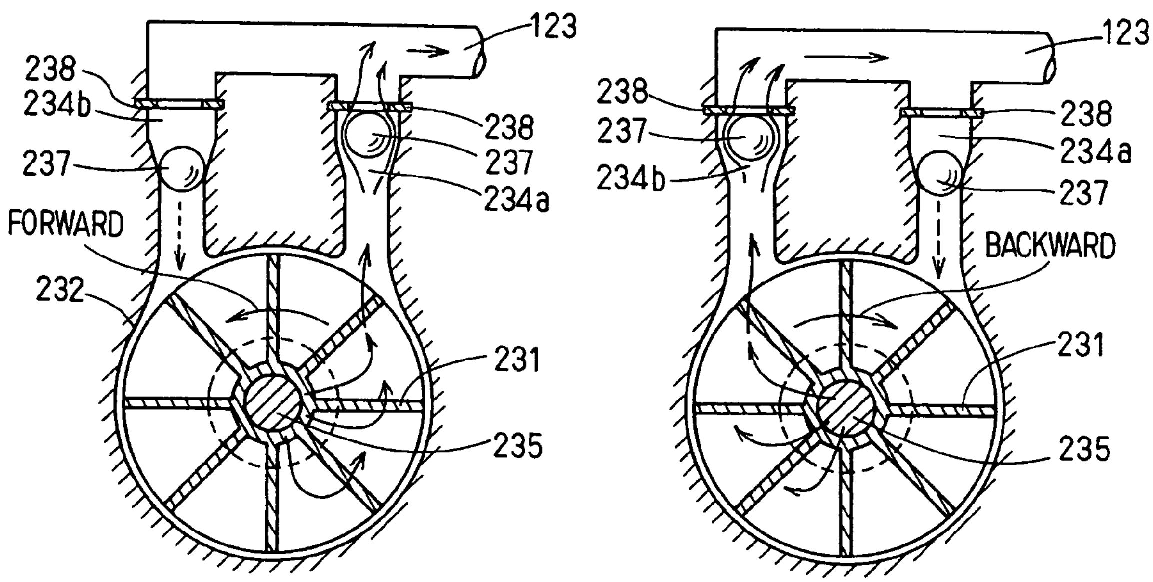


FIG. 15

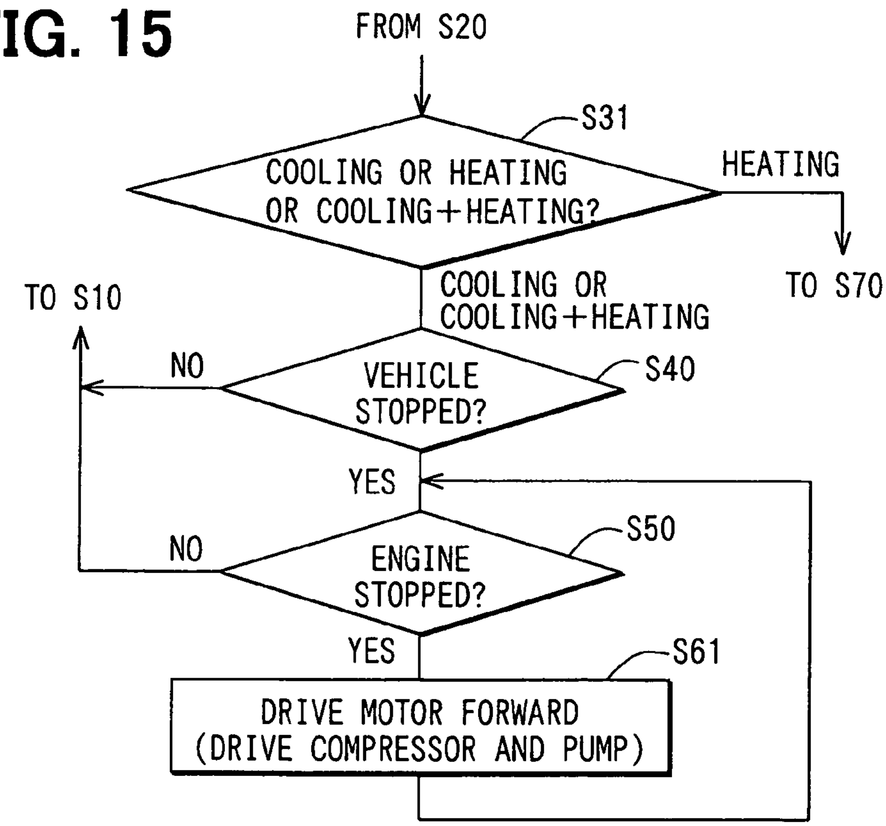


FIG. 16

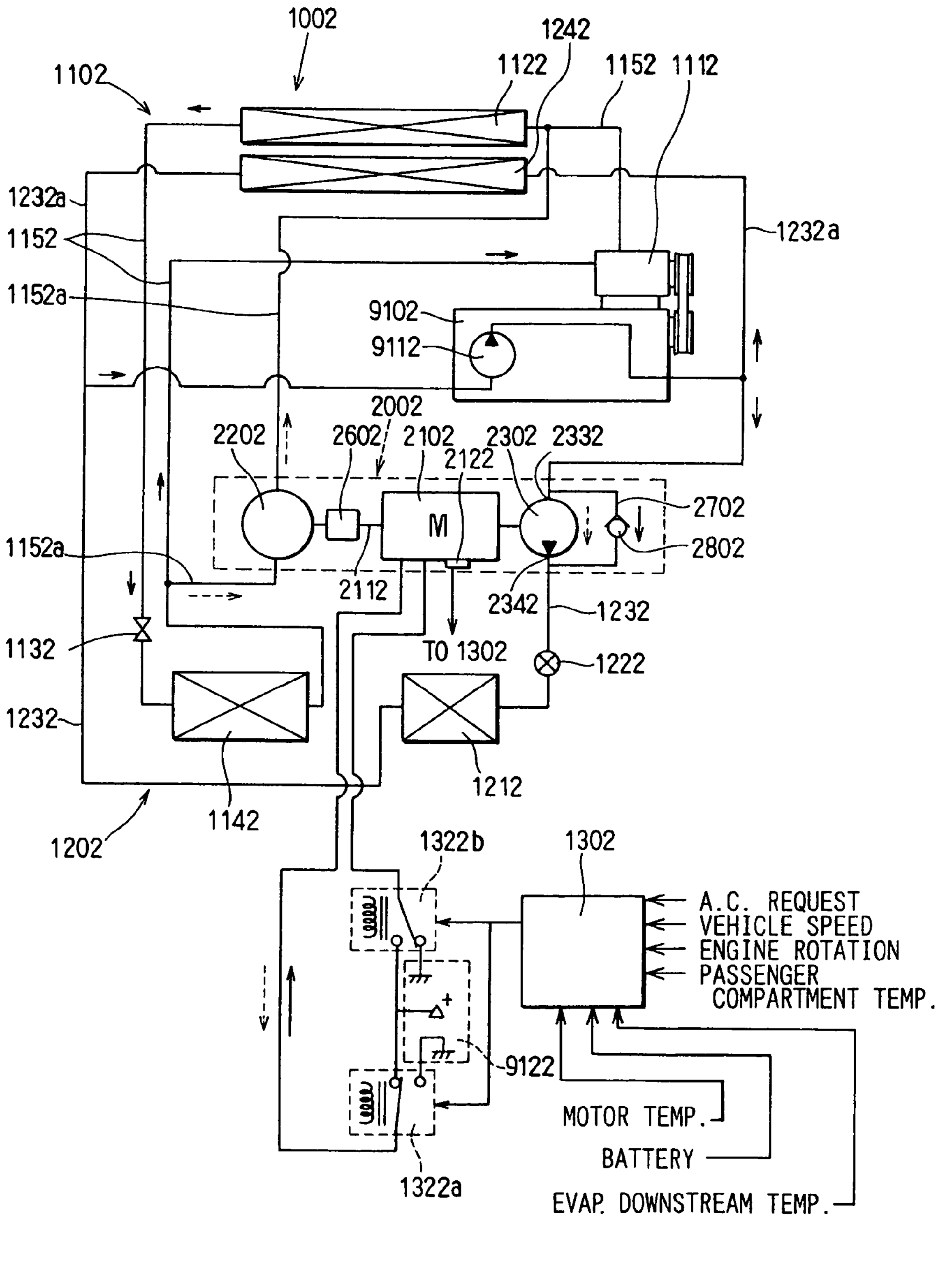

1002
1242
1102
1122
1152
1112
1232a
1152
1152a
9102
9112
2002
2202
2602
2102
2302
2332
2122
M
2702
1152a
2802
2112
2342
1232
1132
TO 1302
1222
1232
1142
1212
1202
1322b
1302
A.C. REQUEST
VEHICLE SPEED
ENGINE ROTATION
PASSENGER
COMPARTMENT TEMP.
9122
MOTOR TEMP.
BATTERY
1322a
EVAP. DOWNSTREAM TEMP.

FIG. 17

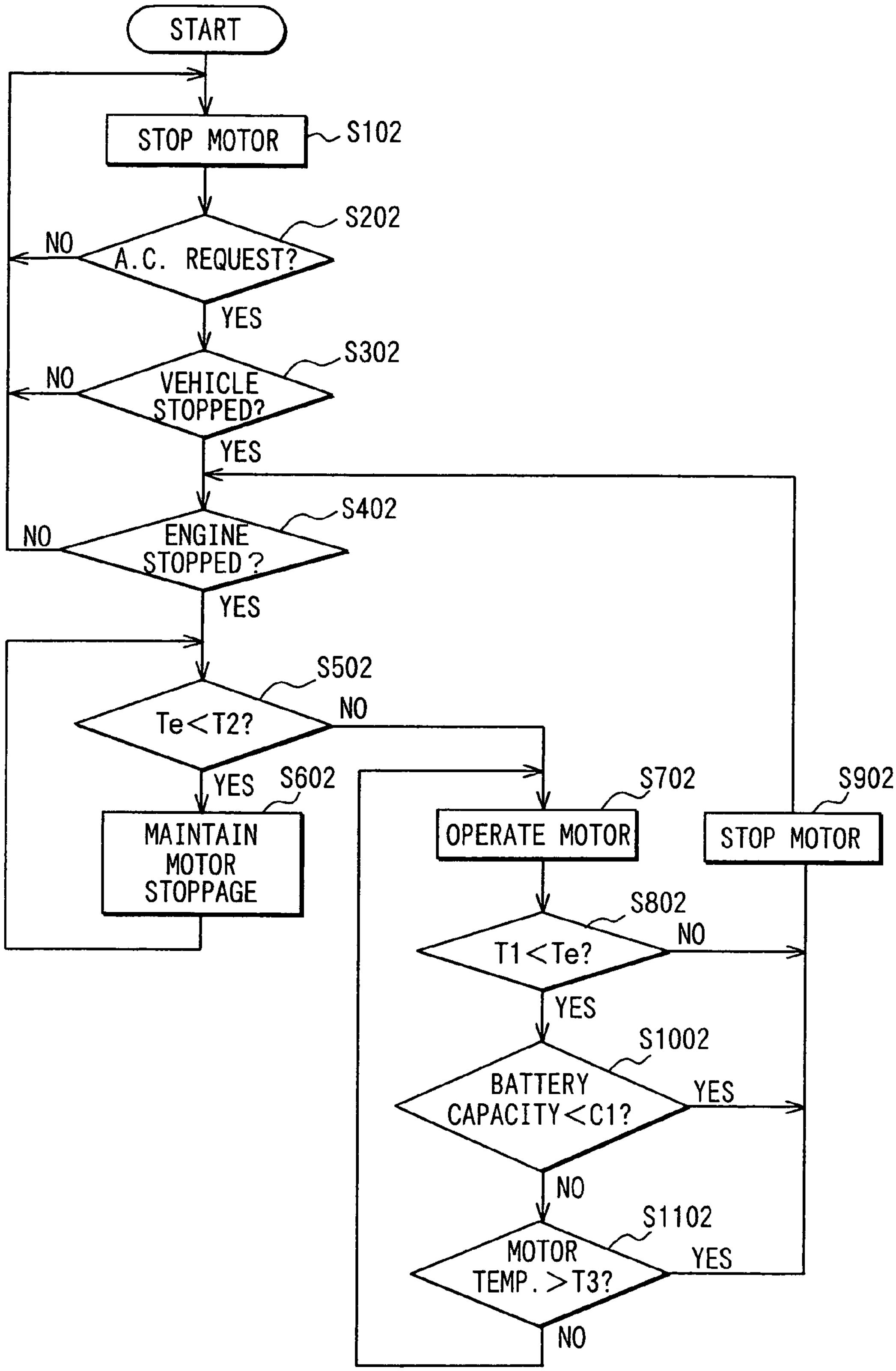

START
STOP MOTOR
S102
A.C. REQUEST?
S202
NO
YES
VEHICLE STOPPED?
S302
NO
YES
ENGINE STOPPED ?
S402
NO
YES
Te<T2?
S502
NO
YES
MAINTAIN MOTOR STOPPAGE
S602
OPERATE MOTOR
S702
STOP MOTOR
S902
T1<Te?
S802
NO
YES
BATTERY CAPACITY<C1?
S1002
YES
NO
MOTOR TEMP.>T3?
S1102
YES
NO

FIG. 18A

FIG. 18B

FIG. 18C

FIG. 18D

FIG. 18E

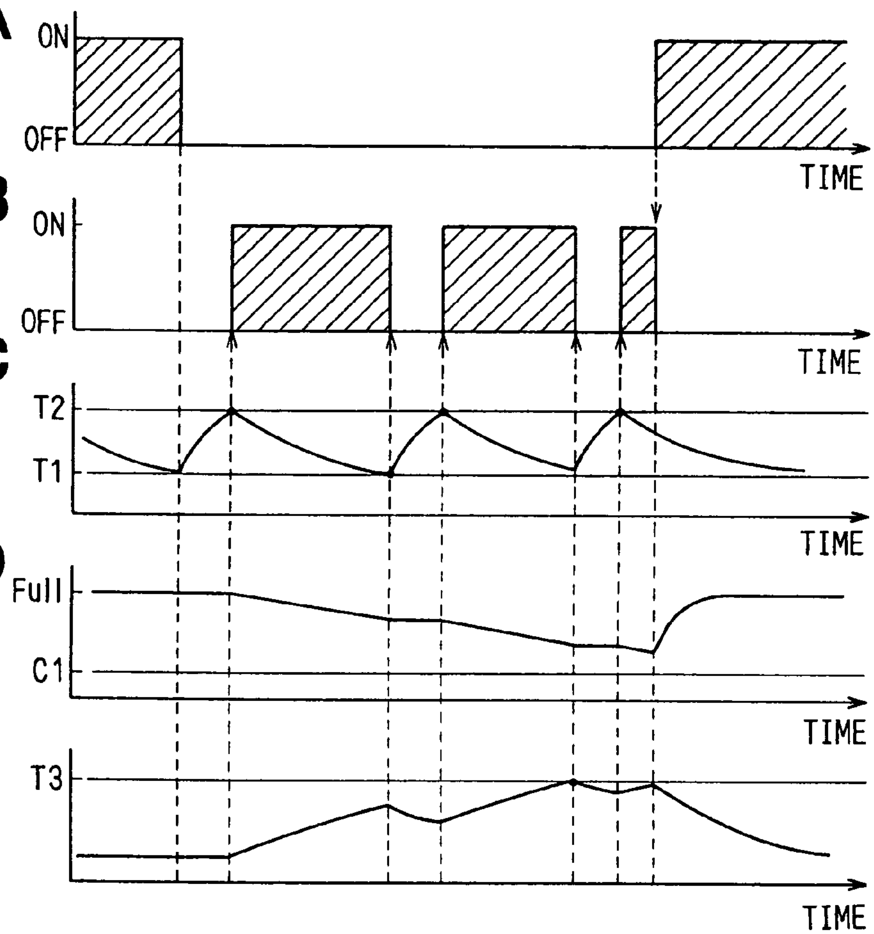


FIG. 19

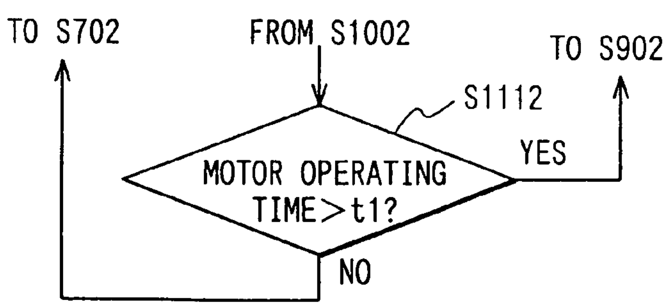


FIG. 20

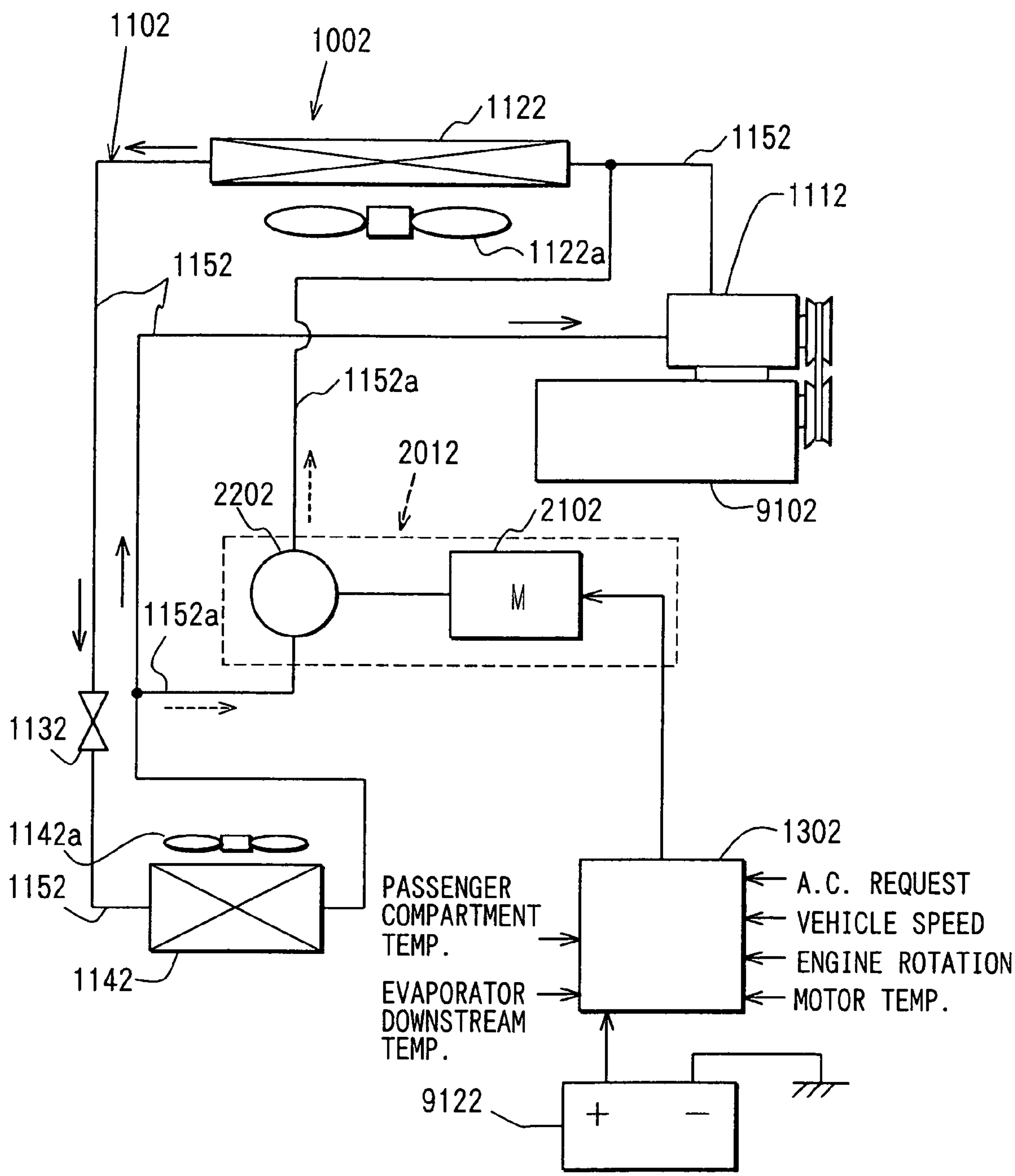

1102
1002
1122
1152
1112
1122a
1152
1152a
2012
2202
2102
9102
M
1152a
1132
1142a
1152
1142
1302
A.C. REQUEST
VEHICLE SPEED
ENGINE ROTATION
MOTOR TEMP.
PASSENGER COMPARTMENT TEMP.
EVAPORATOR DOWNSTREAM TEMP.
9122
+
−

FIG. 21

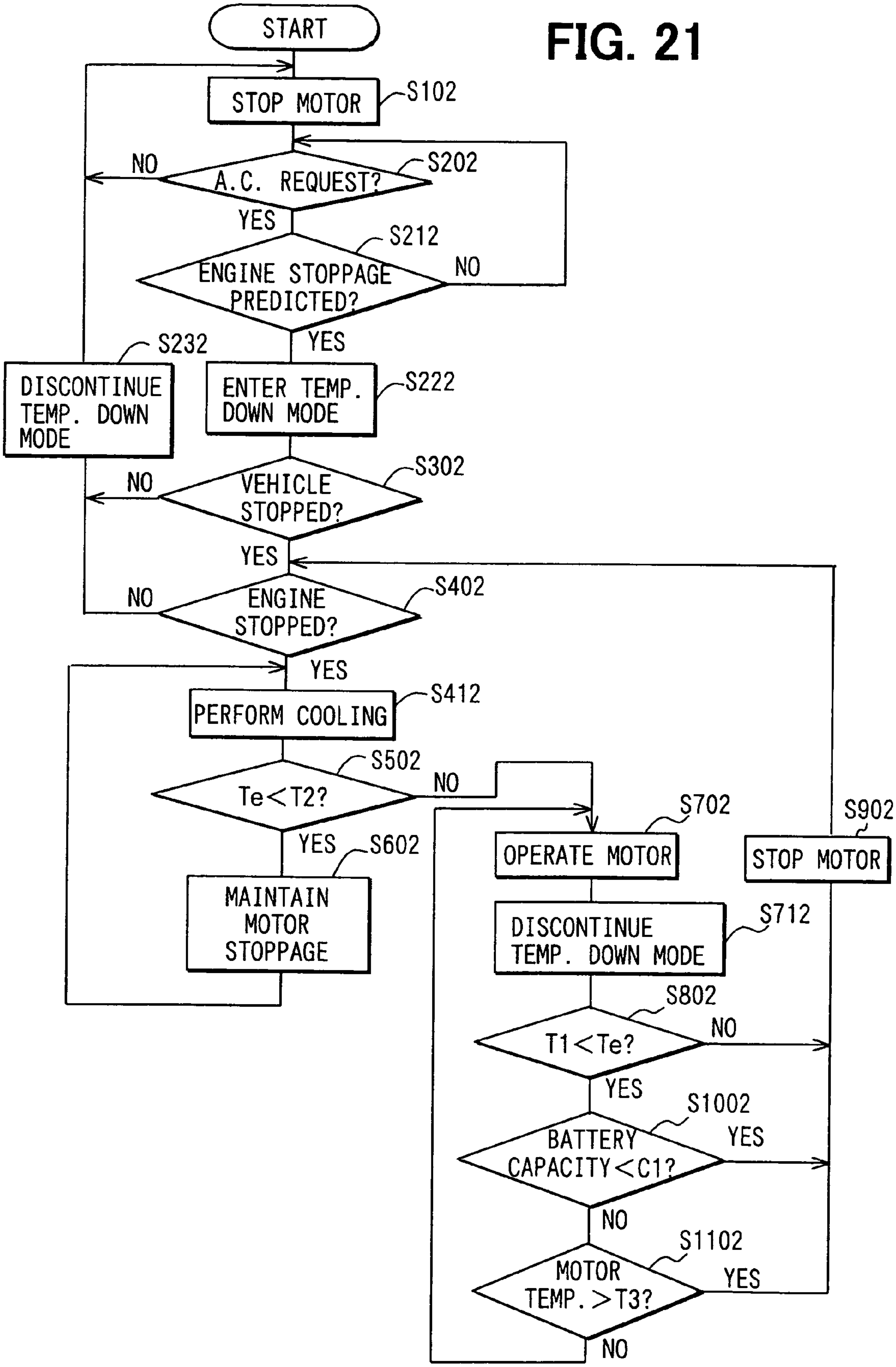

START
STOP MOTOR
S102
A.C. REQUEST?
S202
NO
YES
ENGINE STOPPAGE PREDICTED?
S212
NO
YES
S232
DISCONTINUE TEMP. DOWN MODE
ENTER TEMP. DOWN MODE
S222
VEHICLE STOPPED?
S302
NO
YES
ENGINE STOPPED?
S402
NO
YES
PERFORM COOLING
S412
Te<T2?
S502
NO
YES
S602
MAINTAIN MOTOR STOPPAGE
OPERATE MOTOR
S702
STOP MOTOR
S902
DISCONTINUE TEMP. DOWN MODE
S712
T1<Te?
S802
NO
YES
BATTERY CAPACITY<C1?
S1002
YES
NO
MOTOR TEMP.>T3?
S1102
YES
NO

FIG. 22A

VEHICLE SPEED

FIG. 22B

FLOW RATE OF EVAPORATOR FAN

FIG. 22C

EVAPORATOR DOWNSTREAM TEMP.

FIG. 22D

LEVEL OF LIQUID IN CONDENSER

FIG. 22E

MOTOR

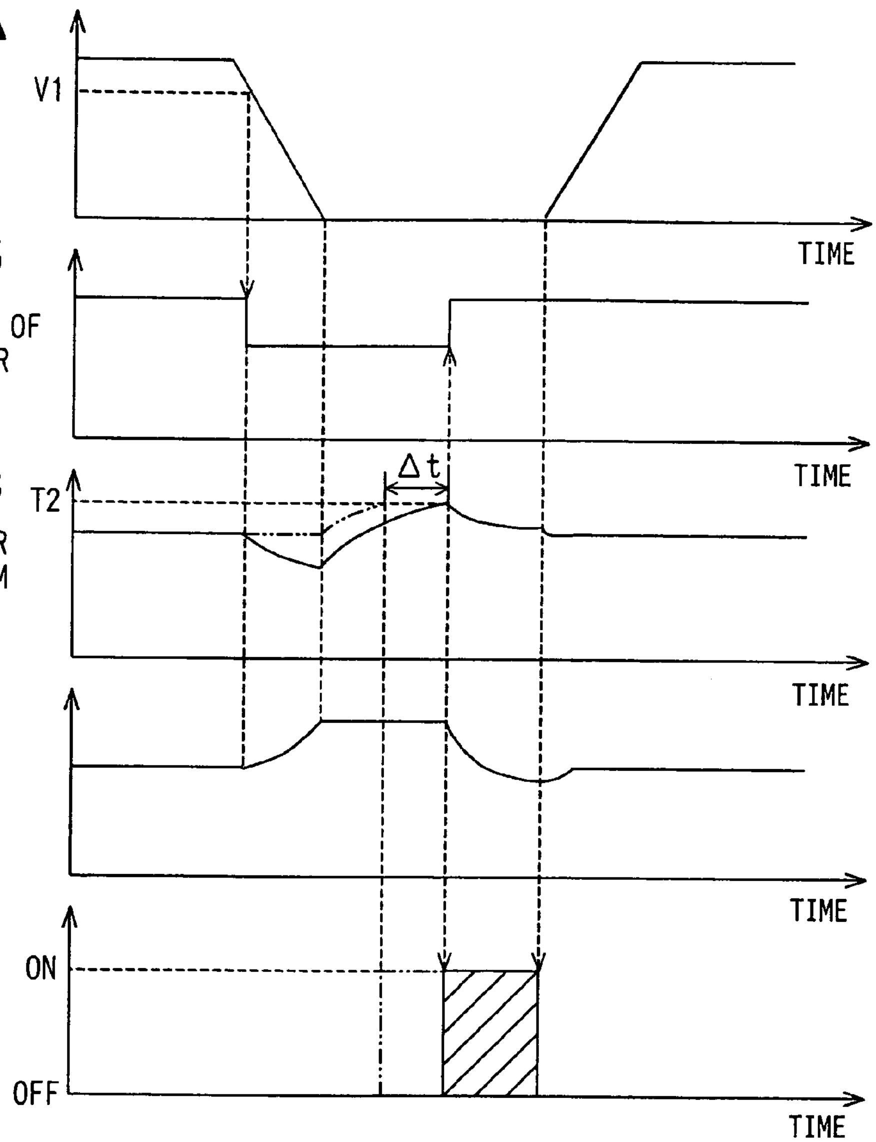


FIG. 23A

FIG. 23B

FIG. 23C

FIG. 23D

FIG. 23E

FIG. 23F

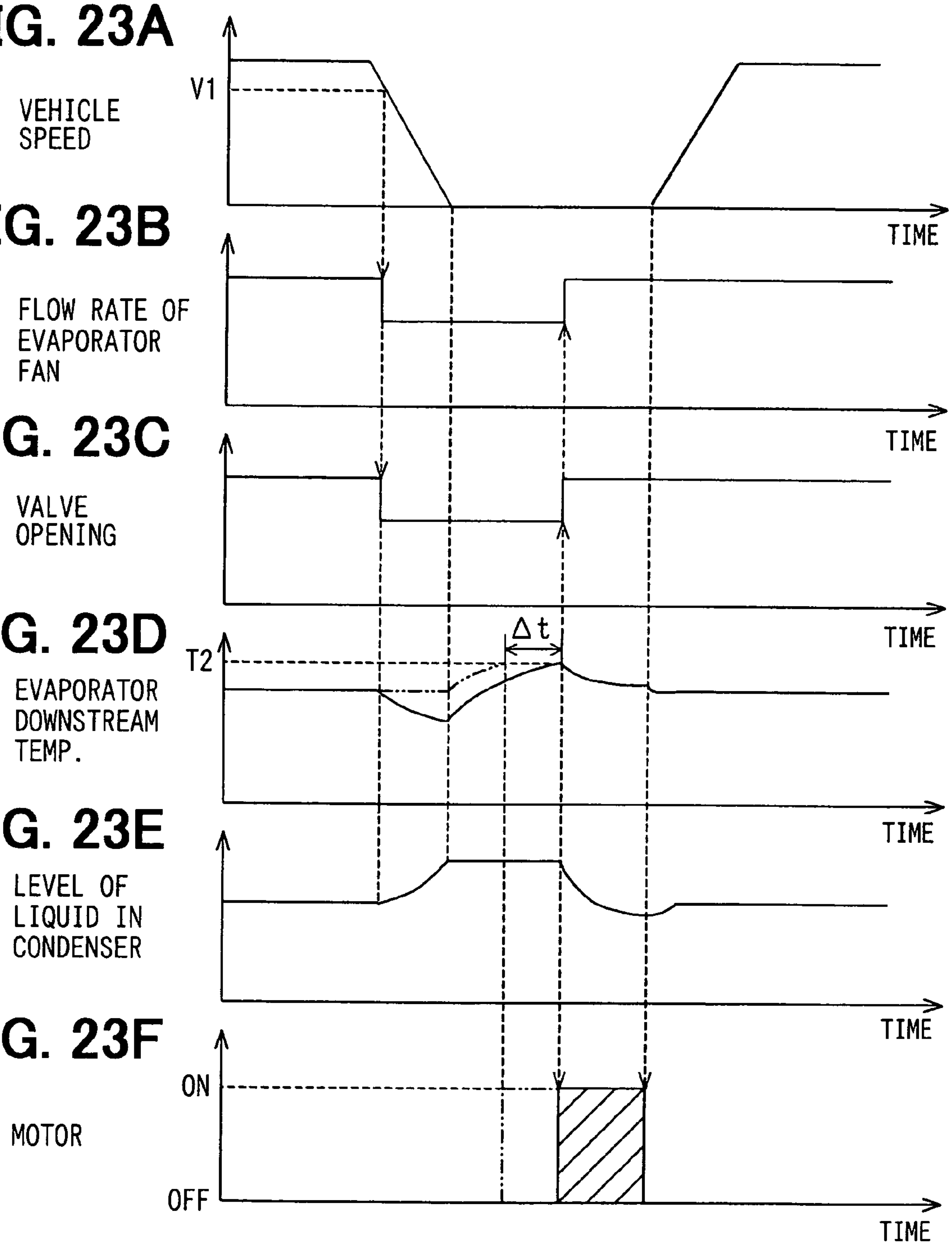


FIG. 24A

FIG. 24B

FIG. 24C

FIG. 24D

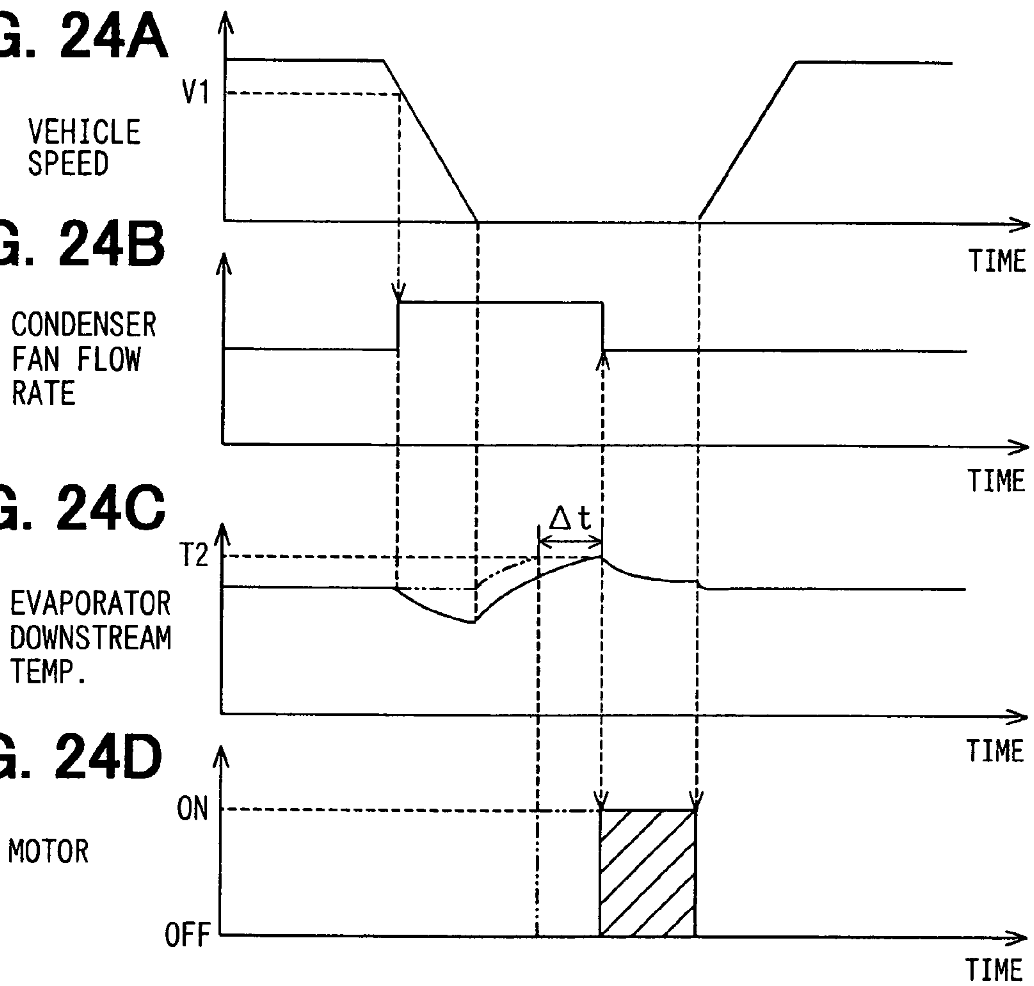


FIG. 25A

FIG. 25B

FIG. 25C

FIG. 25D

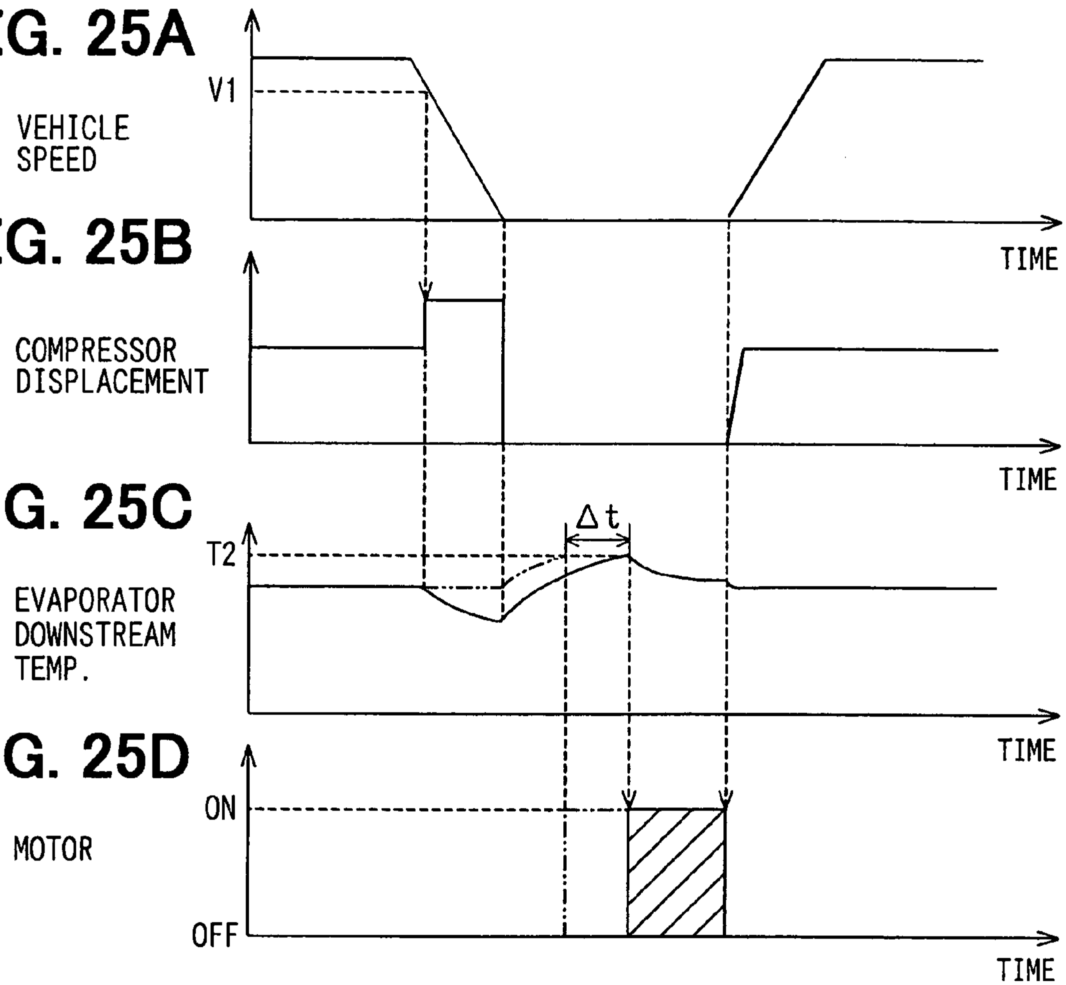


FIG. 26A

FIG. 26B

FIG. 26C

FIG. 26D

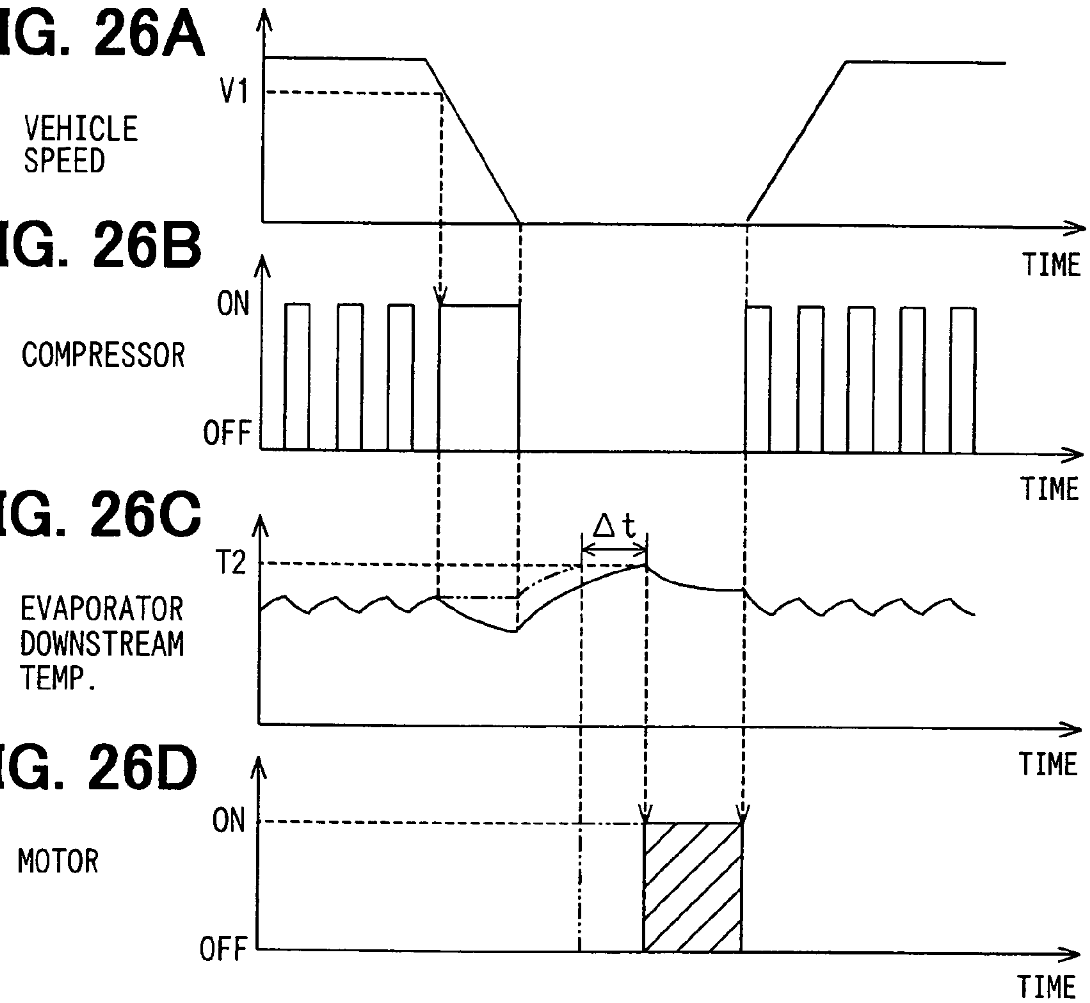


FIG. 27

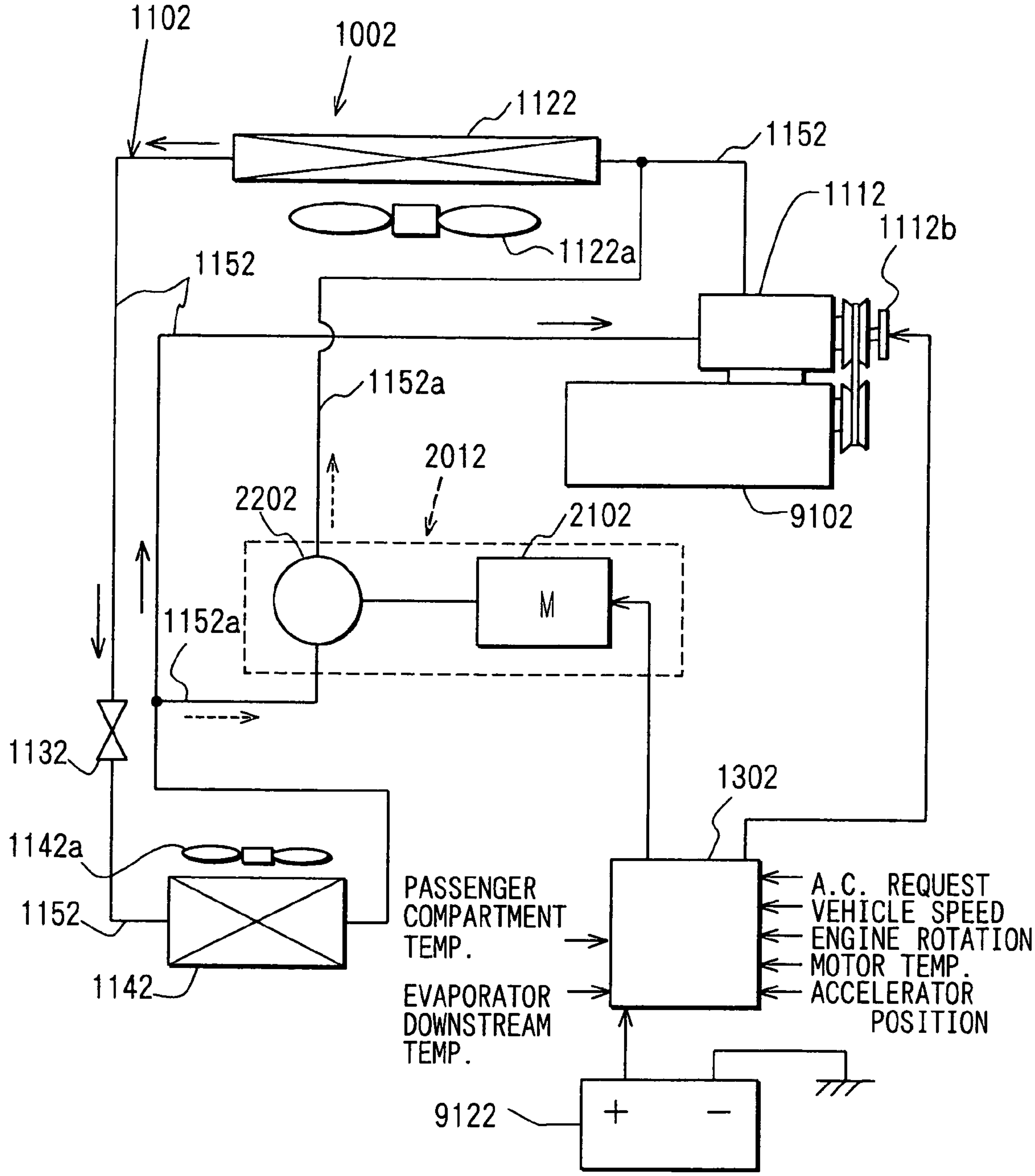

1102
1002
1122
1152
1112
1112b
1122a
1152
1152a
2012
2202
2102
9102
M
1152a
1132
1302
1142a
1152
1142
PASSENGER COMPARTMENT TEMP.
EVAPORATOR DOWNSTREAM TEMP.
A.C. REQUEST
VEHICLE SPEED
ENGINE ROTATION
MOTOR TEMP.
ACCELERATOR POSITION
9122
+
−

FIG. 28

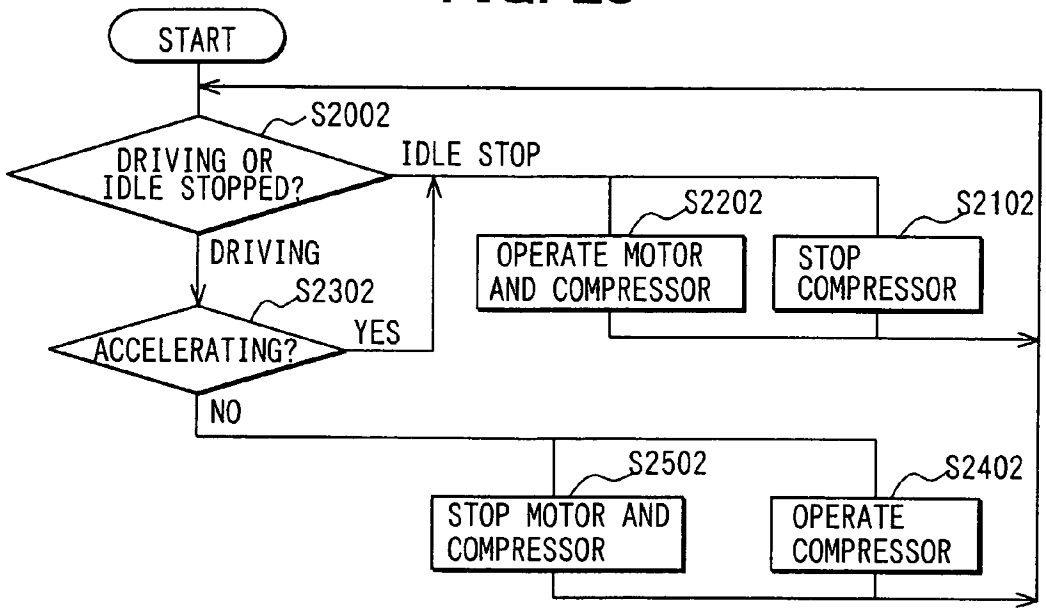


FIG. 29A

FIG. 29B

FIG. 29C

FIG. 29D

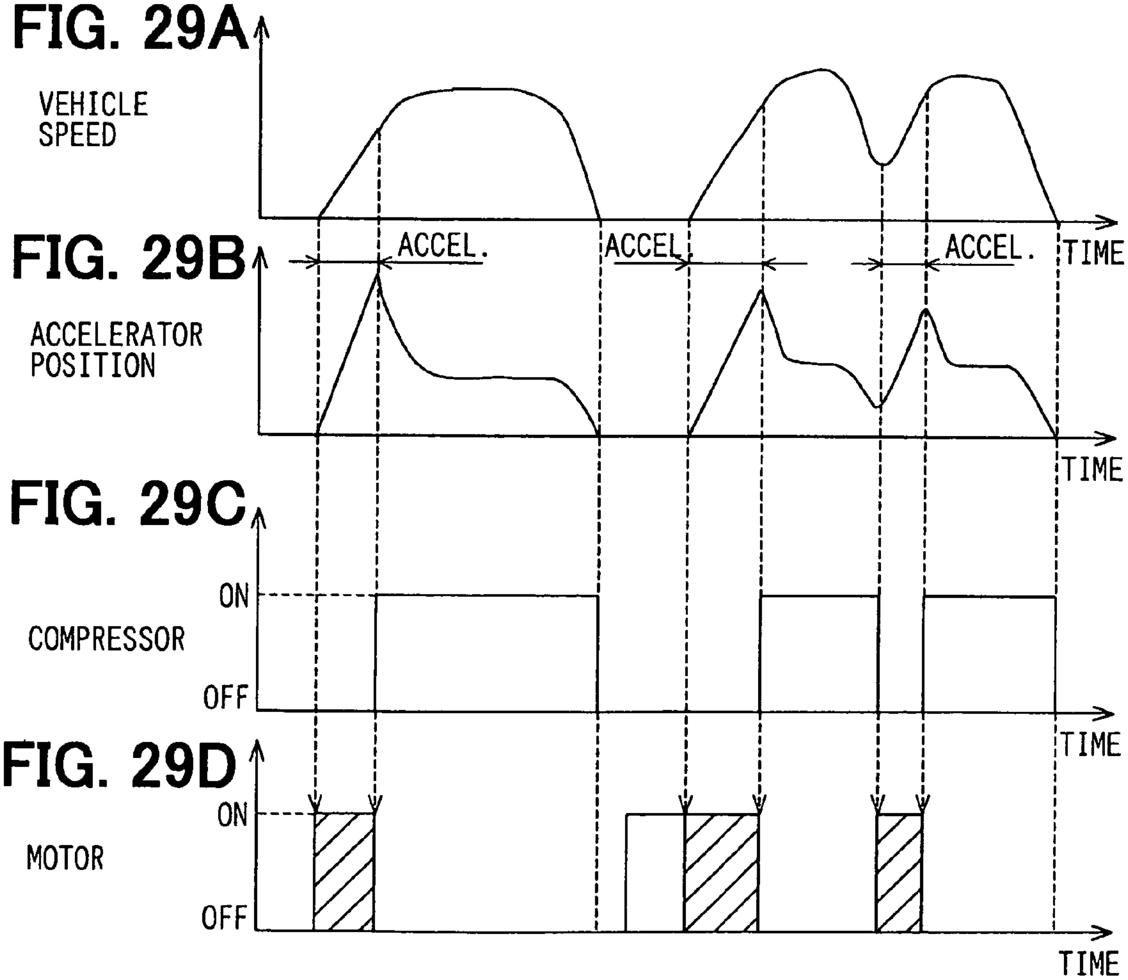


FIG. 30

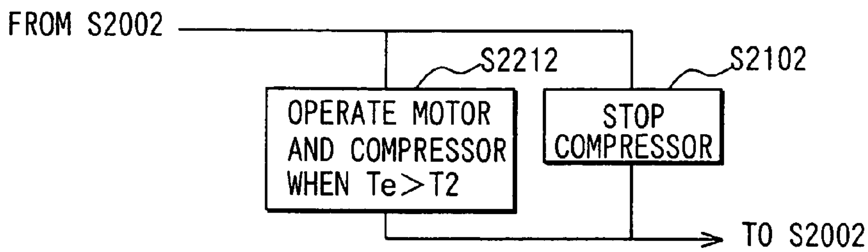


FIG. 31A
FIG. 31B
FIG. 31C
FIG. 31D
FIG. 31E

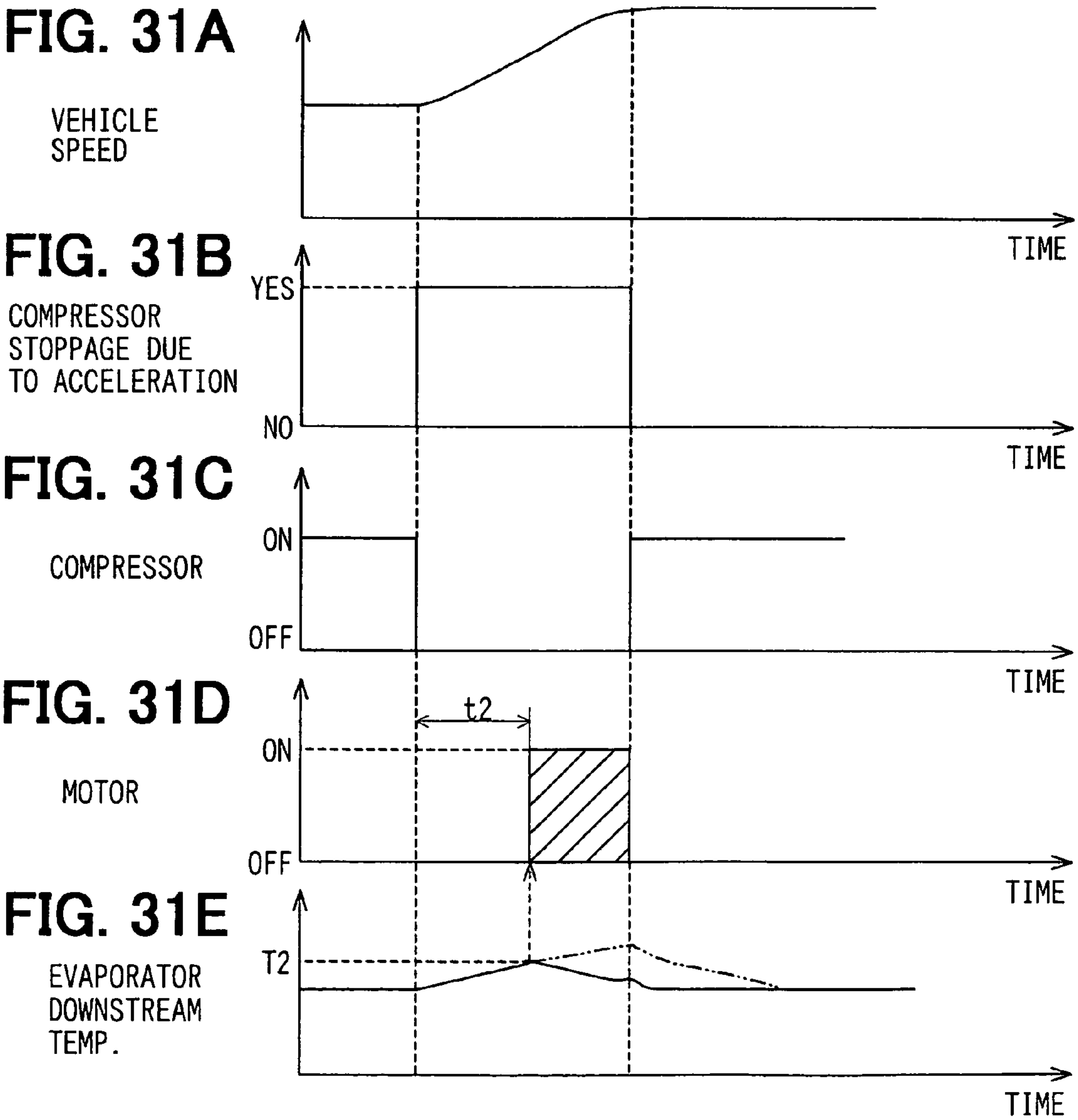


FIG. 32A

FIG. 32B

FIG. 32C

FIG. 32D

FIG. 32E

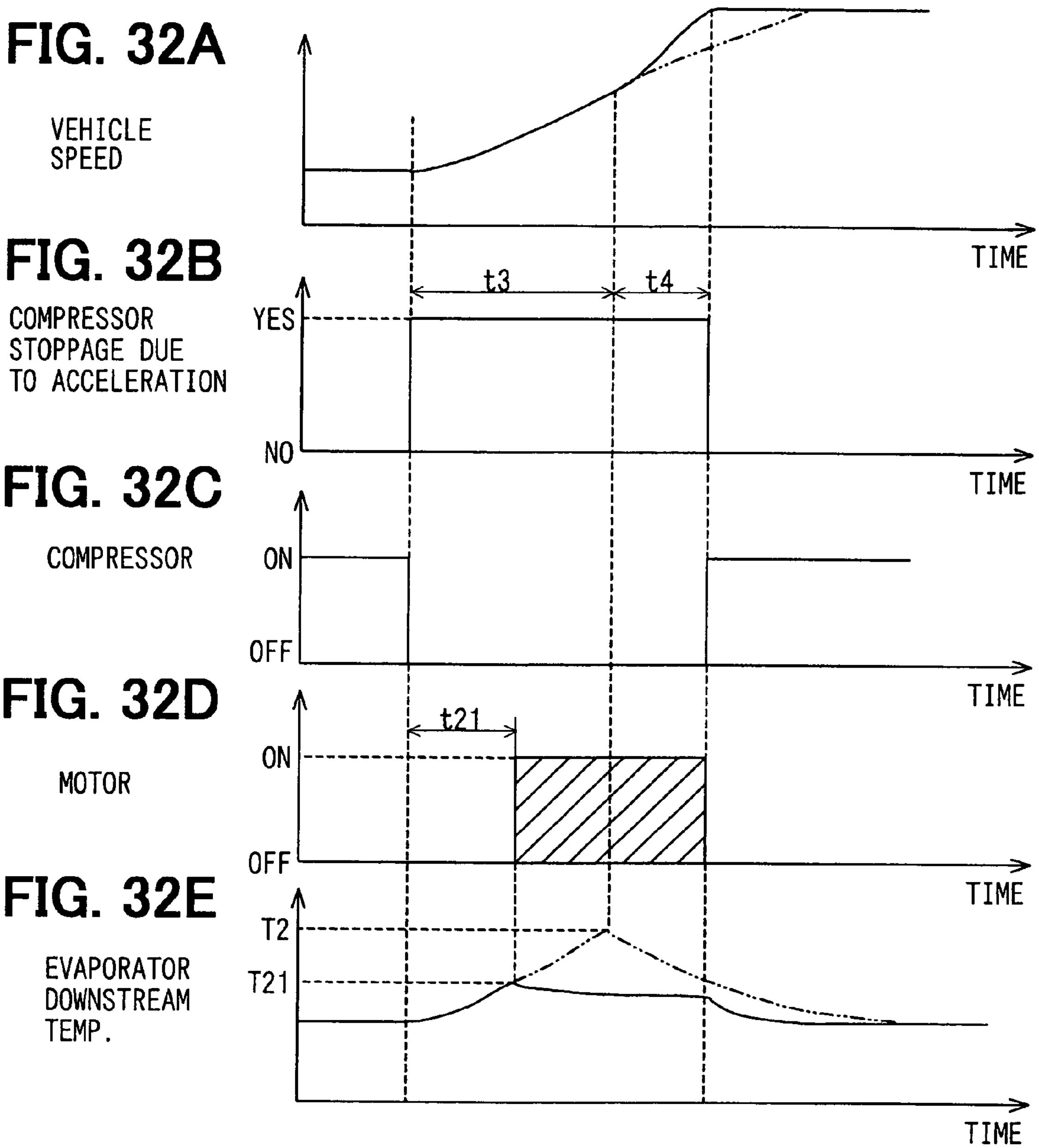


FIG. 33

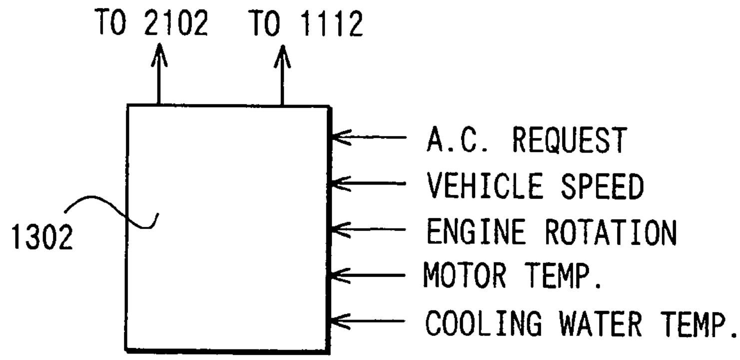
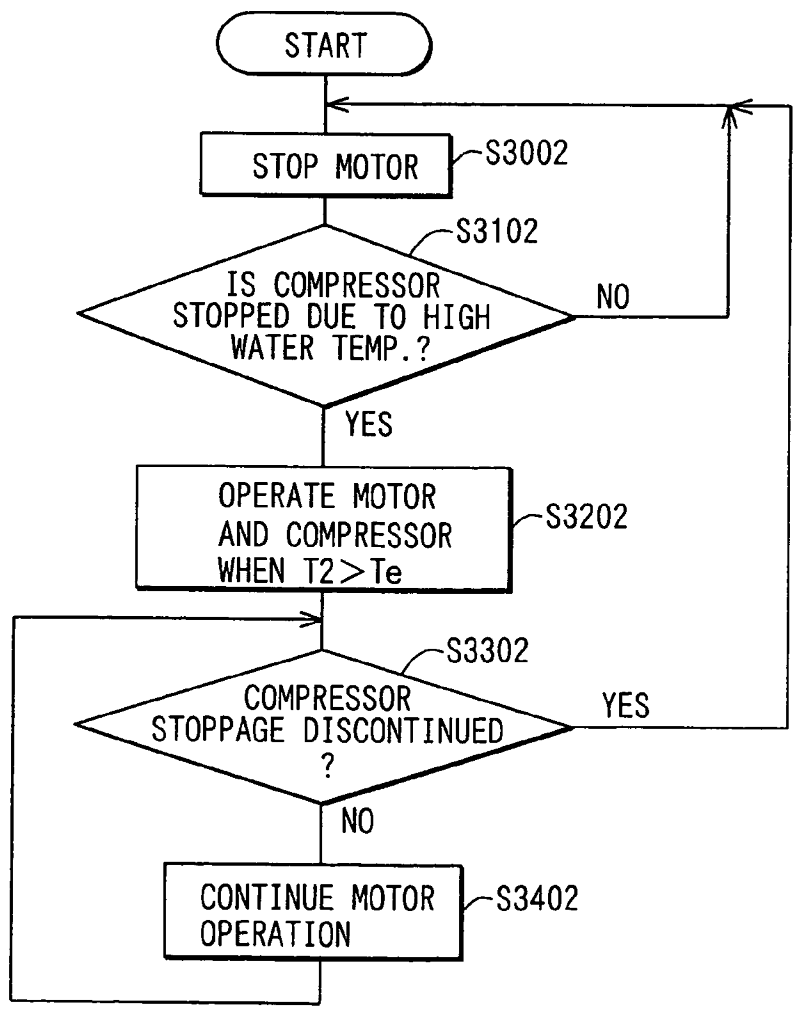

TO 2102
TO 1112
1302
A.C. REQUEST
VEHICLE SPEED
ENGINE ROTATION
MOTOR TEMP.
COOLING WATER TEMP.

FIG. 34


START
STOP MOTOR
S3002
S3102
IS COMPRESSOR STOPPED DUE TO HIGH WATER TEMP.?
NO
YES
OPERATE MOTOR AND COMPRESSOR WHEN T2>Te
S3202
S3302
COMPRESSOR STOPPAGE DISCONTINUED ?
YES
NO
CONTINUE MOTOR OPERATION
S3402

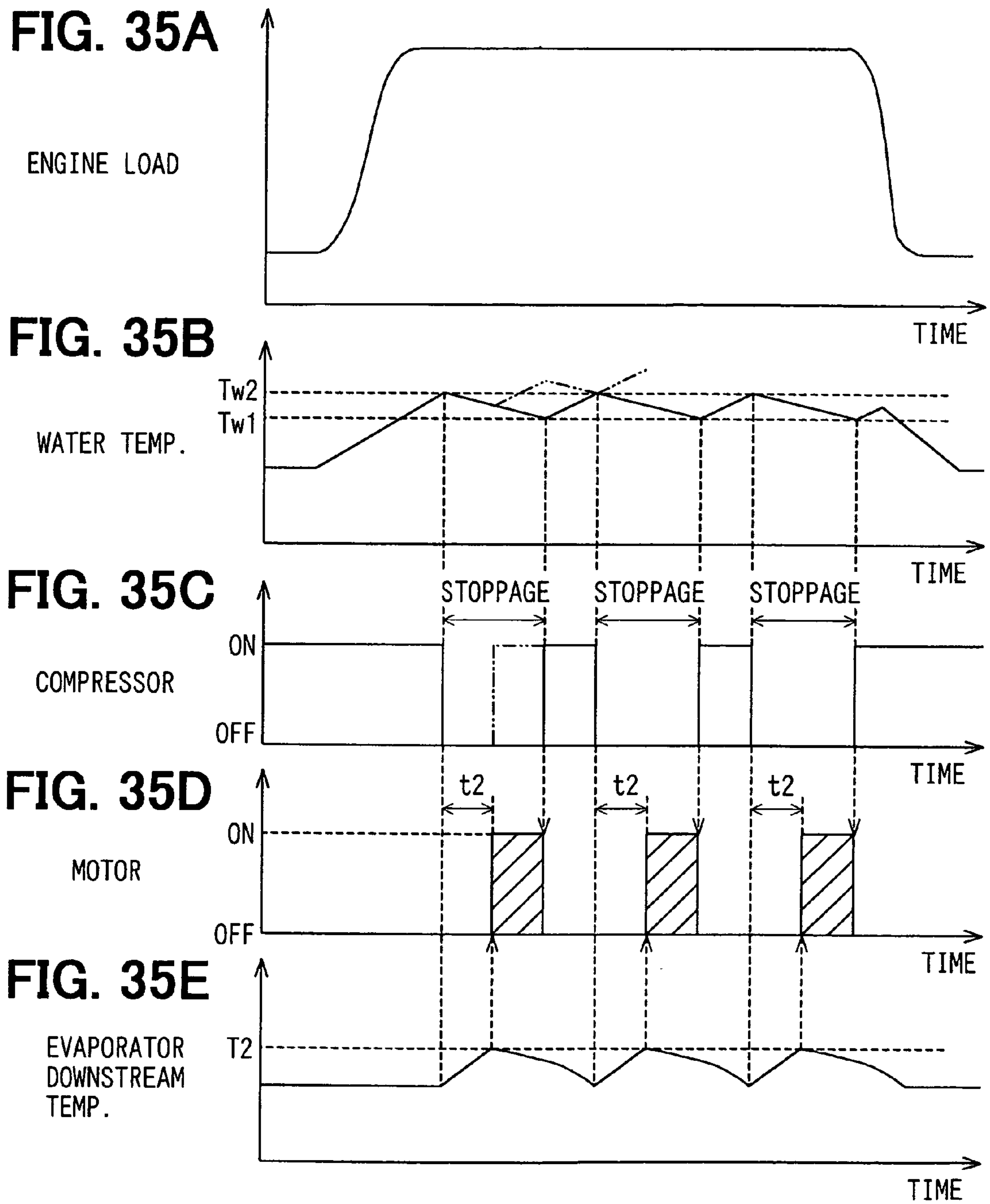


FIG. 35A
ENGINE LOAD
TIME
FIG. 35B
Tw2
Tw1
WATER TEMP.
TIME
FIG. 35C
STOPPAGE
STOPPAGE
STOPPAGE
ON
COMPRESSOR
OFF
TIME
FIG. 35D
t2
t2
t2
ON
MOTOR
OFF
TIME
FIG. 35E
EVAPORATOR
DOWNSTREAM
TEMP.
T2
TIME

FIG. 36

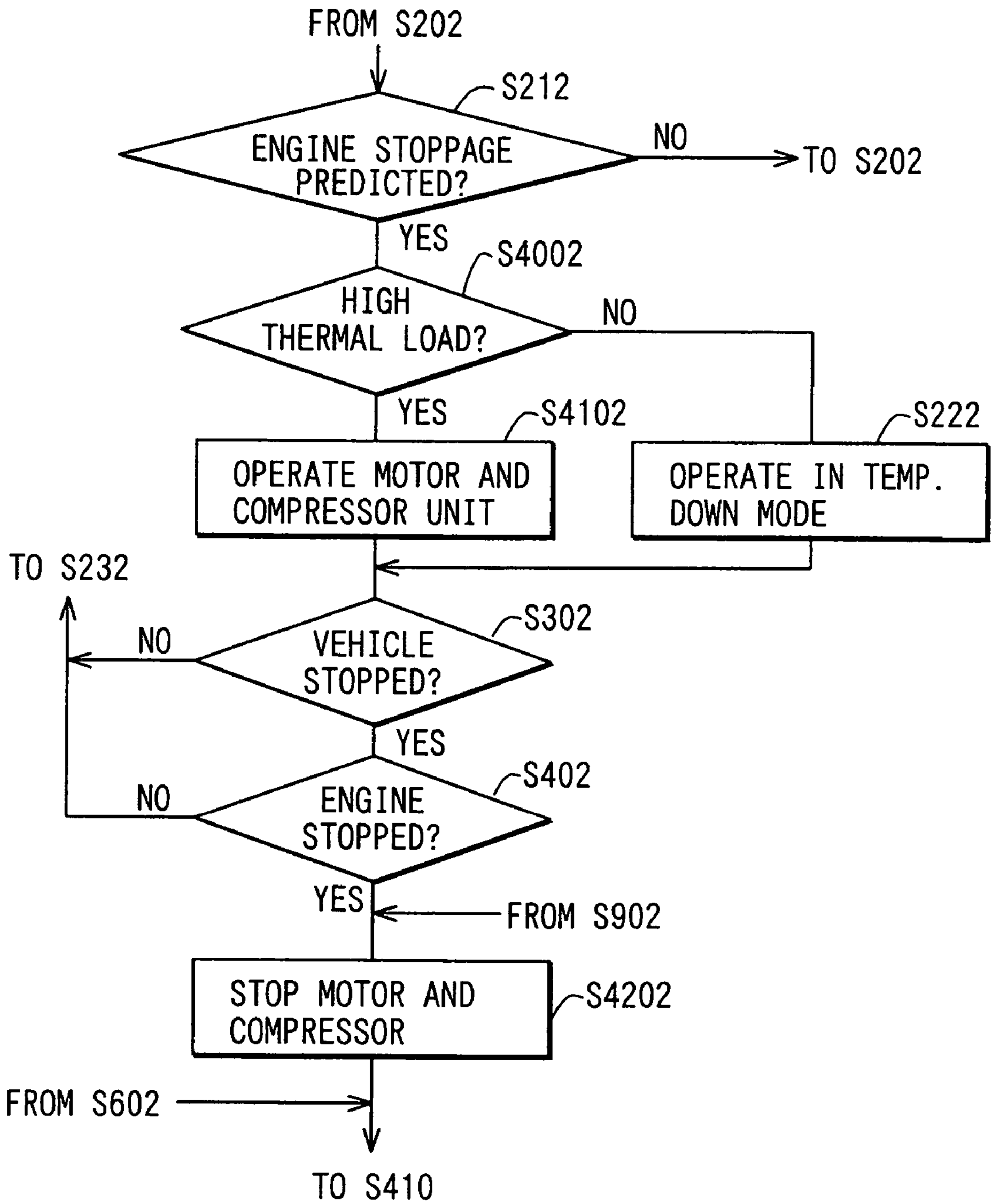

FROM S202
S212
ENGINE STOPPAGE PREDICTED?
NO
TO S202
YES
S4002
HIGH THERMAL LOAD?
NO
YES
S4102
OPERATE MOTOR AND COMPRESSOR UNIT
S222
OPERATE IN TEMP. DOWN MODE
TO S232
S302
NO
VEHICLE STOPPED?
YES
S402
NO
ENGINE STOPPED?
YES
FROM S902
STOP MOTOR AND COMPRESSOR
S4202
FROM S602
TO S410

FIG. 37A

FIG. 37B

FIG. 37C

FIG. 37D

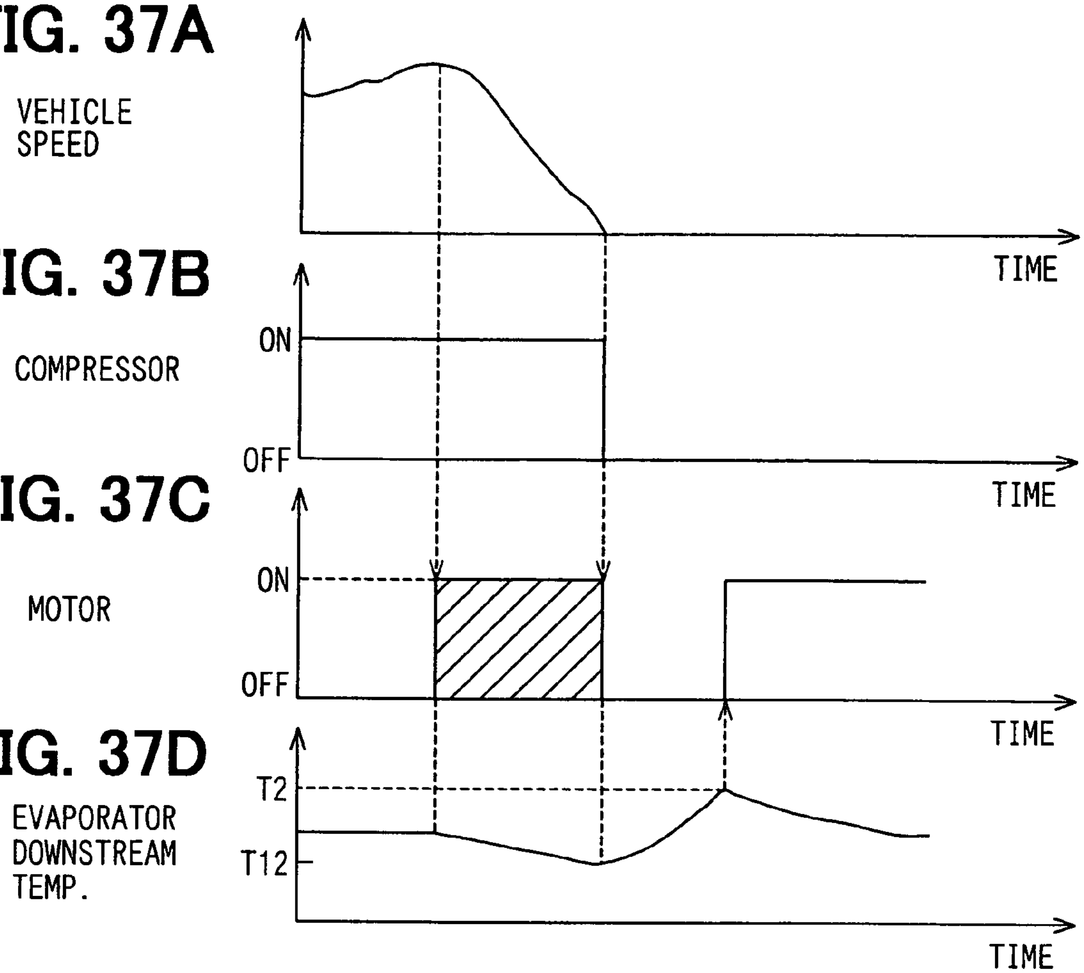


FIG. 38

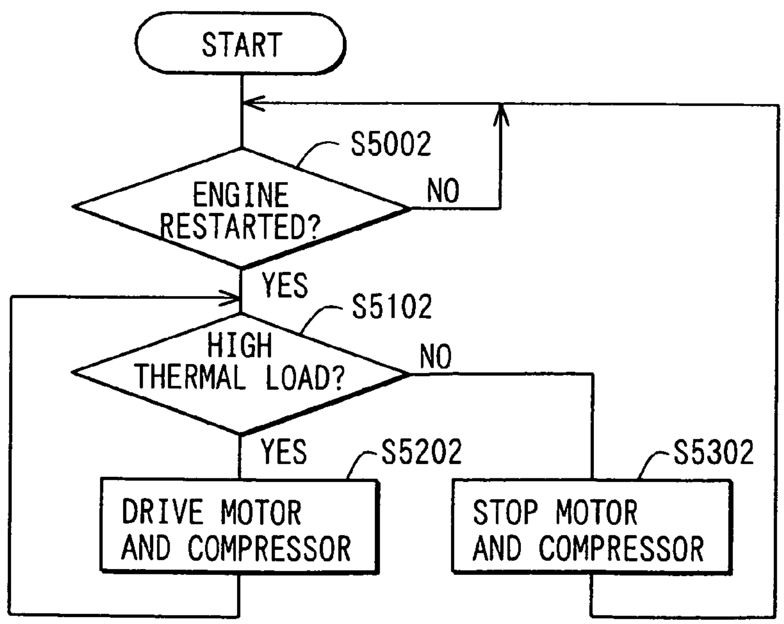


FIG. 39A

FIG. 39B

FIG. 39C

FIG. 39D

FIG. 39E

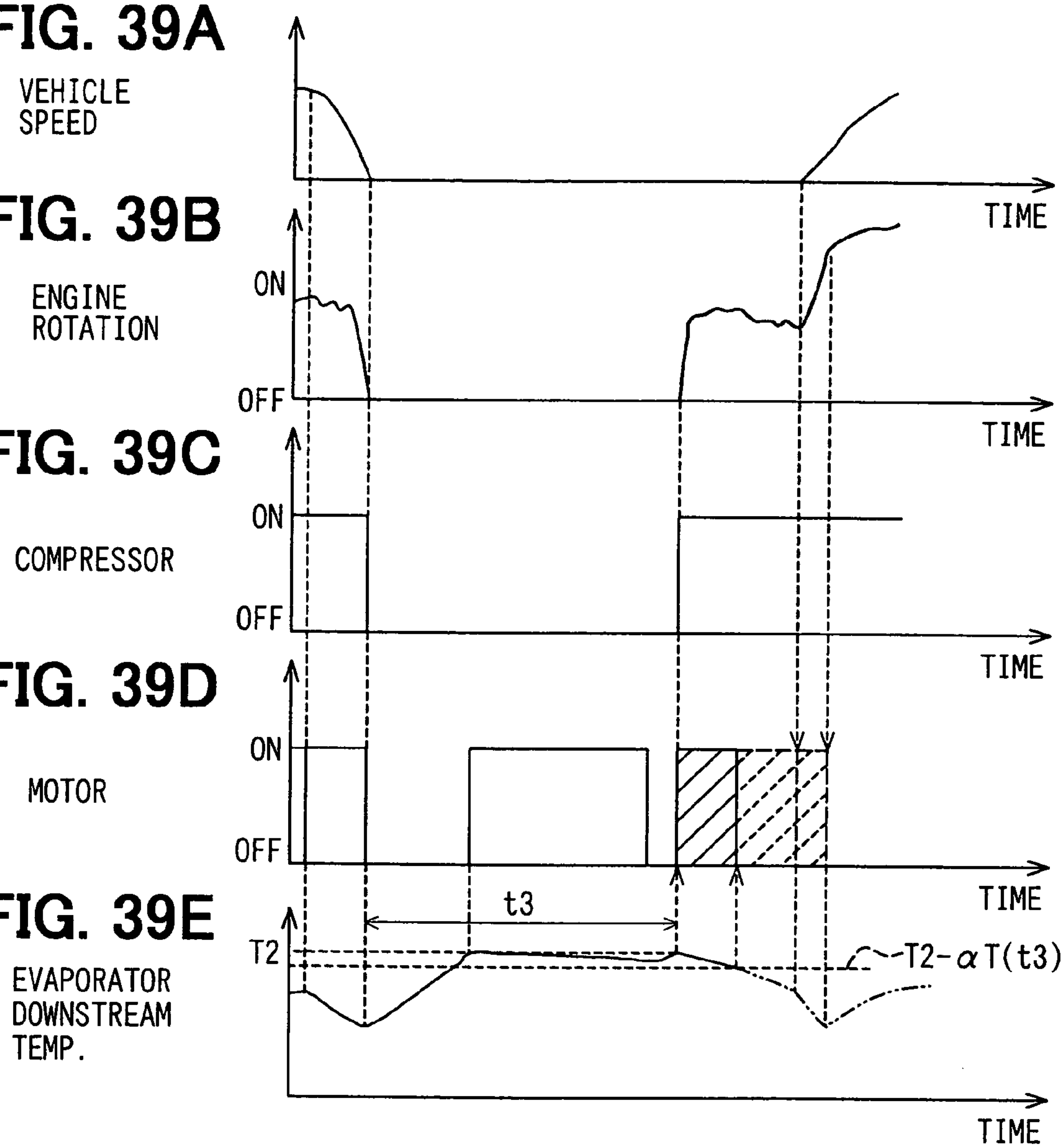


FIG. 40

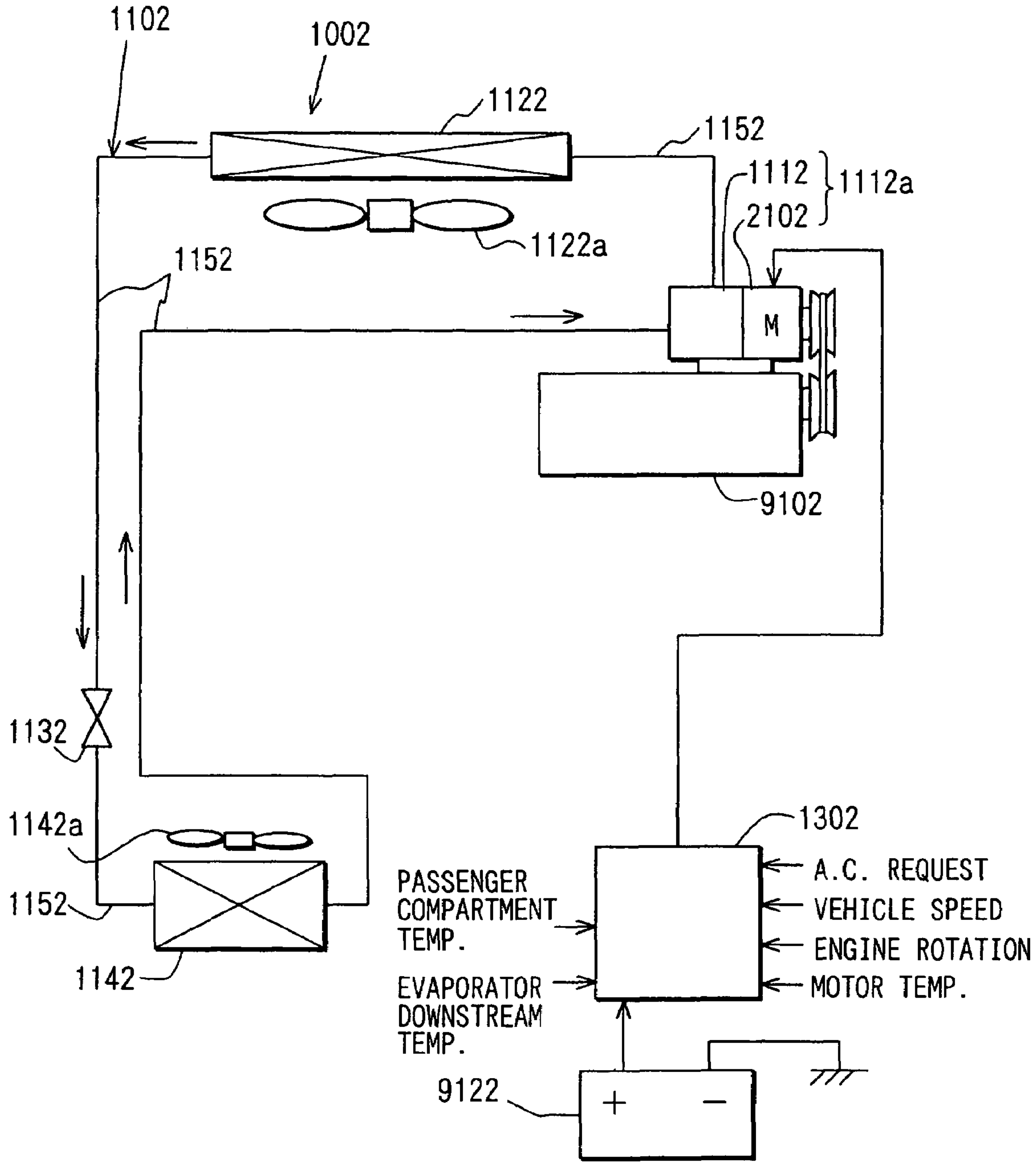

1102
1002
1122
1152
1112
2102
1112a
1122a
1152
M
9102
1132
1142a
1152
1142
1302
A.C. REQUEST
VEHICLE SPEED
ENGINE ROTATION
MOTOR TEMP.
PASSENGER COMPARTMENT TEMP.
EVAPORATOR DOWNSTREAM TEMP.
9122
+
−

FIG. 41

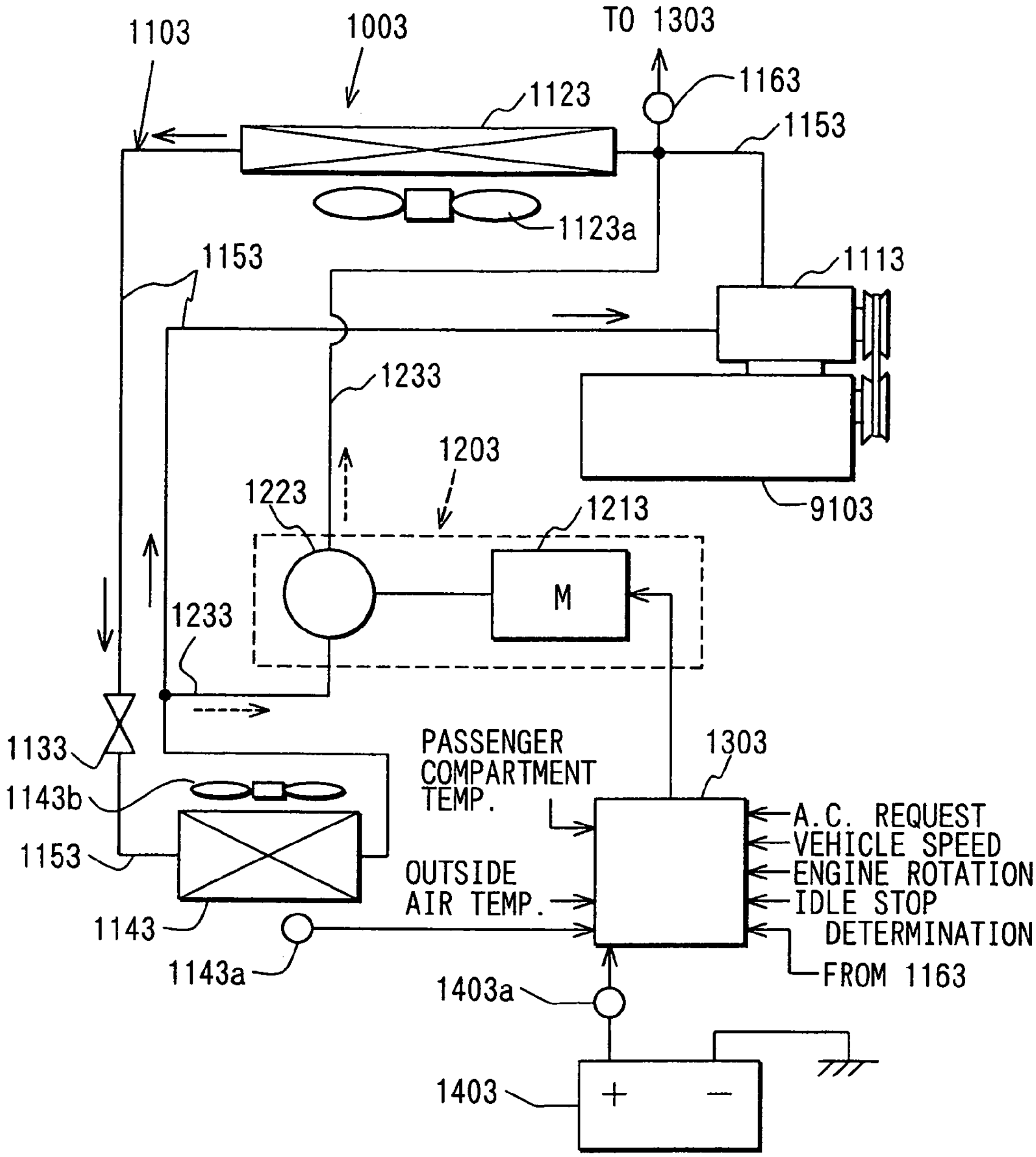


FIG. 42A

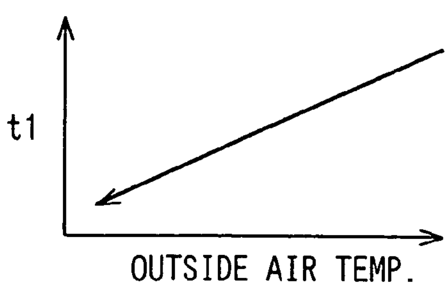


FIG. 42B

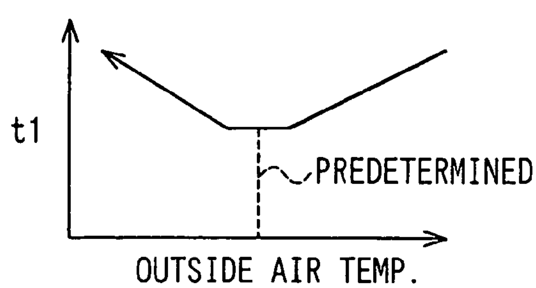


FIG. 43A

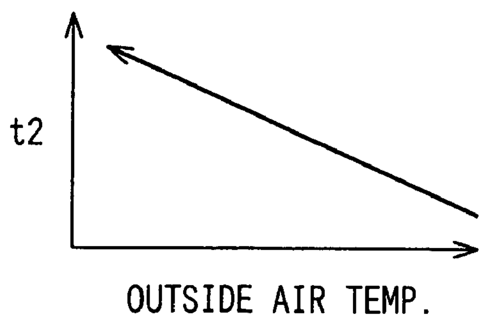


FIG. 43B

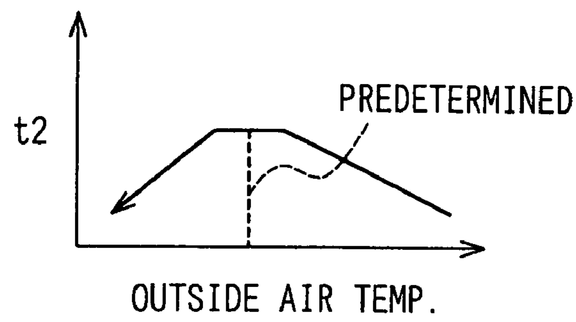


FIG. 44

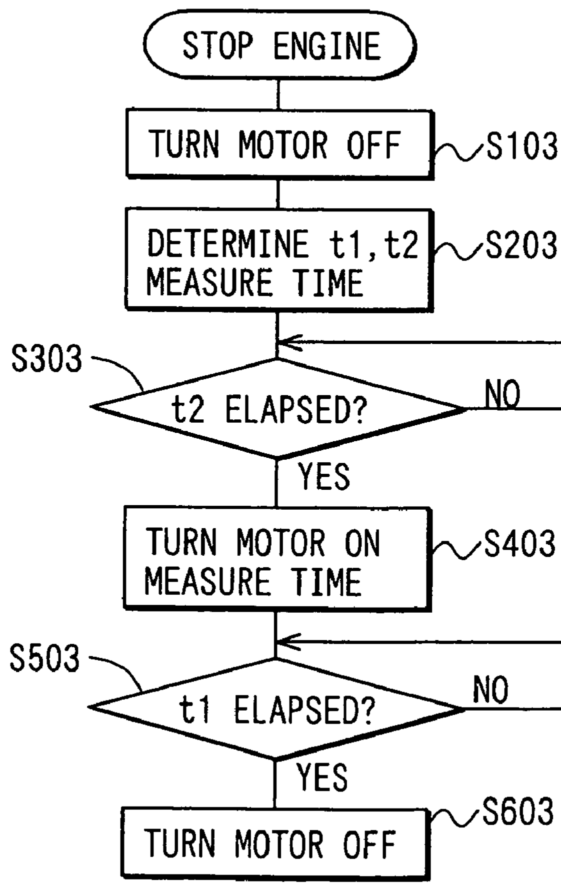


FIG. 45A

FIG. 45B

FIG. 45C

FIG. 45D

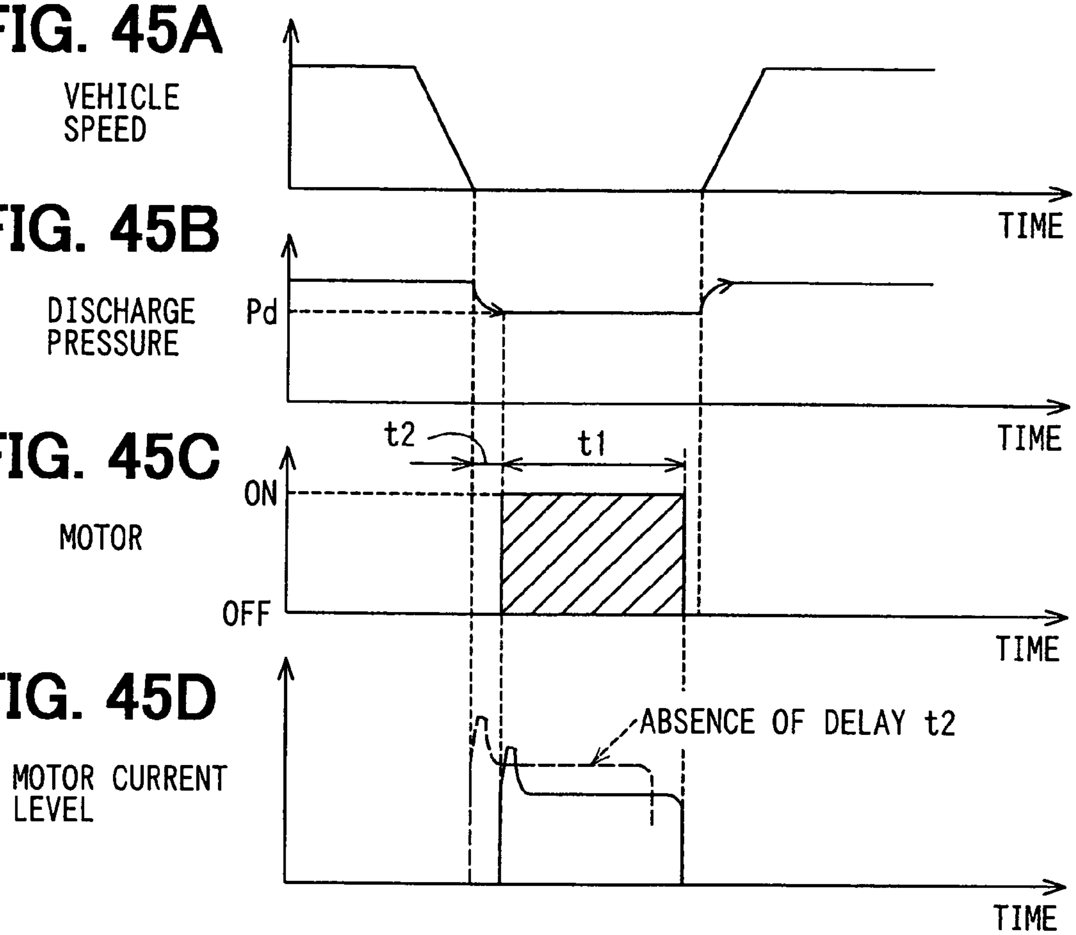


FIG. 46

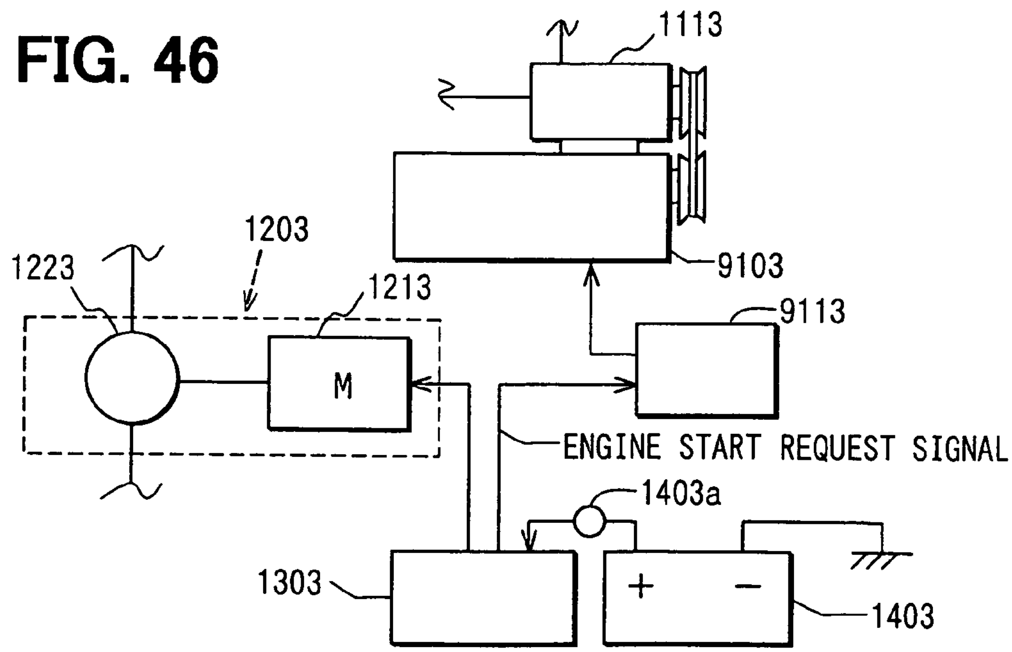
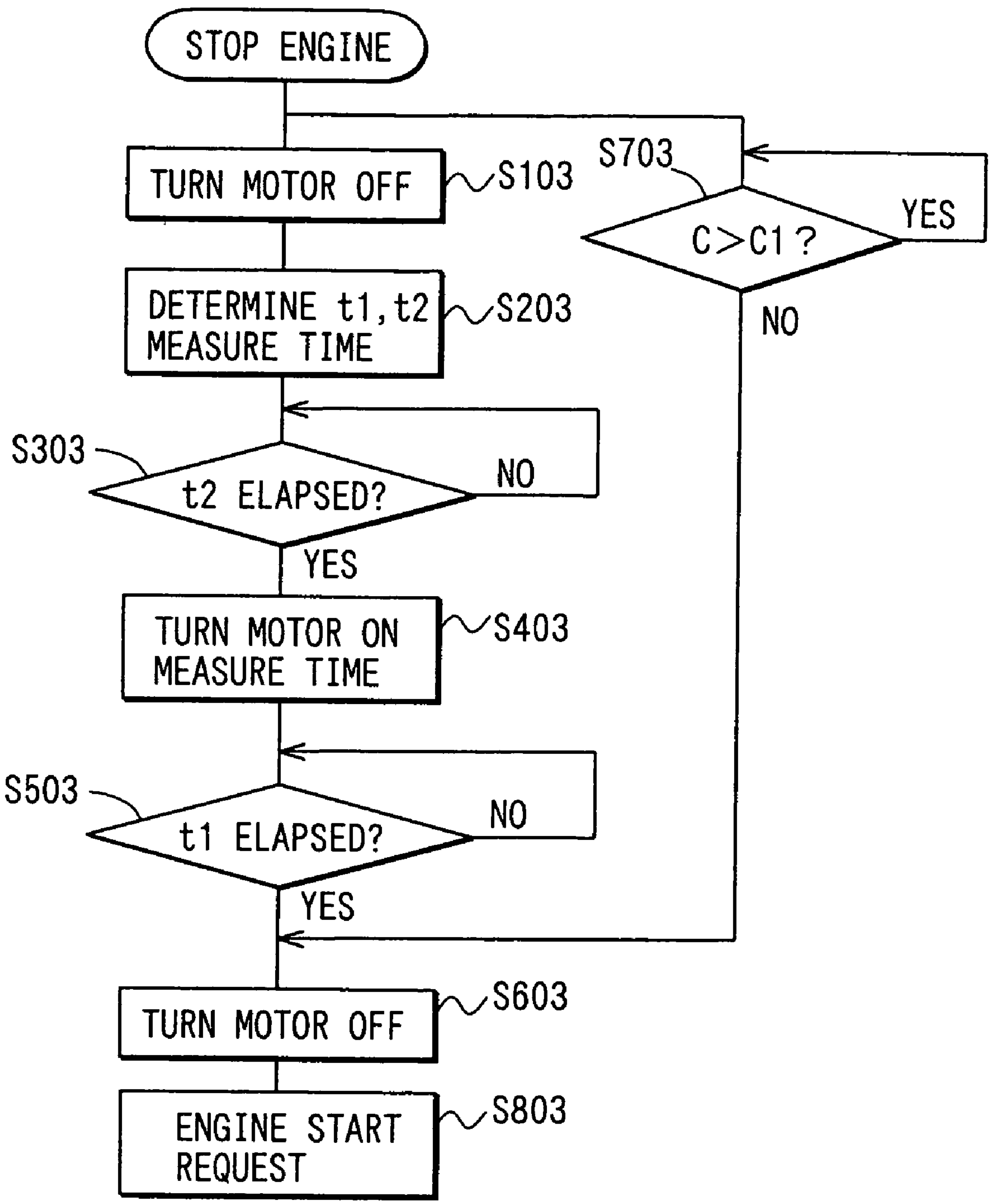


FIG. 47


STOP ENGINE
TURN MOTOR OFF
S103
DETERMINE t1, t2
MEASURE TIME
S203
S303
t2 ELAPSED?
NO
YES
TURN MOTOR ON
MEASURE TIME
S403
S503
t1 ELAPSED?
NO
YES
TURN MOTOR OFF
S603
ENGINE START
REQUEST
S803
S703
C>C1 ?
YES
NO

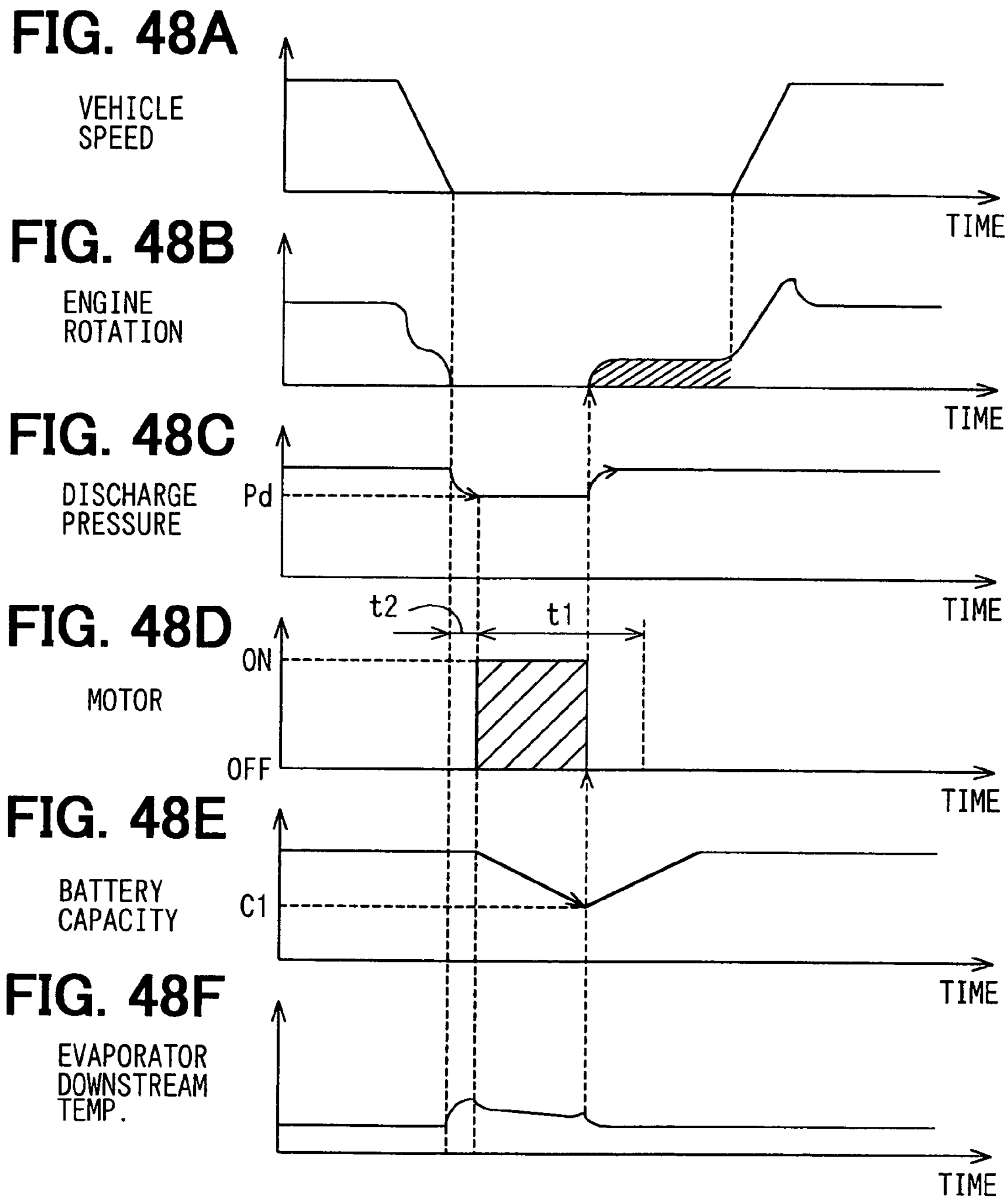


FIG. 48A
VEHICLE SPEED
TIME
FIG. 48B
ENGINE ROTATION
TIME
FIG. 48C
DISCHARGE PRESSURE
Pd
TIME
FIG. 48D
t2
t1
ON
MOTOR
OFF
TIME
FIG. 48E
BATTERY CAPACITY
C1
TIME
FIG. 48F
EVAPORATOR DOWNSTREAM TEMP.
TIME

FIG. 49A
FIG. 49B
FIG. 49C
FIG. 49D
FIG. 49E

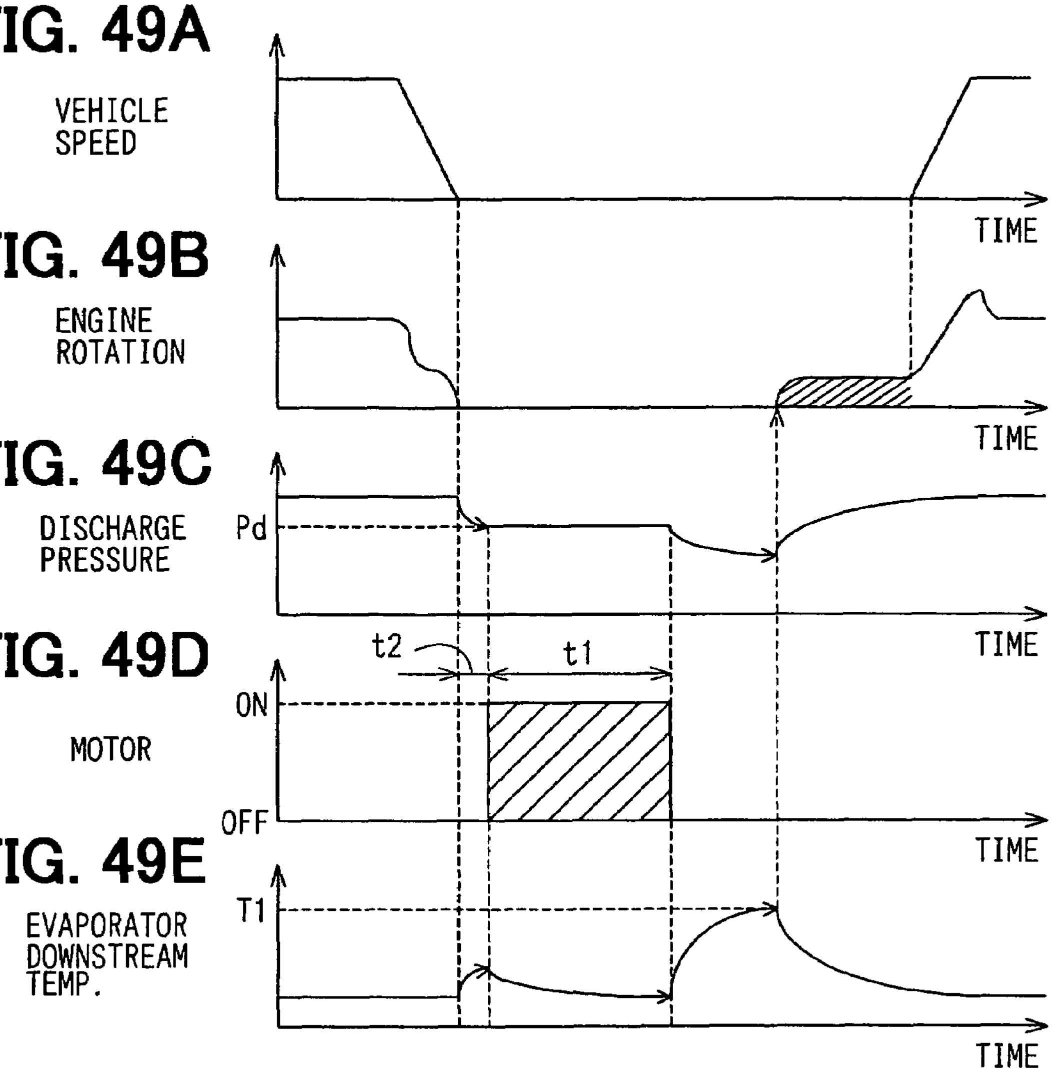


FIG. 50A
FIG. 50B

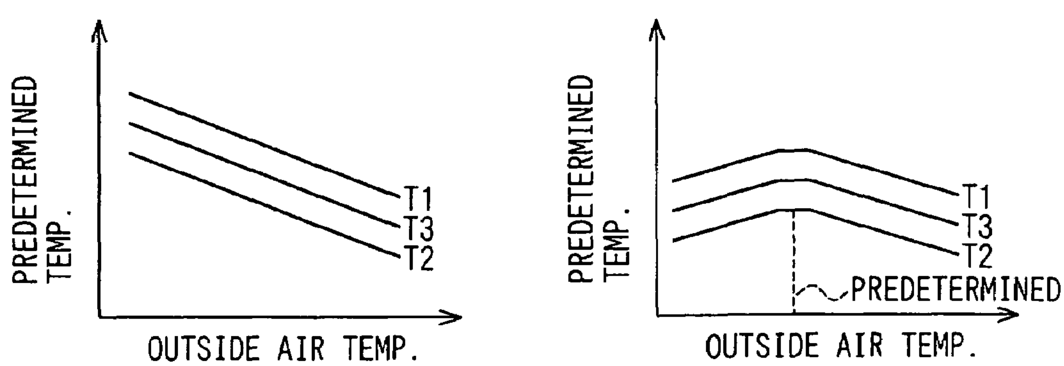


FIG. 51

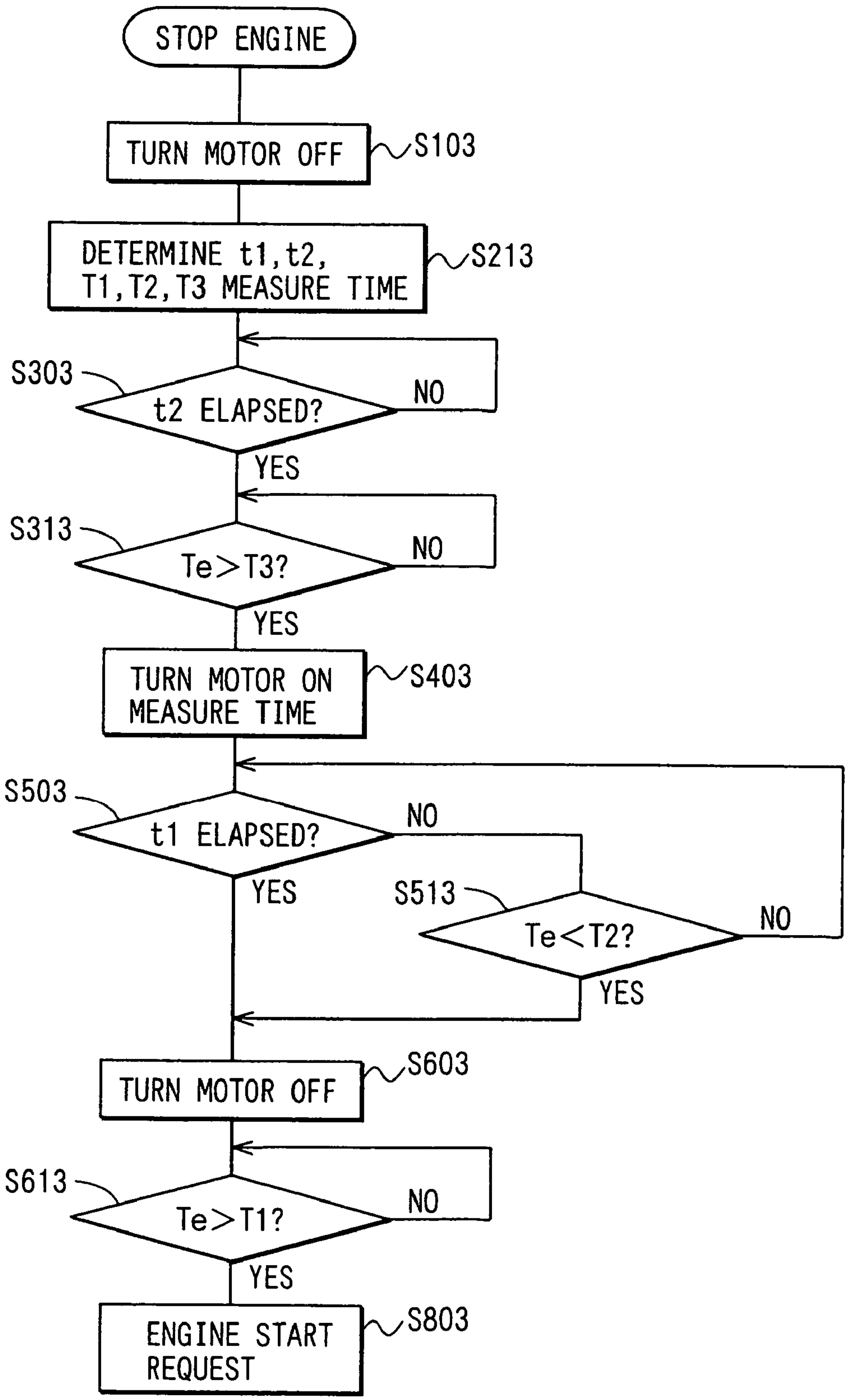

STOP ENGINE
TURN MOTOR OFF
S103
DETERMINE t1,t2, T1,T2,T3 MEASURE TIME
S213
S303
t2 ELAPSED?
NO
YES
S313
Te>T3?
NO
YES
TURN MOTOR ON MEASURE TIME
S403
S503
t1 ELAPSED?
NO
YES
S513
Te<T2?
NO
YES
TURN MOTOR OFF
S603
S613
Te>T1?
NO
YES
ENGINE START REQUEST
S803

FIG. 52A

FIG. 52B

FIG. 52C

FIG. 52D

FIG. 52E

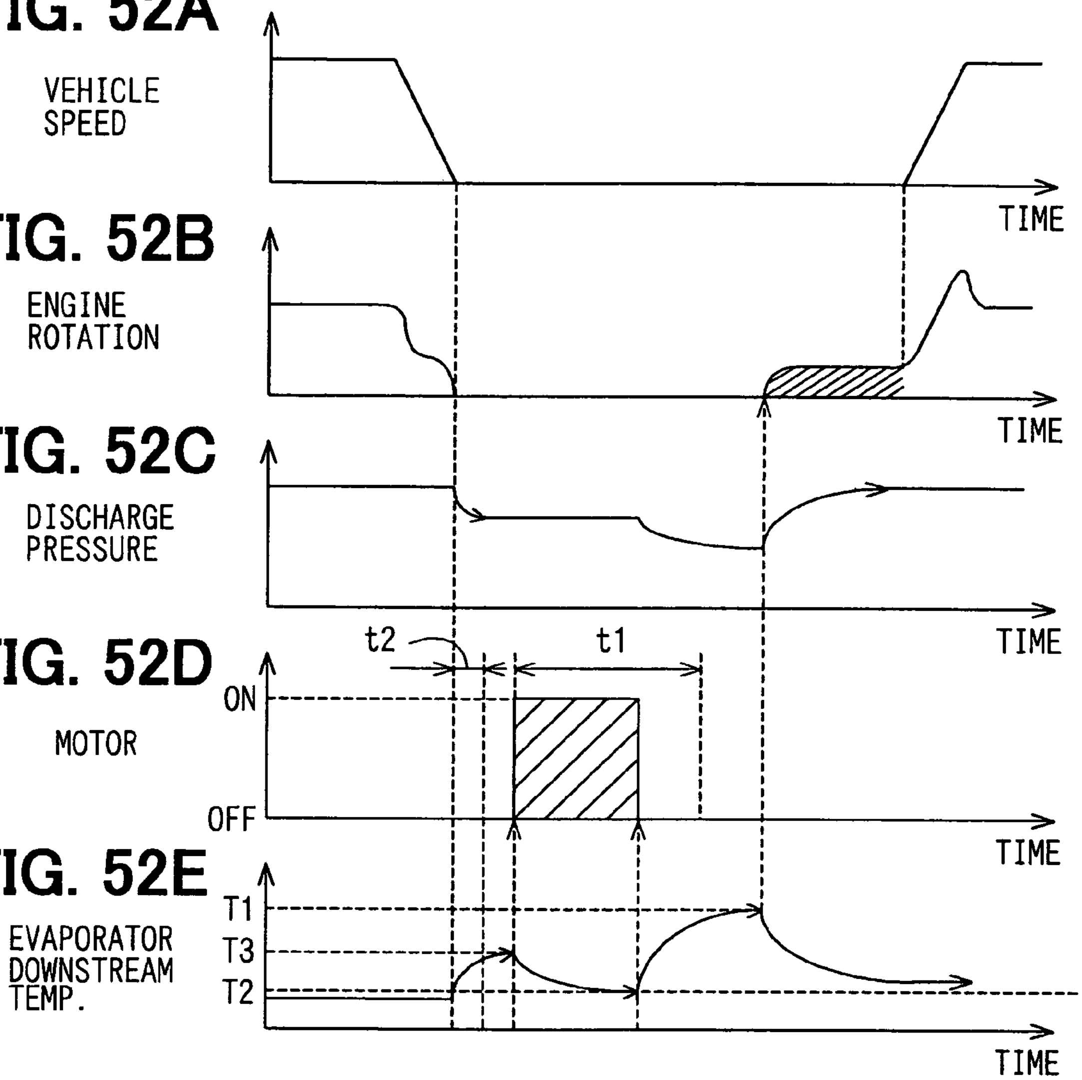
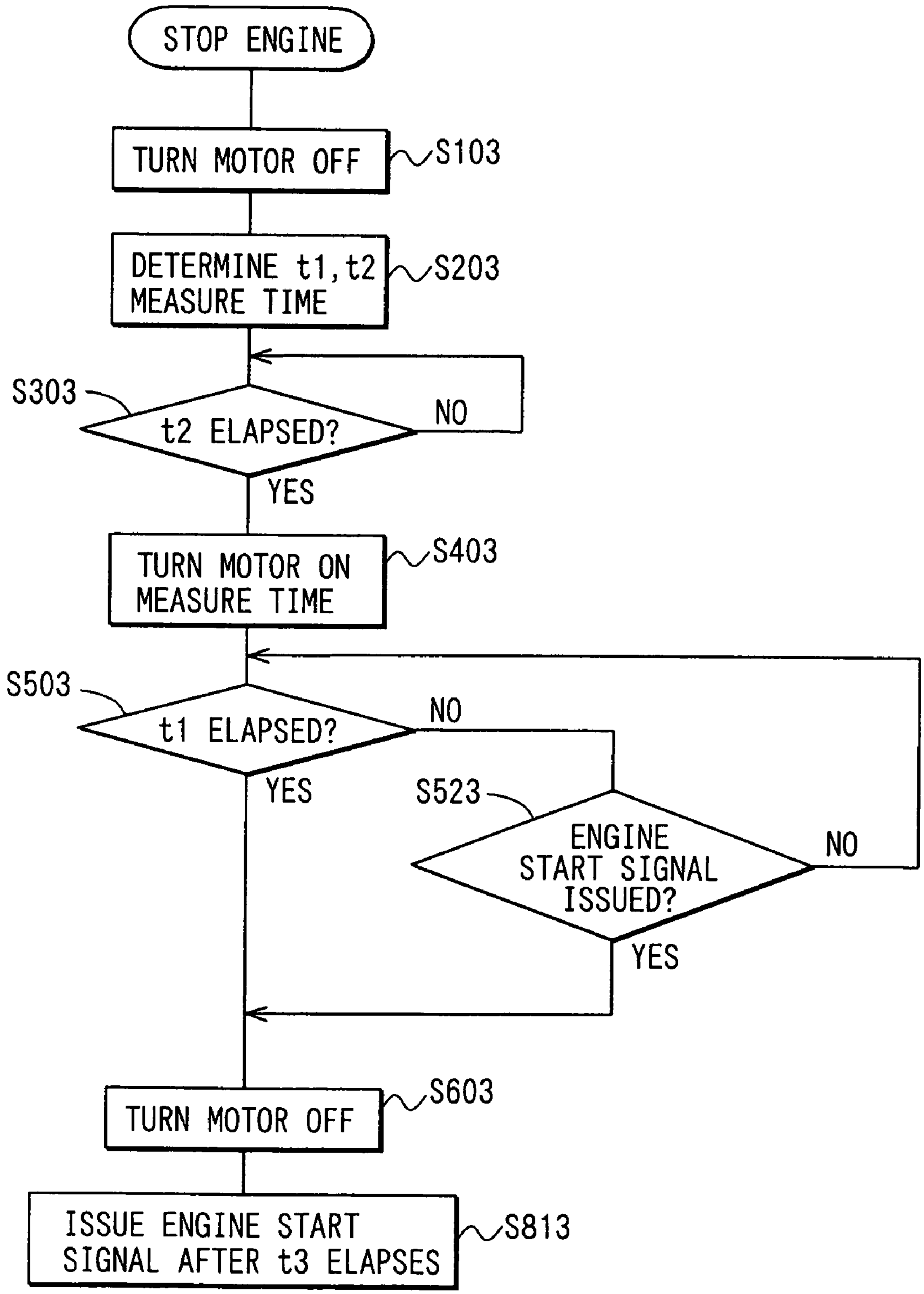


FIG. 53


STOP ENGINE
TURN MOTOR OFF
S103
DETERMINE t1,t2
MEASURE TIME
S203
S303
t2 ELAPSED?
NO
YES
TURN MOTOR ON
MEASURE TIME
S403
S503
t1 ELAPSED?
NO
YES
S523
ENGINE
START SIGNAL
ISSUED?
NO
YES
TURN MOTOR OFF
S603
ISSUE ENGINE START
SIGNAL AFTER t3 ELAPSES
S813

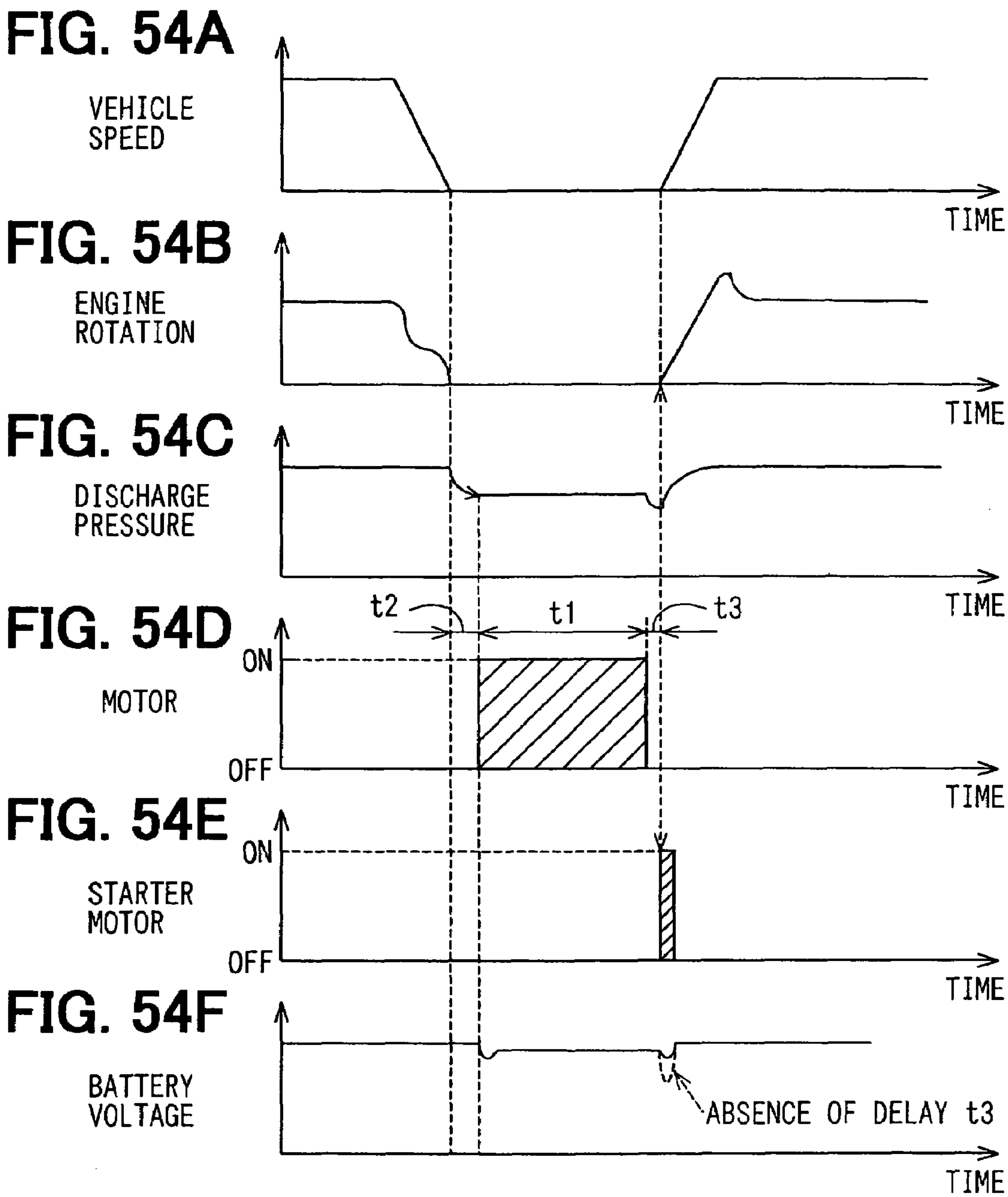


FIG. 54A
VEHICLE SPEED
TIME
FIG. 54B
ENGINE ROTATION
TIME
FIG. 54C
DISCHARGE PRESSURE
TIME
FIG. 54D
t2
t1
t3
ON
MOTOR
OFF
TIME
FIG. 54E
ON
STARTER MOTOR
OFF
TIME
FIG. 54F
BATTERY VOLTAGE
ABSENCE OF DELAY t3
TIME

FIG. 55A

FIG. 55B

FIG. 55C

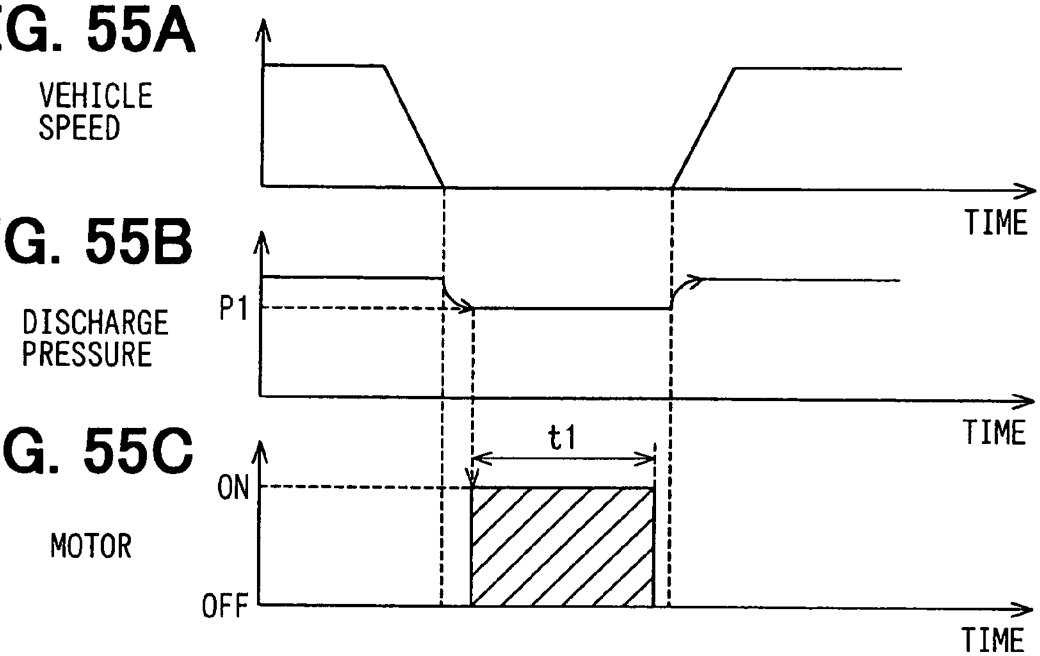


FIG. 56A

FIG. 56B

FIG. 56C

FIG. 56D

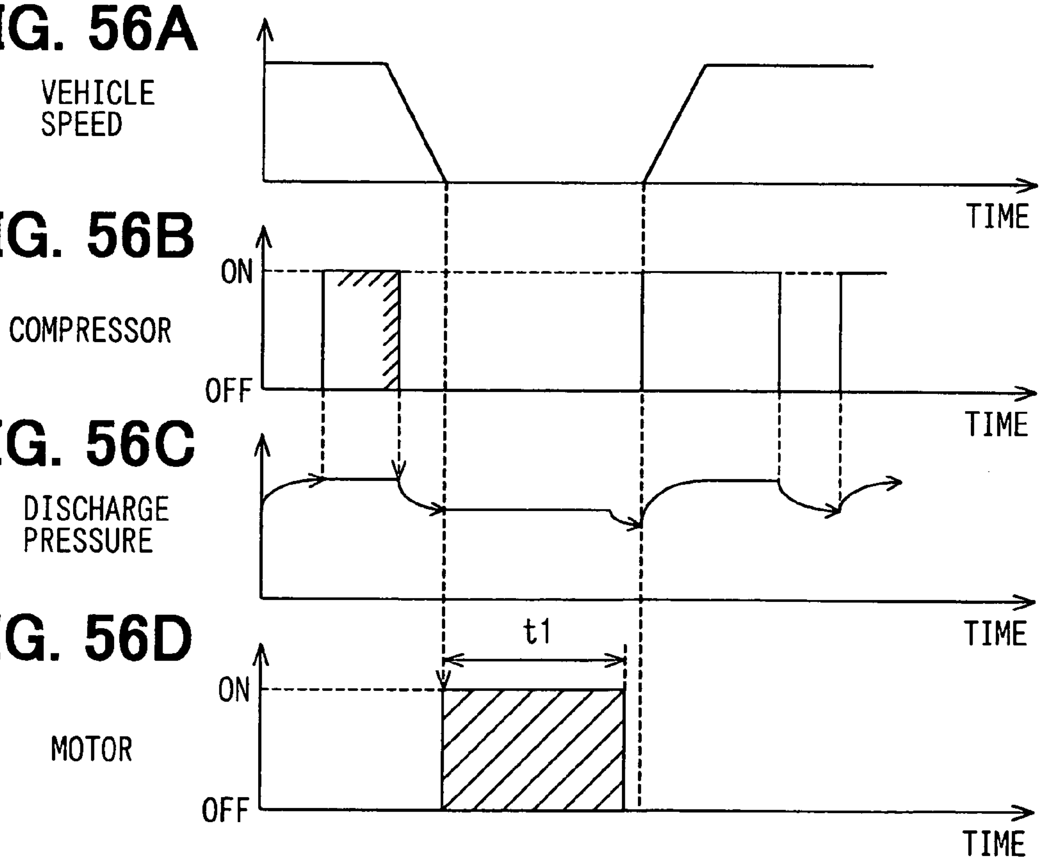


FIG. 57A

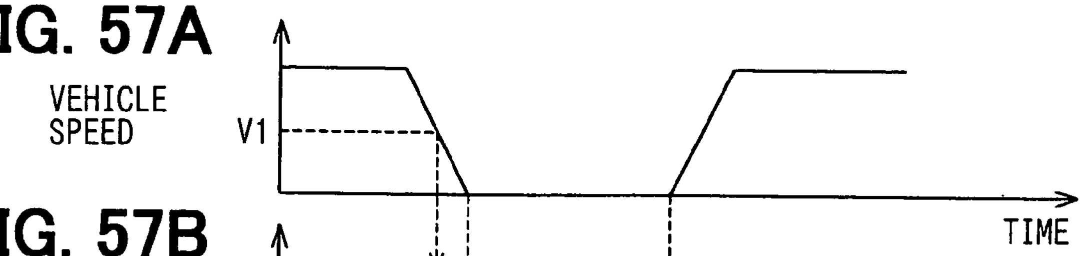


FIG. 57B

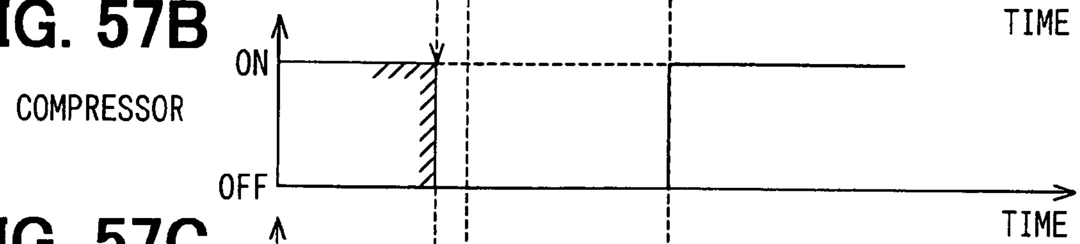


FIG. 57C

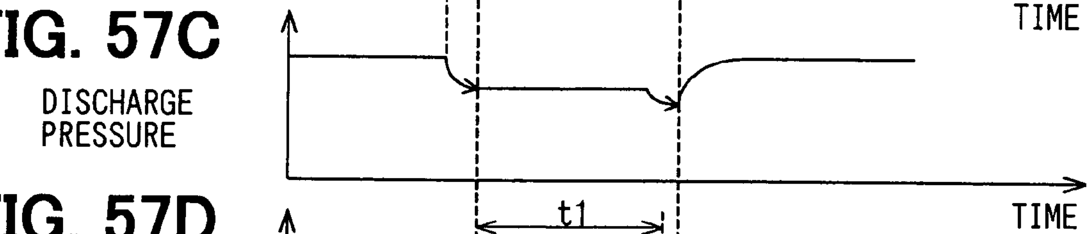


FIG. 57D

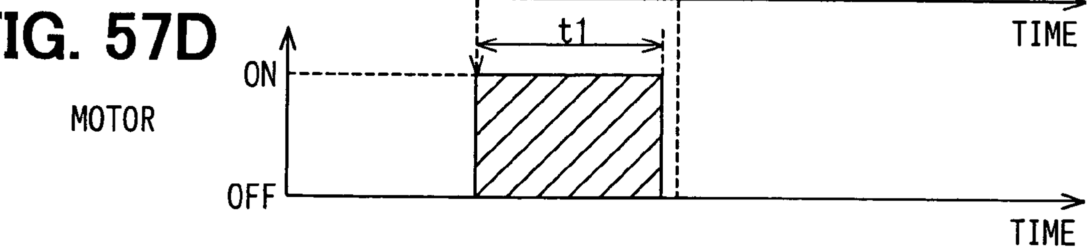


FIG. 58A

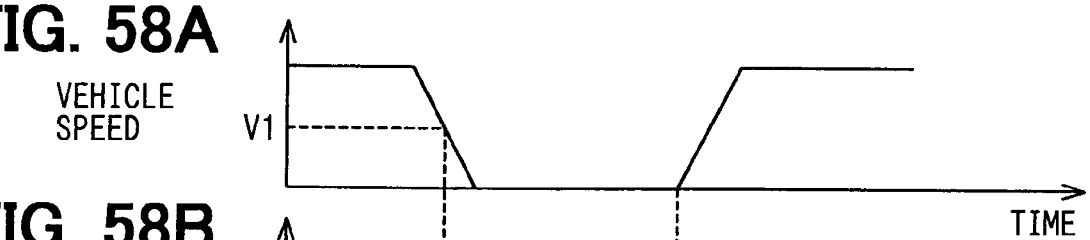


FIG. 58B

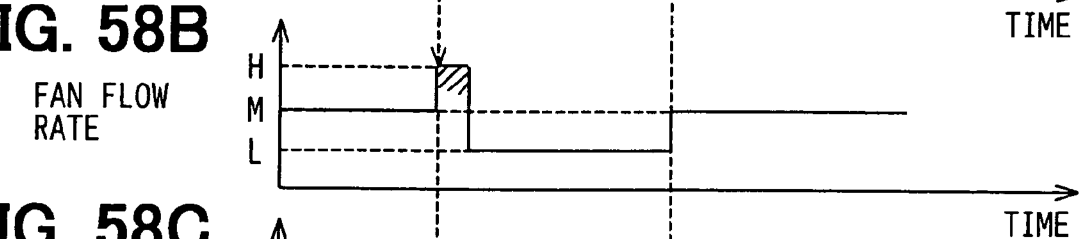


FIG. 58C

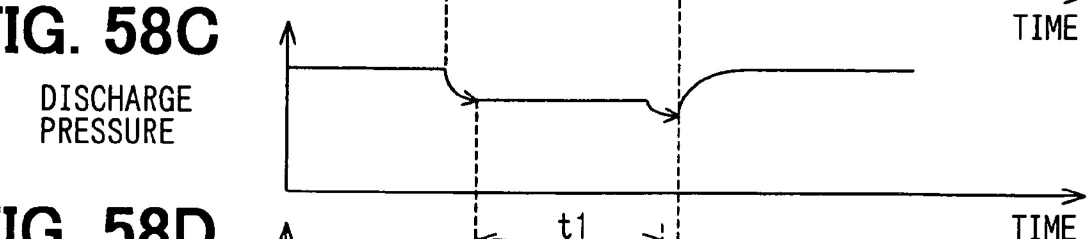


FIG. 58D

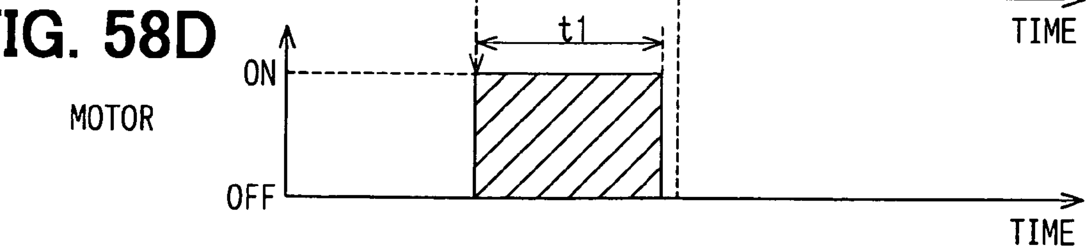


FIG. 59A

FIG. 59B

FIG. 59C

FIG. 59D

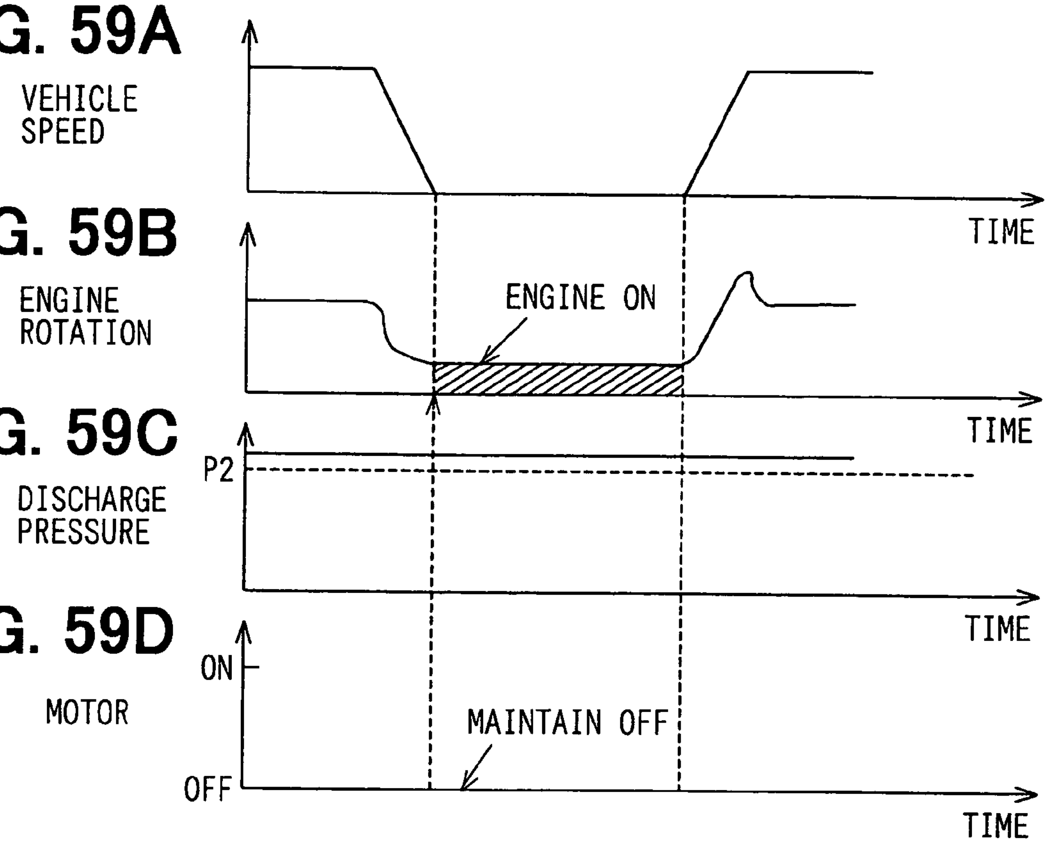


FIG. 60A

FIG. 60B

FIG. 60C

FIG. 60D

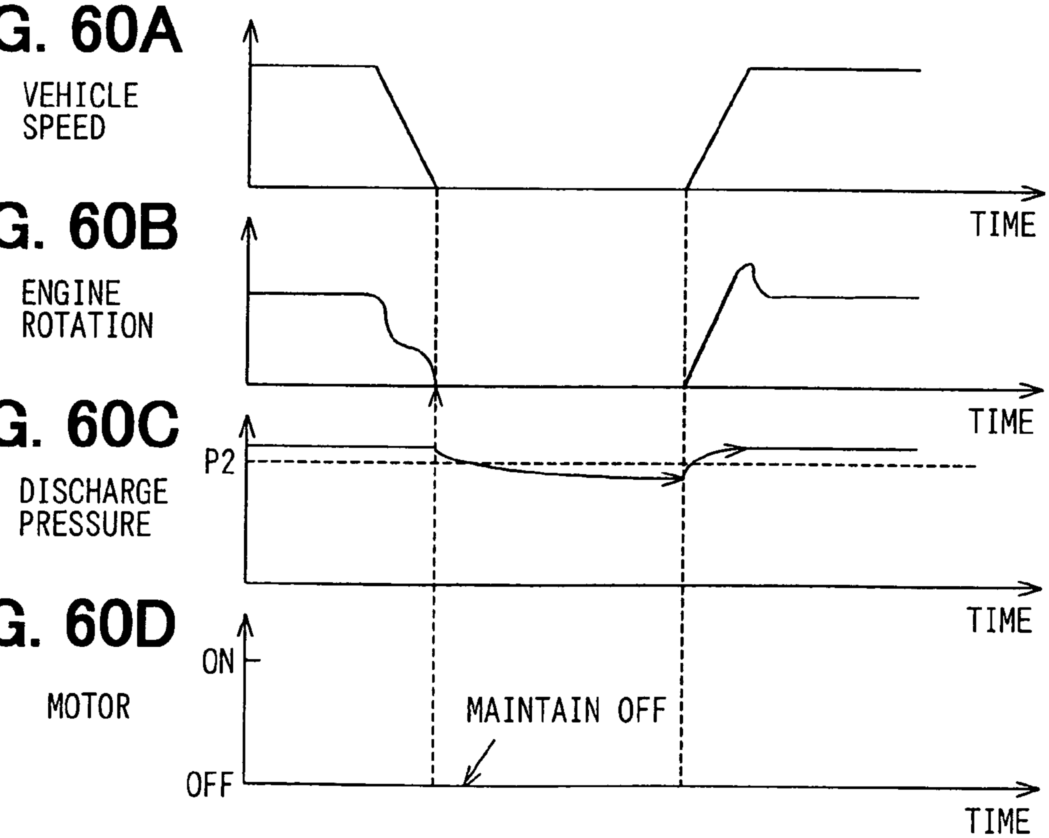
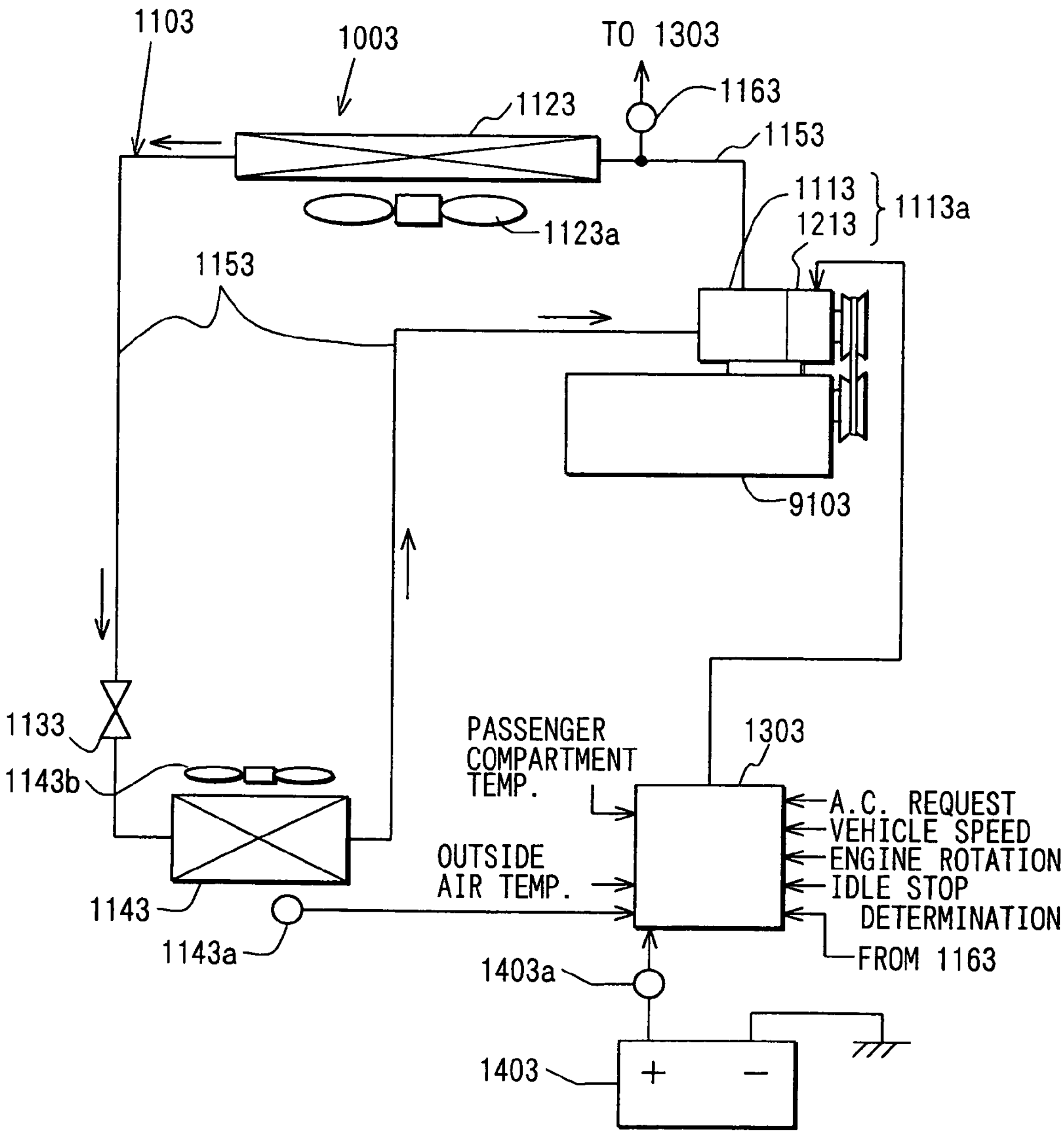


FIG. 61


1103
1003
TO 1303
1163
1123
1153
1123a
1113
1213
1113a
1153
9103
1133
1143b
PASSENGER COMPARTMENT TEMP.
1303
A.C. REQUEST
VEHICLE SPEED
ENGINE ROTATION
IDLE STOP DETERMINATION
FROM 1163
OUTSIDE AIR TEMP.
1143
1143a
1403a
1403
+
−

AIR-CONDITIONING APPARATUS INCLUDING MOTOR-DRIVEN COMPRESSOR FOR IDLE STOPPING VEHICLES

CROSS REFERENCES TO RELATED APPLICATIONS

This is a divisional of U.S. application Ser. No. 10/132,764, which was filed on Apr. 26, 2002, now U.S. Pat. No. 6,981,544. This application relates to and incorporates by reference the following Japanese patent applications: 2001-131605, filed on Apr. 27, 2001; 2001-161921, filed on May 30, 2001; 2001-206890, filed on Jul. 6, 2001; 2001-322607, filed on Oct. 19, 2001; 2001-345038, filed on Nov. 9, 2001; and 2002-22723, filed on Jan. 31, 2002.

BACKGROUND OF THE INVENTION

The present invention relates to a vehicle air conditioning system for use in a so-called idle-stop vehicle having an engine stopped when the vehicle in a running state comes to a temporary halt.

So-called idle-stop vehicles have recently been introduced for saving fuel. Since, such an idle-stop vehicle stops the engine when the vehicle comes to a temporary halt, the air-conditioning compressor, which is driven by the engine, and a mechanical pump for a heating unit are stopped, and the air-conditioning system does not operate while the engine is stopped.

To prevent this, Japanese Patent Laid-Open Publication No. 2000-130323 discloses technology associated with a hybrid compressor provided integrally with an electric motor. When the engine is stopped, the compressor is operated by the electric motor to compress refrigerant, thereby operating a cooling unit.

On the other hand, Japanese Patent Laid-Open Publication (JP-A) No. (Hei) 9-277818 discloses the provision of an electric pump and a bypass channel in the cooling water circuit of a heating unit. When the engine is stopped, the heating unit is operated by driving the electric pump.

These devices allow the operation of either the cooling unit or the heating unit when the engine is stopped.

However, the devices disclosed in these publications are insufficient if the air-conditioning system is assumed to perform both cooling and heating functions throughout the year in an idle-stop vehicle.

If the devices are used in combination, both the cooling and heating functions can be performed satisfactorily. However, this increases the number of parts, resulting in a complicated and very costly system.

In another proposed solution to this problem, Japanese Patent Laid-Open Publication (JP-A) No. 2000-127753 discloses the provision of an electric compressor driven by a battery-powered motor in addition to the compressor of a cooling unit. During a stoppage of the engine, the cooling unit is operated by the electric compressor so that cooling is performed, regardless of whether the engine is running or at rest.

The motor is activated when the engine comes to a stop, and continues operating while the engine is stopped. Consequently, the battery may be overtaxed, which may result in insufficient battery strength the next time the engine is started, and the life of the motor may be too short.

In a further proposed solution, as described in Japanese Patent Laid-Open Publication (JP-A) No. 2000-80348, some systems reduce the load on the motor that operates when the vehicle and engine are stopped. Specifically, the operation of the motor is combined with controls such that an air mix door is fixed to a full cool position or fixed in an inside air circulation mode, and an evaporator anti-freezing temperature is raised by a predetermined value.

Also, when a vehicle stops temporarily, since the compressor intended for the cooling unit and the mechanical pump intended for the heating unit are also stopped, the air-conditioning systems do not operate while the engines are stopped.

As means for solving this problem, Japanese unexamined patent publication (JP-A) No. 2000-127753 has disclosed the provision of an electric compressor driven by a battery-powered motor to assist the compressor of a cooling unit. During an engine stoppage, the cooling unit is operated by the electric compressor so that the cooling function is performed regardless of whether the engine is running or at rest.

The motor is activated when the engine comes to a stop, and continues operating while the engine is stopped. Consequently, the battery capacity may fall to the extent that there is insufficient battery capacity the next time the engine is started, and the life of the motor is in doubt.

As described in Japanese Patent Laid-Open Publication No. 2000-80348, some proposals reduce the load on the motor. Specifically, the operation of the motor and the control of other devices is such that, when the motor operates, an air mix door is fixed to a full cool position or fixed in an inside air circulation mode, and such that an evaporator anti-freezing temperature is raised by a predetermined value.

As a consequence, the compressing load is reduced in comparison to that when the engine is running, so that the compressor consumes less power, which reduces the load on the motor. The battery power supply is thus prevented from extraordinary draining.

While such proposals can reduce the power used by the compressor under steady use conditions, variations are naturally expected in the cooling conditions, the frequency of engine stops during moving, and so forth depending on the passengers. Operating the compressor accordingly even under such unsteady conditions as higher cooling loads and longer engine stop times may overtax the battery. In short, the battery may be exhausted.

SUMMARY OF THE INVENTION

In view of the foregoing, it is therefore an object of the present invention to provide a low-cost vehicle air-conditioning system capable of performing both cooling and heating functions throughout the year in an idle-stop vehicle.

It is another object of the present invention to provide a vehicle air-conditioning system capable of providing satisfactory air-conditioning performance during an engine stoppage while avoiding insufficient battery capacity and a short motor life.

It is another object of the present invention to provide a vehicle cooling system that can deliver an average cooling performance at each individual engine stop and thus prevent a dead battery due to excessive motor operations.

In one aspect, the invention is a vehicle air-conditioning apparatus for use in a vehicle in which an engine for driving the vehicle is stopped when the vehicle comes to a temporary halt from a running state. The apparatus includes a cooling unit for cooling air by compressing a refrigerant with a compressor, which is driven by torque from the engine; condensing the compressed refrigerant; expanding the condensed refrigerant; and evaporating the expanded refrigerant. The apparatus further includes a heating unit for heating the air by using, as a heat source, cooling water that is circulated by a mechanical pump, which is driven by torque from the engine. A compression unit compresses the refrigerant. A pump unit circulates the cooling water. A motor drives both the compression unit and the pump unit. The apparatus further includes a control unit for controlling operation of the motor, wherein the control unit causes the motor to operate the compression unit if the engine is stopped when the cooling unit is operating and causes the motor to operate the pump unit if the engine is stopped when the heating unit is operating.

In another aspect, the invention is a vehicle cooling apparatus for use in a vehicle having an engine that is stopped when the vehicle comes to a temporary halt from a running state. The apparatus includes a battery, a motor, a cooling unit, which includes a compressor, which can be driven by the motor when the engine is stopped temporarily to perform air conditioning of a passenger compartment of the vehicle. The apparatus further includes a pump unit for circulating cooling fluid that cools the engine, and the pump unit is driven by the motor. The apparatus further includes means for controlling the operation of the motor such that the pump unit is driven by the motor if there is a demand for heating the passenger compartment and the compressor is driven by the motor if there is a demand for cooling the passenger compartment.

In another aspect, the invention is a vehicle air-conditioning apparatus for use in a vehicle having an engine that is stopped when the vehicle comes to a temporary halt from a running state the apparatus includes an engine-driven apparatus, and the engine-driven apparatus is at least one of a cooling unit for cooling air by compressing a refrigerant with a compressor, which is driven by torque from the engine, and then subjecting the refrigerant to condensation, expansion, and evaporation, and a heating unit for heating air with engine cooling water as a heat source, wherein the cooling water is circulated by a mechanical pump powered by torque from the engine. The apparatus further includes a battery-powered apparatus, wherein the battery-powered apparatus is driven by a motor, which is powered by a battery, and is a compressor unit for compressing the refrigerant if the engine driven apparatus is the cooling unit and is a pump unit for circulating the cooling water if the engine driven apparatus is the heating unit. The apparatus further includes a control unit for controlling the operation of the motor, such that the motor is driven by the control unit to operate the battery-powered apparatus when the engine comes to a stop, while the battery powered apparatus is in operation, and the control unit causes the motor to operate, while the engine is stopped, to maintain, within a predetermined range, an air-conditioning state produced by the engine-driven apparatus prior to the stoppage of the engine.

In another aspect, the invention is essentially a vehicle cooling apparatus for use in a vehicle having an engine that is stopped when the vehicle comes to a temporary halt from a running state. The apparatus includes a refrigeration unit including a compressor apparatus, which can be driven by a battery-powered motor and by torque of the engine, and the compressor apparatus includes a single compressor driven by both the engine and the motor or a first compressor driven by the engine and a second compressor driven by the motor. The apparatus further includes a control unit for controlling the operation of the motor, wherein the motor is operated by the control unit to drive the compressor apparatus when the engine is stopped while the refrigeration unit is in operation. The control unit operates the motor so that the cumulative operating time of the motor per vehicle halt falls within a predetermined time period.

BRIEF DESCRIPTION OF THE DRAWINGS

These and other objectives and advantages of the invention will become apparent from the following description with reference to the accompanying drawings, wherein:

FIG. **1** is a schematic view showing an overall structure of a first embodiment according to the present invention;

FIG. **2** is a diagram of a circuit connecting a control unit to a motor shown in FIG. **1**;

FIG. **3** is a cross-sectional view of a whole electric compressor-pump shown in FIG. **1**;

FIG. **4** is a flow chart illustrating control over the operation of the electric compressor-pump;

FIG. **5** is a circuit diagram showing a variation of the electric compressor-pump shown in FIG. **2**;

FIG. **6** is a cross-sectional view of a compression unit according to a first variation of a second embodiment of the present invention;

FIG. **7**A is a cross-sectional view of a compression unit according to a second variation of the second embodiment;

FIG. **7**B is a cross sectional view taken along the line **7**B-**7**B in FIG. **7**A;

FIG. **8**A is a cross-sectional view of a compression unit according to a third variation of the second embodiment;

FIG. **8**B is a cross-sectional view taken along the line **8**B-**8**B of FIG. **8**A;

FIG. **9** is a cross-sectional view of a compression unit according to a fourth variation of the second embodiment;

FIG. **10**A is a cross-sectional view of a compression unit according to a fifth variation of a third embodiment;

FIG. **10**B is a cross-sectional view taken along the line **10**BC-**10**BC of FIG. **10**A during rotation in a forward direction;

FIG. **10**C is a cross-sectional view taken along the line **10**BC-**10**BC of FIG. **10**A during rotation in a rearward direction;

FIG. **11** is a cross-sectional view of an electric compressor-pump according to a fourth embodiment of the present invention;

FIG. **12** is a cross-sectional view taken along the line **12**-**12** of FIG. **11**;

FIG. **13**A is a cross-sectional view taken along the line **13**A-**13**A of FIG. **12**;

FIG. **13**B is a cross-sectional view taken along the line **13**B-**13**B of FIG. **12**;

FIG. **14**A is a cross-sectional view showing the operating state of the pump unit shown in FIG. **11** when rotating in the forward direction;

FIG. **14**B is a cross-sectional view showing the operating state of the pump unit shown in FIG. **11** when rotating in the rearward direction;

FIG. **15** is a flow chart illustrating part of control over the operation of the electric compressor-pump shown in FIG. **11**;

FIG. **16** is a schematic diagram showing the overall configuration of a fifth embodiment of the present invention;

FIG. **17** is a flowchart showing the operation control of the electric compressor-pump in FIG. **16**;

FIGS. **18**A-**18**E are timing charts showing the engine operation, the motor operation, an evaporator downstream temperature Te (the temperature downstream of the evaporator), battery capacity C, and a motor temperature Tm during the control procedure of FIG. **17**, respectively;

FIG. **19** is part of a flowchart showing the operation control procedure of an electric compressor-pump according to a sixth embodiment;

FIG. **20** is a schematic diagram showing the overall configuration of a seventh embodiment;

FIG. **21** is a flowchart showing the control procedure of the electric compressor of FIG. **20**;

FIGS. **22**A-**22**E are timing charts for showing a vehicle speed, an evaporator air flow rate, the evaporator downstream temperature Te, the amount of the refrigerant in the condenser, and the operation of the motor during the control procedure of the seventh embodiment, respectively;

FIGS. **23**A-**23**F are timing charts for showing the vehicle speed, the evaporator air flow rate, a valve opening degree, the evaporator downstream temperature Te, the amount of the refrigerant in the condenser, and the operation of the motor during the control procedure according to an eighth embodiment, respectively;

FIGS. **24**A-**24**D are timing charts for showing the vehicle speed, a condenser air flow rate, the evaporator downstream temperature Te, and the operation of the motor during the control procedure according to a ninth embodiment, respectively;

FIGS. **25**A-**25**D are timing charts for showing the vehicle speed, a displacement, the evaporator downstream temperature Te, and the operation of the motor during the control procedure of a variable displacement type compressor according to a tenth embodiment, respectively;

FIGS. **26**A-**26**D are timing charts for showing the vehicle speed, the displacement, the evaporator downstream temperature Te, and the operation of the motor during the control procedure of an ON/OFF control type compressor according to the tenth embodiment, respectively;

FIG. **27** is a schematic diagram showing the overall configuration of an eleventh embodiment of the present invention;

FIG. **28** is a flowchart showing the control procedure of the compressor and the electric compressor in FIG. **27**;

FIGS. **29**A-**29**D are timing charts for showing the vehicle speed, an accelerator throttle opening, the operation of the compressor, and the operation of the motor during the control procedure of FIG. **28**, respectively;

FIG. **30** is a flowchart showing the control procedure of a compressor and an electric compressor according to a twelfth embodiment;

FIGS. **31**A-**31**E are timing charts for showing the vehicle speed, a determination of compressor stoppage due to acceleration, the operation of the compressor, the operation of the motor, and the evaporator downstream temperature during the control procedure of FIG. **30**, respectively;

FIGS. **32**A-**32**E are timing charts for showing the vehicle speed, the determination of a compressor stoppage due to acceleration, the operation of the compressor, the operation of the motor, and the evaporator downstream temperature during the control procedure according to a modified example of the twelfth embodiment, respectively;

FIG. **33** is a schematic diagram showing a control unit according to a thirteenth embodiment;

FIG. **34** is a flowchart showing the control procedure of the electric compressor according to the thirteenth embodiment;

FIGS. **35**A-**35**E are timing charts for showing the working load on the engine, a cooling water temperature, the operation of the compressor, the operation of the motor, and the evaporator downstream temperature during the control procedure according to the thirteenth embodiment, respectively;

FIG. **36** is a flowchart showing the control procedure of an electric compressor according to a fourteenth embodiment;

FIGS. **37**A-**37**D are timing charts for showing the vehicle speed, the operation of the compressor, the operation of the motor, and the evaporator downstream temperature during the control procedure according to the fourteenth embodiment, respectively;

FIG. **38** is a flowchart showing the control procedure of an electric compressor according to a fifteenth embodiment;

FIGS. **39**A-**39**E are timing charts for showing the vehicle speed, the engine speed, the operation of the compressor, the operation of the motor, and the evaporator downstream temperature during the control procedure according to the fifteenth embodiment, respectively;

FIG. **40** is a schematic diagram showing the overall configuration of a further embodiment;

FIG. **41** is a schematic diagram showing the overall configuration of a sixteenth embodiment of the present invention;

FIGS. **42**A and **42**B are graphs showing the relationship between outside air temperature and a first predetermined time in a first pattern and a second pattern, respectively;

FIGS. **43**A and **43**B are graphs showing the relationship between the outside air temperature and a second predetermined time in a first pattern and a second pattern, respectively;

FIG. **44** is a flowchart showing the control procedure for the motor of FIG. **41**;

FIGS. **45**A-**45**D are timing charts showing vehicle speed, discharge pressure, the ON/OFF state of the motor, and a motor current under the control procedure of FIG. **41**, respectively;

FIG. **46** is a schematic diagram showing the partial configuration of a seventeenth embodiment of the present invention;

FIG. **47** is a flowchart showing the control procedure of the motor of FIG. **46**;

FIGS. **48**A-**48**F are timing charts showing vehicle speed, engine speed, discharge pressure, the ON/OFF state of the motor, battery capacity, and an evaporator temperature under the control procedure of FIG. **46**, respectively;

FIGS. **49**A-**49**E are timing charts showing vehicle speed, the engine speed, the discharge pressure, the ON/OFF state of the motor, and the evaporator temperature under the control procedure of a eighteenth embodiment of the present invention, respectively;

FIGS. **50**A and **50**B are graphs showing the relationship between the outside air temperature and first, second, and third predetermined temperatures in a first pattern and a second pattern, respectively;

FIG. **51** is a flowchart showing the control procedure of a motor of a nineteenth embodiment;

FIGS. **52**A-**52**E are timing charts showing vehicle speed, the engine speed, the discharge pressure, the ON/OFF state of the motor, and the evaporator temperature of the nineteenth embodiment, respectively;

FIG. **53** is a flowchart showing the control procedure of a motor of a twentieth embodiment of the present invention;

FIGS. **54**A-**54**F are timing charts showing vehicle speed, the engine speed, the discharge pressure, the ON/OFF state of the motor, the ON/OFF state of a starter, and a battery voltage of the twentieth embodiment, respectively;

FIGS. **55**A-**55**C are timing charts showing vehicle speed, the discharge pressure, and the ON/OFF state of the motor of a twenty-first embodiment of the present invention, respectively;

FIGS. **56**A-**56**D are timing charts showing vehicle speed, the ON/OFF state of the compressor, the discharge pressure, and the ON/OFF state of the motor of a twenty-second embodiment of the present invention, respectively;

FIGS. **57**A-**57**D are timing charts showing vehicle speed, the ON/OFF state of the compressor, the discharge pressure, and the ON/OFF state of the motor of an twenty-third embodiment of the present invention, respectively;

FIGS. **58**A-**58**D are timing charts showing vehicle speed, the air flow rate of a fan, the discharge pressure, and the ON/OFF state of the motor of a twenty-fourth embodiment of the present invention, respectively;

FIGS. **59**A-**59**D are timing charts showing vehicle speed, the engine speed, the discharge pressure, and the ON/OFF state of the motor in a first pattern of a twenty-fifth embodiment of the present invention, respectively;

FIGS. **60**A-**60**D are timing charts showing vehicle speed, the engine speed, the discharge pressure, and the ON/OFF state of the motor in a second pattern of the twenty-fifth embodiment, respectively; and

FIG. **61** is a schematic diagram showing the overall configuration of a further embodiment.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENTS

First Embodiment

A specific structure of a first embodiment according to the present invention will be described herein below with reference to FIGS. **1** to **4**. A vehicle air-conditioning system **100** is for use in a so-called idle-stop vehicle having an engine **10** stopped when the vehicle in a running state comes to a temporary halt. The air-conditioning system **100** is composed of a cooling unit **110**, a heating unit **120**, a control unit **130**, and an electric compressor-pump **200**.

The cooling unit **110** forms a known refrigerating cycle in which a compressor **111** for compressing a refrigerant under high-temperature and high-pressure conditions, a condenser **112** for condensing and liquefying the compressed refrigerant, an expansion valve **113** for adiabatically expanding the liquefied refrigerant, and an evaporator **114** for evaporating the expanded refrigerant and for cooling air by using latent heat resulting from evaporation are connected in succession with a refrigerant pipe **115**.

The compressor **111** is configured to be operated upon receiving a driving force from the engine **10** transmitted via a pulley and a pulley belt.

The heating unit **120** is a known unit in which a mechanical pump **11** provided in the engine **11** and a heater **121** for heating air by using cooling water for cooling the engine **10** as a heat source are connected to each other with a cooling water pipe **123**. A water valve **122** for adjusting the flow rate of the cooling water is provided on the cooling-water in-coming side of the heater **121**.

The mechanical pump **11** is operated upon receiving the driving force of the engine **10** and circulates the cooling water in the heater **121**.

The cooling water for the engine **10** is cooled by a radiator **124** provided in the cooling water pipe **123***a* to perform temperature control.

The control unit **130** controls the operation of the motor **210** of an electric compressor-pump **200**, which will be described later. The control unit **130** operates the motor **210** based on signals from various sensors not shown, i.e., a vehicle speed signal, an engine speed signal, an evaporator rear temperature signal, an in-car temperature, and an A/C request signal, and the control unit **130** controls the direction of rotation of the motor **210**.

Specifically, as shown in FIG. **2**, a circuit connecting the control unit **130** to the motor **210** by using a transistor **131** is provided. The circuit has an energization pattern indicated by the solid or broken lines to effect control of the rotation of the motor **210** in one direction or in the opposite direction.

Referring to FIG. **3**, a structure of the electric compressor-pump **200** as a principal portion of the present invention will be described.

The electric compressor-pump **200** is composed of a compression unit **220** and a pump unit **230** provided integrally on both end portions of the rotating shaft **211** of the motor **210**.

The motor **210** is a known DC motor having a rotor **213** and a stator **217** provided within a motor housing **212** such that the rotating shaft **211** extending through the rotor **213** is supported by bearings **214***a* and **214***b*. The motor **210** is driven to rotate with power supplied from a brush **216** to a commutator **215**.

The compression unit **220** is formed as a rotary compression unit which performs a normal compressing operation only during rotation in one direction (hereinafter referred to as a forward direction). More specifically, the compression unit **220** is formed as a scroll compression unit in the present embodiment.

The compression unit **220** is composed of a fixed scroll **223**, a movable scroll **224**, and an eccentric shaft **225** provided in a front housing **226** and a rear housing **227**. The eccentric shaft **225** is configured to be connected to one end portion of the rotating shaft **211** of the motor **210** via a unidirectional clutch **260**. The unidirectional clutch **260** allows the forward torque of the motor **210** to be transmitted and used to drive the movable scroll **224**, while preventing the torque from being transmitted during rotation in the rearward direction.

When the motor **210** is rotating in the forward direction, therefore, the movable scroll **224** revolves by the unidirectional clutch **260** and the eccentric shaft **225** such that the refrigerant flowing, through an inlet **228**, into a compression chamber **221** formed between itself and the opposing fixed scroll **223** is compressed (hereinafter referred to as the normal compressing operation) and discharged through an outlet **229**. On the other hand, the movable scroll **224** does not revolve when the motor **210** is rotating in the rearward direction so that the compressing operation is not performed (hereinafter referred to as a non-operating state).

To prevent the refrigerant in the compression unit **220** from leaking toward the motor **210**, a shaft sealing unit **240** is provided between the motor **210** and the compression unit **220**, specifically between the rotating shaft **211** and the eccentric shaft **225**.

The pump unit **230** is composed of a centrifugal impeller **231** provided in a pump housing **232** formed with an inlet **233** and an outlet **234** each for the cooling water. The pump unit **230** is configured to be located at the other end of the rotating shaft **211** of the motor **210**.

The shaft **235** of the impeller **231** and the rotating shaft **211** of the motor **210** are provided with respective magnetic couplings **250** so that the torque of the motor **210** is transmitted by the magnetic couplings **250** to the shaft **235** to rotate the impeller **231**.

The blade of the impeller **231** and the pump housing **232** are configured to pump the cooling water (hereinafter referred to as a normal pumping operation) during rotation in the direction opposite to the forward direction in which the unidirectional clutch **260** causes the compression unit **220** to perform the normal compressing operation. Due to the configurations of the blade and the pump housing **232**, the impeller **231** is designed to rotate idly (hereinafter referred to as the non-operating state) during rotation in the forward direction.

A separator **236** is provided between the magnetic couplings **250** to prevent the cooling water in the pump unit **230** from flowing toward the motor **210**.

The motor **210**, the compression unit **220**, and the pump unit **230** constitute the integral electric compressor pump **200** with the respective housings **212**, **226**, **227**, and **232** thereof being connected to each other.

As shown in FIG. **1**, the compression unit **220** of the electric compressor-pump **200** is located in the cooling unit **110** such that the compression unit **220** is connected parallel to the compressor **111** via the refrigerant pipe **115***a*, specifically, such that the compression unit **220** is connected between the upstream side of the condenser **112** and the downstream side of the evaporator **114**.

The pump unit **230** is located in the heating unit **120** such that the pump unit **230** is in series with the mechanical pump **11**, specifically, such that the pump unit **230** is positioned between the mechanical pump **11** and the water valve **122**.

The cooling water pipe **123** is provided with a bypass channel **270** along which the cooling water from the mechanical pump **11** passes through the pump unit **230** when the motor **210** and the pump unit **230** are in the non-operating state. The bypass channel **270** is provided with a check valve **280** for preventing a short circuit from the outlet **234** to the inlet **233** during the normal operation of the pump unit **230**.

A description will of the operation of the air-conditioning system follows. When the vehicle is running, i.e., when the engine **10** is operated, the cooling unit **110** and the heating unit **120** operate like those of prior art systems. In the cooling unit **110**, the compressor **111** operates upon receiving torque from the engine **10** and compresses the refrigerant. The compressed refrigerant is condensed to be liquefied in the condenser **112**, adiabatically expanded in the expansion valve **113**, and then evaporated in the evaporator **114**. By using latent heat resulting from evaporation, the air passing through the evaporator **114** is cooled.

In the heating unit **120**, the mechanical pump **11** is operated upon receiving the driving force from the engine **10** to open the water valve **122** and circulate the cooling water in the heater **121**. By using the cooling water as a heat source, the air passing through the heater **121** is heated.

Since the air-conditioning system is used in an idle-stop vehicle, the engine **10** is stopped when the vehicle comes to a temporary halt so that the compressor **111** and the mechanical pump **11**, each using the engine **10** as a driving source, do not operate. The electric compressor-pump **200** is configured to be operated at this time.

The electric compressor-pump **200** is controlled by the control unit **130**. The control operation will be described in detail with reference to the flow chart of FIG. **4**.

First, in step S**10**, the motor **210** is halted. In step S**20**, it is determined whether or not there is a request for a cooling or heating operation based on an A.C. request signal. If there is no request, the process returns to step S**10** and the halted state of the motor **210** is maintained. If there is a request for a cooling or heating operation, it is determined in step S**30** whether the request is for a cooling operation or a heating operation.

If it is determined in step S**30** that the request is for a cooling operation, the whole process advances to step S**40** where it is determined whether or not the vehicle is halted based on a vehicle speed signal. If the vehicle is halted, it is determined in step S**50** whether or not the engine **10** is stopped. If it is determined that the engine **10** is stopped, the motor **210** is operated to rotate in the forward direction (the direction of rotation which operates the unidirectional clutch **260**) in step S**60** such that the compression unit **220** performs the normal compressing operation (at this time, the pump **230** is in the non-operating state). Thereafter, step S**60** is sustained while the engine **10** is halted so that the compression unit **220** is operated continuously.

If it is determined in step S**40** that the vehicle is not halted (is running) or if it is determined in step S**50** that the engine **10** is not stopped (is rotating), the process returns to step S**10** so that the motor **210** is halted and the compression unit **220** is halted.

If it is determined in step S**30** that the request is for a heating operation, it is determined in steps S**70** and S**80** whether or not the vehicle and the engine **10** are halted, respectively, as in steps S**40** and S**50**. If the vehicle is at a halt and the engine **10** is at a stop, the motor **210** is operated in step S**90** in the direction (the direction of rotation which brings the unidirectional clutch **260** into the non-operating state) opposite to the forward direction so that the pump unit **230** performs the normal pumping operation (at this time, the compression unit **220** is in the non-operating state). Thereafter, step S**90** is sustained while the engine **10** is stopped so that the pump unit **230** is operated continuously.

If it is determined in step S**70** or S**80** that the vehicle is not halted (is running) or that the engine **10** is not stopped (is rotating), the process returns to step S**10** so that the motor **210** is halted and the pump **230** is halted.

A description of the effects of the invention follows. If the engine **10** is stopped while the cooling unit **110** is operating, the motor **210** is controlled to rotate in the forward direction such that the compression unit **220** performs the normal compressing operation. This allows the compression unit **220** to compress the refrigerant in place of the compressor **111** used originally to compress the refrigerant and allows the cooling function to be performed continuously.

If the engine **10** is stopped while the heating unit **120** is operating, the motor **210** is controlled to rotate in the rearward direction such that the pump unit **230** performs the normal pumping operation. This allows the pump unit **230** to circulate the cooling water in place of the mechanical pump **11** used originally to circulate the cooling water and allows the heating operation to be performed continuously. In short, both the cooling and heating functions can be performed reliably throughout the year even when the engine **10** is stopped.

What results is a compact and low-cost air-conditioning system that can be used selectively for the heating and cooling operations by selecting the compression unit **220** or the pump unit **230** by changing the direction of the single motor **210**.

Since the compression unit **220** is located in the cooling unit **110** in parallel relation to the compressor **111**, the compression unit **220** and the compressor **111** do not allow the refrigerant condensed under a high pressure to flow from one to the other. This obviates the need to excessively increase the pressure strength of the inlet port of each of the compression unit **220** and the compressor **111**, thereby preventing increased cost.

Since the pump unit **230** is located in the heating device **120** in series relation to the mechanical pump **11**, the piping is not complicated by incorporating the pump unit **230** into the cooling water pipe **123** of the heating unit **120**. This provides a constant supply of cooling water flowing in the engine **10** to the heater **121** and prevents a reduction in heating ability.

Since the compression unit **220** and the pump unit **230** can be used selectively for cooling and heating operations by changing the direction of rotation of the motor **210**, a low-cost air-conditioning system is provided.

Since the compression unit **220** and the pump unit **230** are provided at the both ends of the rotating shaft **211** of the motor **210**, the number of the shaft sealing units **240** for preventing the refrigerant and the cooling water from leaking through the rotating shaft **211** can be minimized. In short, it is sufficient to provide the shaft sealing unit **123** for the refrigerant between the compression unit **220** and the motor **210** and to provide the shaft sealing unit **241** for the cooling water between the pump unit **230** and the motor **210**. In particular, the shaft sealing unit **241** between the pump unit **230** and the motor **210** can be omitted in the present embodiment since the pump unit **230** is operated to rotate by the motor **210** via the magnetic couplings **250**.

Since the unidirectional clutch **260** is provided between the compression unit **220** and the motor **210**, the normal operation performed by the compression unit **220** or the pump unit **230** when the motor **210** is rotating in the forward or rearward direction can be used selectively.

The circuit for connecting the control unit **130** to the motor **210** may use a relay **132** as shown in FIG. **5** instead of using the transistor **131** shown in FIG. **2**. The direction of rotation of the motor **210** can be controlled as indicated by the solid lines and the broken lines in the drawings.

The provision of the bypass channel **270** may be omitted depending on the load (resistance of water flow) received by the mechanical pump **11** when the pump **230** is in the non-operating state.

Second Embodiment

A second embodiment of the present invention shows variations of the structure using, as the compression unit **220**, the rotary compression unit **220** which performs the normal compressing operation only during rotation in the forward direction. The second embodiment omits the provision of the unidirectional clutch **260**.

Specifically, as shown in FIG. **6**, the second embodiment is obtained by taking the unidirectional clutch **260** away from the scroll compression unit **220** described in the first embodiment. The second embodiment operates the movable scroll **224** of the compression unit **220** via the rotational shaft **211** and the eccentric shaft **225**.

In the present embodiment, the compression unit **220** performs the normal compressing operation when the motor **210** is rotating in the forward direction, while the pump **230** is in the non-operating state. When the motor **210** is rotating in the rearward direction, the pump unit **230** performs the normal pumping operation, while the compression unit **220** is in the non-operating state. Specifically, even if the rotary compression unit **220** having no suction valve is operated to rotate in the rearward direction, it does not perform the refrigerant compressing operation, and a vacuum pump effect is exerted in the compression chamber **221** so that the power consumed by the motor **210** is significantly reduced. This allows selective use of the normal compressing operation performed by the compression unit **220** during rotation in the forward direction and the non-operating state of the compression unit **220** during rotation in the rearward direction and obviates the necessity to use the unidirectional clutch **260**.

Likewise, a structure using a rolling piston compression unit **220** in which the refrigerant is compressed by using a revolving rotor **220***a* and a vane **220***b* as shown in FIG. **7** and a structure using a rotary vane compression unit **220** having a rotor **220***a* and a plurality of vanes **220***b* as shown in FIG. **8** may also be used as other variations of the structure using the rotary compression unit **220**.

In the pump unit **230** also, the impeller **231** may be connected directly to the shaft **211** of the motor **210** with the shaft sealing unit **241** located as shown in FIG. **9**.

The bypass channel **270** extending through the pump unit **230** may also be formed integrally with the pump unit **230** as a bypass channel **271** provided with a bypass valve **281**.

Third Embodiment

FIG. **10** shows a third embodiment of the present invention. The third embodiment is obtained by providing the scroll compression unit **220** according to the second embodiment shown in FIG. **6** with a releasing mechanism for releasing the compression chamber **221** when the motor **210** is rotating in the rearward direction, i.e., a radius compensating mechanism **222**.

The radius compensating mechanism **222** is obtained by forming a tip portion of the eccentric shaft **225** into a plate having a width across flat with intervention of a bush **220***d*. When the motor **210** is rotating in the forward direction, the radius compensating mechanism **222** functions to increase the radius of revolution of the movable scroll **224** in the direction a of the width across flat under a counterforce F**1** resulting from the compression of the refrigerant, as shown in FIG. **10**B, thereby improving the seal with respect to the fixed scroll **223**.

When the motor **210** is rotating in the rearward direction, however, the radius compensating mechanism functions to reduce the radius of revolution of the movable scroll **224** in the direction b under a frictional force F**2** resulting from the revolution of the movable scroll **224**, as shown in FIG. **10***c*, thereby forming an interscroll space **220***e* between the movable scroll **224** and the fixed scroll **223**. This further reduces a loss in the compression unit **220** when the motor **210** is rotating in the rearward direction.

If either one of the fixed scroll **223** and the movable scroll **224** of the scroll compression unit **220** is made of a resin, vibration between the fixed and movable scrolls **223** and **224** during rotation in the rearward direction and noise resulting from the interference between the scrolls is prevented.

Fourth Embodiment

FIGS. **11** to **15** show a fourth embodiment of the present invention. In contrast to the first embodiment, the fourth embodiment brings each of the compression unit **220** and the pump unit **230** into the operating state when each of the cooling unit **110** and the heating unit **120** is operating, thereby performing a dehumidifying heating function. In addition, the fourth embodiment has changed the position of the shaft sealing unit **240** to reduce power consumed by the motor **210**.

First, as shown in FIGS. **11** and **12**, the pump unit **230** of the electric compressor-pump **200** is configured to perform the normal ejecting operation when the motor **210** is rotating in each of the forward and rearward directions. Specifically, the pump housing **232** is formed with two outlets **234***a* and **234***b*. A contra-flow preventing ball **237** and a stopper **238** having a hole for stopping the movement of the ball **237** in the direction in which the cooling water flows and allowing the passage of the cooling water are located in each of the outlets **234***a* and **234***b*. In the pump housing **232**, the radial clearance between the housing **232** and the impeller **231** is circumferentially uniform, while the axial clearance between the housing and the impeller **231** gradually increases in a direction from the counter-outlet side toward the outlet side as shown by a, b, and c in FIGS. **13**A and **13**B, so that the cooling water flows more smoothly from the counter-outlet side toward the outlet side.

When the motor **210** is rotating in the forward direction, as shown in FIG. **14**A, the ball **237** shown on the right-hand portion of the drawing is pressed toward the stopper **238** under positive pressure (ejection pressure) of the flowing cooling water, while the ball **237** on the left-hand portion of the drawing is attracted toward the impeller **231** under negative pressure indicated by the broken line to block the outlet **234***b*, so that the cooling water is ejected from the outlet **234***a*. When the motor **210** is rotating in the rearward direction, the direction in which the cooling water flows is reversed so that the cooling water is ejected from the outlet **234***b*. Thus, the pump unit **230** is configured to perform the normal pumping operation when the motor **210** is rotating in each of the forward and rearward directions. Alternatively, the outlet **234** of the pump housing **232** may also be composed of one outlet **234** extending radially from the center of the impeller **231**, allowing a reduction in pump efficiency without providing the balls **237** and the stoppers **238**.

On the other hand, the unidirectional clutch **260** is located between the motor **210** and the compression unit **220** (shown herein as a rolling piston compression unit), similarly to the first embodiment, so that the compression unit **220** performs the normal compressing operation when the motor **210** is rotating only in the forward direction (the compression unit **220** is in the non-operating state when the motor **210** is rotating in the rearward direction). The shaft sealing unit **240** for preventing the leakage of the refrigerant is provided at a position closer to the compression unit **220**, specifically on the eccentric shaft **225** of the compression unit **220**.

The operation of the electric compressor-pump **200** is controlled based on the flow chart shown in FIG. **15**. The flow chart shown in FIG. **15** is basically the same as that (FIG. **4**) described in the first embodiment except that step S**31** is provided in place of step S**30** and step **61** is provided in place of step S**60**. Specifically, conditions for determination when each of the cooling unit **110** and the heating unit **120** is operating are added in step S**31**. If each of the cooling and heating units **110** and **120** is operating, the motor **210** is controlled to rotate in the forward direction in step S**61** such that each of the compression unit **220** and the pump unit **230** performs the normal operation.

By changing the direction of rotation of the motor **210**, the compression unit **220** and the pump unit **230** can be used selectively for each of the cooling and heating operations. When each of the cooling unit **110** and the heating unit **120** is operating, the motor **210** is controlled to operate in the forward direction, thereby allowing each of the compression unit **220** and the pump unit **230** to operate. What results is an air-conditioning system capable of performing a dehumidifying heating operation.

Since the shaft sealing unit **240** is provided at the position closer to the compressor unit **220** (on the eccentric shaft **225**) in the structure having the unidirectional clutch **260** provided between the compressor unit **220** and the motor **210**, the unidirectional clutch **260** is disconnected to bring the compression unit **220** into the non-operating state. When the pump unit **230** is to be operated, the motor **210** is not subjected to the sliding resistance of the shaft sealing unit **240**, so that power consumed by the motor **210** is reduced.

Other Variations

Although, in each of the first to fourth embodiments, the operations of the compression unit **220** and the pump unit **230** are used selectively by controlling the direction of rotation of the motor **210**, it is also possible to selectively use the operations of the compression unit **220** and the pump unit **230** by providing respective clutch mechanisms between the compression unit **220** and the motor **210** and between the pump unit **230** and the motor **210** and intermittently controlling the clutch mechanisms by using the control unit **130**. In this variation, the motor **210** need not rotate in two directions; one direction is sufficient.

This ensures selective use of the compression unit **220** and the pump unit **230**. It is also possible to simultaneously operate the compression unit **220** and the pump unit **230** when each of the cooling unit **110** and the heating unit **120** is operating. The result is an air-conditioning system capable of performing a dehumidifying heating operation.

Each of the compression unit **220** and the pump unit **230** may be provided at the same end of the rotating shaft **211** of the motor **210**.

Fifth Embodiment

FIGS. **16**-**18**E show a fifth embodiment of the present invention.

With reference to FIG. **16**, a vehicle air-conditioning system **1002** is used in a so-called idle-stop vehicle, which has an engine **9102** that is stopped when the vehicle comes to a temporary halt. The vehicle air-conditioning system **1002** includes a cooling unit **1102**, a heating unit **1202**, an electric compressor-pump **2002**, and a control unit **130**.

The cooling unit **1102** performs a known refrigeration cycle. The cooling unit **1102** includes a compressor **1112**, a condenser **1122**, an expansion valve **1132**, and an evaporator **1142** connected in series by refrigerant piping **1152**. The first compressor **1112** compresses a refrigerant in the refrigeration cycle to high temperature and high pressure. The condenser **1122** condenses and liquefies the compressed refrigerant. The expansion valve **1132** expands the liquefied refrigerant adiabatically. The evaporator **1142** evaporates the expanded refrigerant so that air passing through the evaporator **1142** is cooled by the latent heat of vaporization.

The first compressor **1112** is powered by the engine **9102** via pulleys and a pulley belt.

The heating unit **1202** is well-known and includes a mechanical pump **9112** and a heater **1212**, which are connected by cooling water piping **1232**. The mechanical pump **9112** is arranged in the engine **9102**. The heater **1212** heats air and employs the engine cooling water as a heat source. The heater **1212** is provided with an upstream water valve **1222** for adjusting the flow rate of the cooling water.

The mechanical pump **9112** is powered by the engine **9102** and circulates the cooling water through the heater **1212**.

Note that a radiator **1242** is arranged on cooling water piping **1232***a* so that the cooling water of the engine **9102** is cooled, for temperature control.

The electric compressor-pump **2002** includes a motor **2102**, a second compressor **2202**, and a pump unit **2302**. The motor **2102** has a rotating shaft **2112**, on both ends of which the second compressor **2202** and the pump unit **2302** are integrally arranged, respectively. A unidirectional clutch **2602** is located between the motor **2102** and the second compressor **2202**.

The motor **2102** is a well-known known direct current motor that is powered with a battery **9122**. The operation of the motor **2102** is controlled by the control unit **1302**, which is described later. The motor **2102** and the battery **9122** are connected to each other with two relays **1322***a* and **1322***b* located between them, so that the control circuit **1302** can exercise ON/OFF control and unidirectional (hereinafter, referred to as forward) or reverse rotation control over the motor **2102**, the details of which will be given later.

The motor **2102** also has a motor temperature sensor **2122** arranged on a predetermined portion thereof, e.g., on the outside of the motor housing. The motor temperature sensor **2122** detects changes in the temperature of the motor **2102** during operation and produces a signal accordingly. The temperature signal is input to the control unit **1302**.

The second compressor **2202** compresses and discharges the refrigerant when forward torque is transmitted from the motor **2102** through the unidirectional clutch **2602**. Reverse-rotating torque from the motor **2102** is not transmitted by the clutch **2602**, which renders the second compressor **2202** non-operational.

The pump unit **2302** is arranged on the other end of the rotating shaft **2112** of the motor **2102**. The pump unit **2302** includes a pump housing, which has a cooling water inlet **2332** and outlet **2342**, and an unillustrated centrifugal impeller. The impeller is directly connected to the rotating shaft **2112** of the motor **2102**, and the impeller is rotated by torque transmitted from the motor **2102**.

The blades of the impeller and the pump housing are configured so that the impeller discharges the cooling water (hereinafter, referred to as a normal discharging operation) while rotating opposite to the forward direction, in which the unidirectional clutch **2602** causes the second compressor **2202** to perform the normal compressing operation. In forward rotation, the impeller rotates in an idle mode (hereinafter, referred to as non-operation) due to the configuration of its blades and the pump housing.

The motor **2102**, second compressor **2202**, and pump unit **2302** are all accommodated in a housing to constitute the integral electric compressor-pump **2002**.

The second compressor **2202** of the electric compressor-pump **2002** is connected in the cooling unit **1102** in a manner parallel with the first compressor **1112** with refrigerant piping **1152***a*. Specifically, the second compressor **2202** is located between the upstream side of the condenser **1122** and the downstream side of the evaporator **1142**.

The pump unit **2302** is arranged in the heating unit **1202** to be in series with the mechanical pump **9112**. Specifically, the pump unit **2302** is located between the mechanical pump **9112** and the water valve **1222**.

Note that the cooling water piping **1232** has a bypass channel **2702**, through which the cooling water from the mechanical pump **9112** bypasses the pump unit **2302** when the motor **2102** and the pump unit **2302** are not in operation. A check valve **2802** is arranged on the bypass channel **2702** to avoid a short circuit between the outlet **2342** and the inlet **2332** during the normal discharging operation of the pump unit **2302**.

Now, description will be given of the control unit **1302** which is an essential part of the present invention.

The control unit **1302** is intended to control the operation of the motor **2102** in the electric compressor-pump **2002** described above. The control unit **1302** turns the motor **2102** on and off and controls the direction of rotation of the motor **2102** based on signals from various sensors which are not shown. The signals indicate vehicle speed, engine speed, the evaporator downstream temperature Te, the passenger compartment temperature Tr, the motor temperature Tm, the battery capacity C, and an A.C. request signal.

Specifically, the motor **2102** is driven or stopped depending on the signals, i.e., the A.C. request signal, an air-conditioning state signal (the evaporator downstream temperature Te and the passenger compartment temperature Tr), a battery capacity C signal, and a motor temperature Tm signal when the vehicle speed signal is zero (vehicle is halted) and the engine speed signal is zero (engine is stopped).

The A.C. request signal is for activating either the cooling unit **1102** or the heating unit **1202**. When cooling is requested, the relay **1322***a* is closed and the relay **1322***b* is opened so that a current flows in the direction of the solid-lined arrow. This rotates the motor **2102** in the forward direction, thereby putting the second compressor **2202** in the normal compressing operation (here, the pump unit **2302** is not in operation). When heating is requested, the relay **1322***a* is opened and the relay **1322***b* is closed so that a current flows in the direction of the broken-lined arrow. This rotates the motor **2102** in the reverse direction, thereby putting the pump unit **2302** in the normal discharging operation (here, the second compressor **2202** is not in operation). To stop the motor **2102**, the relays **1322***a* and **1322***b* are both opened.

In cooling, the air-conditioning state signal is represented by the characteristics of the evaporator downstream temperature Te. The predetermined range shall cover the temperatures between a minimum allowable temperature T**1** and a maximum allowable temperature T**2** (T**1**<T**2**) established in advance. In heating, the air-conditioning state signal is represented by the characteristic of the passenger compartment temperature Tr. The predetermined range shall cover the temperatures between a temperature T**10** and a temperature T**20** (T**10**<T**20**) established in advance.

Then, in cooling, the motor **2102** is activated in the forward direction when the evaporator downstream temperature Te exceeds the maximum allowable temperature T**2** of the predetermined range. The motor **2102** is stopped when the temperature Te falls below the minimum allowable temperature T**1** of the predetermined range. Similarly, in heating, the motor **2102** is activated in the reverse direction when the passenger compartment temperature Tr falls below the minimum temperature T**10** of the predetermined range. The motor **2102** is stopped when the temperature Tr exceeds the maximum temperature T**20** of the predetermined range.

With regard to the battery capacity C signal to be input from the battery **9122**, the minimum capacity required to restart the engine **10** after a stoppage is determined in advance as a predetermined capacity C**1**. The motor **2102** is stopped when the battery capacity C falls below the predetermined capacity C**1**.

With regard to the motor temperature Tm signal, a predetermined temperature (first predetermined temperature) T**3** at a representative location (here, the external housing as mentioned above) during operation is determined in advance in consideration of the life of the motor **2102**. When the predetermined temperature T**3** is exceeded, the motor **2102** is stopped.

Having described the configuration, description will now be given of the operation of the present embodiment.

When the vehicle is moving, i.e., the engine **9102** is running, the cooling unit **1102** and the heating unit **1202** operate in a well known manner. More specifically, in the cooling unit **1102**, the first compressor **1112** is driven by the engine **9102** to compress refrigerant. The compressed refrigerant is subsequently passed through the condenser **1122**, the expansion valve **1132**, and the evaporator **1142** for condensation, adiabatic expansion, and evaporation in succession, to cool the air passing through the evaporator **1142** by the latent heat of vaporization.

In the heating unit **1202**, the mechanical pump **9112** is driven by the engine **9102**. The water valve **1222** is opened to circulate the cooling water through the heater **1212** (in the pump unit **2302**, water passes through the bypass channel **2702**). With the cooling water as the heat source, the air passing through the heater **1212** is heated.

Nevertheless, since the air-conditioning system is employed in an idle-stop vehicle, the engine **9102** is stopped when the vehicle comes to a temporary halt. The first compressor **1112** and the mechanical pump **9112** are powered by the engine **9102** and thus quit operating. The electric compressor-pump **2002** is activated at this time.

The electric compressor-pump **2002** is controlled by the control unit **1302** as mentioned above. Hereinafter, the control procedure will be detailed with reference to a flowchart shown in FIG. **17**.

For the sake of simplicity, the following description will deal with the case where the A.C. request signal for cooling is given. That is, to drive the motor **2102**, the relay **1322***a* is closed and the relay **1322***b* is opened. It follows that a current flows in the direction of the solid-lined arrow in FIG. **16** to rotate the motor **2102** in the forward direction, thereby putting the second compressor **2202** into the normal compressing operation through the unidirectional clutch **2602** (at this time, the pump unit **2302** becomes non-operational).

Initially, at step S**102**, the motor **2102** is stopped. At step S**202**, the presence or absence of a request for A.C. (cooling) is determined from the A.C. request signal. If none, the process returns to step S**102** so that the motor **2102** remains stopped. If there is a request for cooling, the process moves to step S**302** to determine from the vehicle speed signal whether the vehicle is halted or not. If the vehicle is halted, the process moves to step S**402** to determine from the engine speed signal whether the engine **9102** is stopped or not. Incidentally, in the case of a negative result at step S**302**, the process returns to step S**102**.

If it is determined at step S**402** that the engine **9102** is stopped, the process moves to step S**502** to determine whether or not the air-conditioning state is within the predetermined range. More specifically, it is determined whether or not the evaporator downstream temperature Te falls within the predetermined temperatures T**1** to T**2** as mentioned above.

If the evaporator downstream temperature Te is determined to fall within the predetermined temperatures T**1** to T**2**, the motor **2102** is maintained in the initial stopped state at step S**602**. The process is then repeated from step S**502**.

In other words, the motor **2102** is not operated because the temperature of the air originally cooled by the first compressor **1112** (evaporator downstream temperature Te) when the engine **9102** was operating can be maintained within the predetermined temperature range (T**1**-T**2**) without activating the second compressor **2202** immediately after the stoppage of the engine.

Subsequently, if the result is negative at step S**502**, or equivalently, if the evaporator downstream temperature Te has increased gradually to exceed the predetermined range T**1**-T**2**, or the maximum allowable temperature T**2** in this case, the process moves to step S**702**. The motor **2102** is activated so that the second compressor **2202** performs the normal compressing operation.

Then, at step S**802**, whether or not the evaporator downstream temperature Te falls below the predetermined range is determined again. If the result is negative, the process moves to step S**902** to stoppage the motor **2102**. That is, after the motor **2102** is activated at step S**702**, the evaporator downstream temperature Te decreases toward the minimum allowable temperature T**1** by the action of the second compressor **2202**. Consequently, falling below the minimum allowable temperature T**1** is considered to indicate excessive cooling, and the motor **2102** is thus stopped to conserve power. Then, the process returns to step S**402**.

On the other hand, if it is determined at step S**802** that the evaporator downstream temperature Te falls within the range of T**1**-T**2** (in the process of falling from T**2** to T**1**), the process moves to step S**1002**. Here, whether or not the battery capacity C is below the predetermined capacity C**1** is determined. If the result is negative, the process moves to step S**1102** to determine whether or not the motor temperature Tm exceeds the predetermined temperature T**3**. If the result is negative, the process returns to step S**702** to keep the motor **2102** operating.

On the contrary, if the battery capacity C falls below the predetermined capacity C**1** at step S**1020** or if the motor temperature Tm exceeds the predetermined temperature T**3** at step S**1102**, the motor **2102** is stopped at step S**902** for the sake of battery capacity and motor life, respectively.

If the engine **9102** is running at step S**402**, the process returns to step S**102** so that the motor **2102** is stopped.

For heating control of the motor **2102**, the following modifications are made to the control flow described above. That is, the passenger compartment temperature Tr is used as the air-conditioning state. The predetermined range is replaced with the temperature range T**10**-T**20**. If the passenger compartment temperature Tr falls below the minimum temperature T**10** at step S**502**, the motor **2102** is driven in the reverse direction at step S**702**. If the passenger compartment temperature Tr exceeds the maximum temperature T**20** at step S**802**, the motor **2102** is stopped at step S**902**.

Having described the configuration and operation, description will now be given of the effects of the present embodiment.

Unlike the prior art, the motor **2102** is neither activated concurrently with the stoppage of the engine **9102** nor kept operated. Instead, as described above, the control flow is provided with the determination flow of steps S**502** and S**802** for motor activation and motor stoppage. The motor **2102** is thus activated just as much as needed to maintain the air-conditioning state within the predetermined range (T**1**-T**2**) as shown in timing charts of FIGS. **18**A-**18**E. Consequently, the air-conditioning performance can be ensured while the operating time of the motor **2102** is reduced to prevent overtaxing the battery **9122** and wear on the motor **2102**.

Since the control flow includes the determination flow of step S**502**, the air-conditioning state at the start of the engine **9102** can be maintained within the predetermined range (T**1**-T**2**) immediately after the stoppage of the engine **9102**. This eliminates the need for the operation of the motor **2102**, which reduces the operating time of the motor **2102**.

With regard to the air-conditioning state, the predetermined range (T**1**-T**2** or T**10**-T**20**) is judged based on the air temperature at a predetermined location (here, the evaporator downstream temperature Te or the passenger compartment temperature Tr). Consequently, temperature signals typically used in controlling an air-conditioning system can be utilized as is, without modification, which facilitates the control of the motor **2102**.

Moreover, even after the motor **2102** is driven, the motor **2102** can be stopped when the battery capacity C falls below the predetermined capacity C1 or the motor temperature Tm at a predetermined portion of the motor **2102** exceeds the predetermined temperature T**3**. It is therefore possible to reliably prevent overtaxing the battery **9122** and a wear on the motor **2102**.

Sixth Embodiment

FIG. **19** shows a sixth embodiment of the present invention. The sixth embodiment differs from the fifth embodiment in that step S**1102** of the control flow is changed to step S**1112**.

Here, instead of the motor temperature sensor **2122** arranged on the motor **2102**, the control unit **1302** is provided with a timer function for keeping the operating time tm of the motor **2102**. For a predetermined time period (first predetermined time period) t**1**, the maximum operating time per activation is previously calculated from the operating life of the motor **2102**. When the operating time tm of the motor **2102** exceeds the predetermined time t**1**, the motor **2102** is stopped at step S**902**. This provides the same effects as in the fifth embodiment.

Seventh Embodiment

FIGS. **20**-**22**E show a seventh embodiment of the present invention. The seventh embodiment is based chiefly on the cooling unit **1102**. Before the stoppage of the engine **9102**, the temperature of the evaporator **1142** is reduced in a temperature down mode, so that the evaporator **1142**, when the engine **9102** is stopped, performs cooling while less power is consumed by the motor **2102**. In FIG. **20**, as compared to the fifth embodiment, the heating unit **1202** is omitted and the electric compressor-pump **2002** is replaced with an electric compressor **2012**.

FIG. **20** shows the overall configuration of the vehicle air-conditioning system **1002**, in which a fan **1142***a* for sending air to the evaporator **1142** is provided. The air flow rate of the fan **1142***a* can be adjusted by the control unit **1302**.

The control unit **1302** also has an engine stoppage predicting function for predicting whether the engine **9102** will stop while the vehicle is moving. Specifically, this function is based on the vehicle speed signal during deceleration. The engine **9102** is predicted (determined) to come to a stop when the vehicle speed signal falls below a predetermined vehicle speed V**1**, which is established in advance. If the stoppage of the engine **9102** is predicted, the flow rate of the fan **1142***a* is reduced in comparison to the time immediately before the prediction.

Now, the control performed by the control unit **1302** over the fan **1142***a* and the motor **2102** will be described with reference to a flowchart shown in FIG. **21** and timing charts shown in FIGS. **22**A-**22**E. The flowchart shown in FIG. **21** is that of the fifth embodiment shown in FIG. **17** to which steps S**212**-S**232**, S**412**, and S**712** are added. Hereinafter, description will be given with particular emphasis on these additional steps.

Initially, the presence of an A.C. request is determined at step S**202**. If the stoppage of the engine **9102** is predicted at step S**212** from the vehicle speed signal of the vehicle moving under deceleration, the process moves to step S**222** to enter the temperature down mode. That is, the flow rate of the fan **1142***a* is reduced in comparison to the time immediately before the prediction (FIG. **22**B). As the air flow rate is reduced, the thermal load on the evaporator **1142** decreases. As a result, the temperature of the evaporator **1142**, or equivalently, the evaporator downstream temperature Te of the air cooled by the evaporator **1142**, drops further (FIG. **22**C). Then, the amount of evaporation of the refrigerant in the evaporator **1142** decreases with the decreasing thermal load. Accordingly, the refrigerant is accumulated in the condenser **1122** and the level of liquid refrigerant increases due to the condensation (FIG. **22**D). Here, since the evaporator downstream temperature Te drops according to the reduction in the air flow rate of the fan **1142***a*, the level of cooling experienced by the occupants is maintained.

After the engine **9102** is stopped, cooling is effected at step S**412** by means of the air from the lower temperature evaporator **1142**. At steps S**502** and S**602**, the cooling is continued with the motor **2102** stopped, until the evaporator downstream temperature Te reaches the maximum allowable temperature T**2**.

When the evaporator downstream temperature Te exceeds the maximum allowable temperature T**2** at step S**502** and the motor **2102** is activated at step S**702** (FIG. **22**E), the temperature down mode of step S**222** is discontinued at step S**712**. That is, the air flow rate of the fan **1142***a* is restored to the level immediately before the prediction of stoppage of the engine **9102** (FIG. **22**B). Additionally, the liquid refrigerant accumulated in the condenser **1122** and the refrigerant compressed by the operation of the motor **2102** (the operation of the second compressor **2202**) are used for air cooling (FIGS. **22**C, **22**D, and **22**E).

If the result is negative at either of steps S**302** and S**402**, i.e., if the vehicle is not halted and the engine **9102** is not stopped, the temperature down mode of step S**222** is discontinued at step S**232**.

As a result, the time that elapses before the evaporator downstream temperature Te reaches the maximum allowable temperature T**2** of the predetermined range (T**1**-T**2**) can be extended, with a further reduction in the operating time of the motor **2102**. Specifically, the time can be extended by Δt as compared to the case where the temperature down mode is not employed as shown by the double-dashed line in FIG. **22**C. The operating time of the motor **2102** can thus be made shorter than shown by the dotted and dashed line in FIG. **22**E.

Additionally, the reduced thermal load decreases the amount of evaporation of the refrigerant, so that a greater amount of liquid refrigerant can be accumulated in the condenser **1122**. When the motor **2102** is operated, the liquid refrigerant accumulated can be used to reduce the work of the motor **2102**. Consequently, aside from the effect that the stop time of the motor **2102** is extended by the cooling from the cooler evaporator **1142**, the load reducing effect during the operation of the motor **2102** allows a further reduction in the power consumption of the motor **2102**. It is therefore possible to avoid overtaxing the battery **9122** and to reduce wear on the motor **2102**.

Eighth Embodiment

FIGS. **20**, **21**, and **23**A-**23**F show an eighth embodiment of the present invention. The eighth embodiment is a modified variation of the seventh embodiment, in which the temperature down mode of the evaporator **1142** includes a valve opening control of the expansion valve **1132** aside from the air flow rate control of the fan **1142***a*.

The expansion valve **1132** in FIG. **20** is a solenoid valve that can be adjusted in valve opening by the control unit **1302**. If the stoppage of the engine **9102** is predicted, the process moves to step S**222** of FIG. **21** to enter the temperature down mode. Here, the air flow rate of the fan **1142***a* is reduced to lower the evaporator downstream temperature Te as in the seventh embodiment (FIGS. **22**B and **22**D). At the same time, the valve opening is reduced (FIG. **22**C) so that the refrigerant flows to the evaporator **1142** at a flow rate smaller than immediately before the prediction of stoppage of the engine **9102**. Incidentally, when the motor **2102** is operated, the temperature down mode is discontinued at step S**712** so that the air flow rate and the valve opening are restored to the respective values immediately before the prediction of stoppage of the engine **9102** (FIGS. **23**B and **23**C).

Consequently, the amount of evaporation of the refrigerant in the evaporator **1142** decreases, and a greater amount of liquid refrigerant can be accumulated to the condenser **1122** (FIG. **23**E) than in the seventh embodiment. This allows a further reduction in the power consumption during the operation of the motor **2102**. Note that while the condenser **1122** accumulates a greater amount of liquid refrigerant, the flow rate of the refrigerant to the evaporator **1142** decreases, which decreases the temperature drop in the evaporator downstream temperature Te. It is thus desirable to determine the size, or degree, of the valve opening to balance these considerations.

Ninth Embodiment

FIGS. **20**, **21**, and **24**A-**24**D show a ninth embodiment of the present invention. The ninth embodiment is a second variation of the seventh embodiment, in which the temperature down mode of the evaporator **1142** includes control of the air flow rate of a cooling fan **1122***a* of the condenser **1122**.

As shown in FIG. **20**, the condenser **1122** has the cooling fan **1122***a* for promoting condensation/liquefaction. The cooling fan **1122***a* is adjusted in the cooling air flow rate by the control unit **1302**.

In FIG. **21**, if the control unit **1302** predicts the stoppage of the engine **9102** at step **21**, the process moves to step S**222** to enter the temperature down mode. Here, the flow rate of the cooling fan **1122***a* is increased in comparison to the time immediately before the prediction of stoppage of the engine **9102** (FIG. **24**B). When the motor **2102** is operated, the temperature down mode is discontinued at step S**712** so that the cooling air flow rate is restored to the level that existed immediately before the prediction of stoppage of the engine **9102** (FIG. **24**B).

Consequently, the condensation in the condenser **1122** is promoted, which lowers the discharge-side pressure of the first compressor **1112**. This increases the enthalpy difference across the evaporator **1142**, which improves the cooling performance and lowers the evaporator downstream temperature Te (FIG. **24**C). As a result, the stop time of the motor **2102** at step S**602** after the stop of the engine **9102** is extended, which reduces the power consumption of the motor **2102**.

Tenth Embodiment

FIGS. **20**, **21**, and **25**A-**25**D show a tenth embodiment of the present invention. The tenth embodiment is a modified variation of the seventh embodiment, in which the temperature down mode of the evaporator **1142** includes control of the discharge of the condenser **1112**.

The first compressor **1112** employed here is of variable displacement type (for example, a known swash plate type variable displacement compressor) the displacement per rotation of which can be adjusted by the control unit **1302** as the thermal load on the cooling unit **1102** increases. In FIG. **21**, if the stoppage of the engine **9102** is predicted at step **21**, the process moves to step S**222** to enter the temperature down mode. Here, the displacement is increased, considering that the thermal load at this time has a higher value (FIG. **25**B).

As a result, the increased displacement improves the cooling performance and lowers the evaporator downstream temperature Te (FIG. **25**C). The stop time of the motor **2102** after the stoppage of the engine **9102** can thus be extended, which reduces the power consumption of the motor **2102**.

The first compressor **1112** is not limited to a variable displacement type compressor. It is also possible to use a compressor that is controlled by on and off switching, which is performed by the control unit **1302**. That is, the control unit **1302** turns the compressor on when the air temperature at a predetermined location of the cooling unit **1102** (for example, the evaporator downstream temperature Te) is higher than or equal to a predetermined temperature (second predetermined temperature). In this case, as shown in FIG. **26**B, the predetermined temperature has only to be varied to a lower value when the stoppage of the engine **9102** is predicted, so that the entire operation time is extended with an increase in discharge.

Moreover, the control of the discharge of the first compressor **1112** may be combined with the control of reducing the air flow rate of the fan **1142***a* described in the seventh embodiment. By so doing, the amount of the liquid refrigerant accumulated in the condenser **1122** can be increased for a further reduction in the power consumption upon the activation of the motor **2102**.

Eleventh Embodiment

FIGS. **27**-**29**D show an eleventh embodiment of the present invention. The eleventh embodiment is based primarily on the cooling unit **1102** of the fifth embodiment. This embodiment has an additional function of stopping the first compressor **1112** depending on the working load on the engine **9102**, so that the power performance, or acceleration performance, of the engine **9102** improves, without impairing the cooling performance of the cooling unit **1102**.

FIG. **27** shows the basic configuration of the present embodiment. As compared to the fifth embodiment shown in FIG. **16**, the heating unit **1202** is omitted and the electric compressor-pump **2002** is replaced with an electric compressor **2012**. In addition, a signal of an accelerator throttle opening is input to the control unit **1302** in order to grasp the working load on the engine **9102**. If the accelerator throttle opening is operated to increase, the vehicle is determined as accelerating.

The pulley of the first compressor **1112** is provided with an electromagnetic clutch **1112***b*, which is engaged or disengaged by the control unit **1302**. As is well known, engaging the clutch **1112***b* transmits the torque of the engine **9102** to the first compressor **1112**. When the electromagnetic clutch **1112***b* is disengaged, the first compressor **1112** stops even if the engine **9102** is running.

The operation under the configuration will be described with reference to a control flowchart shown in FIG. **28** and timing charts shown in FIGS. **29**A-**29**D. Initially, at step S**2002**, it is determined whether the vehicle is moving or in an idle-stop state. If the vehicle is in the idle-stop state, the first compressor **1112** stops, at step S**2102**, when the engine **9102** is stopped. At step S**2202**, the motor **2102** drives the second compressor **2202**.

On the other hand, if the vehicle is determined to be moving at step S2002, then it is determined at step S2302 whether or not the vehicle is accelerating. If not accelerating, the first compressor 1112 is powered by the torque of the engine 9102 at step S2402. At step S2502, the motor 2102 is stopped. That is, the second compressor 2202 is stopped.

If it is determined at step S2302 that the vehicle is accelerating, and more specifically, if it is determined that the accelerator throttle opening is increased, as shown in FIG. 29B, then the process advances to steps S2102 and S2202. The electromagnetic clutch 1112*b* is disengaged to stop the first compressor 1112, and the motor 2102 is driven to operate the second compressor 2202 (the operation of the motor 2102 is shown by shading in FIG. 29D).

Consequently, the power of the engine 9102 is conserved, since the first compressor 1112 is not operated. This allows improved power performance for situations where higher working loads are needed, such as acceleration. The motor 2102 is driven to operate the second compressor 2202, which allows the cooling unit 1102 to continue functioning.

Whether the engine 9102 is accelerating or not is determined from the accelerator throttle opening of the engine 9102. Existing control data used in controlling the engine 9102 can be used for this purpose.

Otherwise, the determination that the vehicle is accelerating may be given when the accelerator throttle opening is greater than or equal to a predetermined opening and is being increased. Aside from the accelerator throttle opening, the signals available to determine acceleration include the engine intake pressure, the engine speed, the engine cooling water temperature, and changes in the vehicle speed. The first compressor 1112 may be a variable displacement type compressor, in which case the stoppage of the first compressor 1112 may be replaced with a near-zero-discharge control by the control unit 1302.

Twelfth Embodiment

FIGS. 30 and 31A-31E show a twelfth embodiment of the present invention. The twelfth embodiment differs from the eleventh embodiment in that the motor 2102 remains stopped after the first compressor 1112 is stopped during acceleration of the vehicle, and that the motor 2102 is driven at a point when the air-conditioning state of the cooling unit 1102 exceeds a predetermined level (the maximum allowable temperature T2).

The present embodiment has the same basic configuration as that of the eleventh embodiment. During acceleration, the first compressor 1112 is stopped, and the motor 2102 is driven to operate the second compressor 2202 at a point when the air-conditioning state of the cooling unit 1102, or the evaporator downstream temperature Te, exceeds the maximum allowable temperature T2 of the predetermined range.

As shown in FIG. 30, the control flowchart is like that of the eleventh embodiment, shown in FIG. 28, except that step S2202 is changed to step S2212. (If a stoppage of the compressor due to acceleration is determined, the motor 2102 is driven at a point when the evaporator downstream temperature Te exceeds the maximum allowable temperature T2 (See FIGS. 31D and 31E).

In the prior art, stoppage of the compressor due to acceleration has caused a rise in the evaporator downstream temperature Te as shown by the double-dashed line in FIG. 31E, with a deterioration in the comfort of the passenger compartment. However, the operation of the compressor unit 2102 can improve the acceleration performance of the engine 9102 with no deterioration in passenger comfort. A motor stop time of t2 is used as shown in FIG. 31D after the stoppage of the first compressor 1112. The power consumption of the motor 2102 is reduced accordingly.

In a variation of the twelfth embodiment, as shown in FIGS. 32A-32E, the evaporator downstream temperature Te, at which the motor 2102 is driven after the stoppage of the first compressor 1112, may be changed to a temperature T21, which is lower than the maximum allowable temperature T2 (here, the stop time of the motor 2102 decreases to t21). As a result, the stoppage time due to acceleration, which was limited to t3 in the prior art, can be extended by t4 as shown in FIG. 32B. This permits a further improvement in acceleration capability as compared to the prior art shown by the double-dashed line in FIG. 32A.

Otherwise, the timing for driving the motor 2102 to operate the second compressor 2202 may be determined from the time that elapses after the stoppage of the first compressor 1112. To be more specific, the time that elapses before the evaporator downstream temperature Te reaches the maximum allowable temperature T2 (or a level lower than that) since the stoppage of the first compressor 1112 is determined in advance as a predetermined time (second predetermined time) t2, and the motor 2102 is driven after a lapse of the predetermined time period t2.

This alternative can eliminate the possibility of a delayed response in detecting temperature and improve the precision of the controls (accelerating ability, cooling performance, motor power consumption) as compared to the control procedure based on the temperature (evaporator downstream temperature Te).

Thirteenth Embodiment

FIGS. 33-35E show a thirteenth embodiment of the present invention. The thirteenth embodiment differs from the twelfth embodiment in that, when a cooling water temperature Tw, or the working load on the engine 9102 in the vehicle, exceeds a maximum allowable cooling water temperature Tw2, which is predetermined, the first compressor 1112 is stopped (high-cooling water-temperature compressor stoppage) and the motor 2102 is driven. In this connection, when the cooling water temperature Tw falls to a minimum allowable cooling water temperature Tw1 which is set below the maximum allowable cooling water temperature Tw2, the stoppage of the first compressor 1112 is discontinued.

With regard to the basic configuration, a signal of the engine cooling water temperature is input to the control unit 1302 as shown in FIG. 33. Based on the engine cooling water temperature signal, the first compressor 1112 and the motor 2102 are controlled while the vehicle is moving.

The control procedure appears in a flowchart shown in FIG. 34. Initially, at step S3002, the motor is stopped. At step S3102, whether the compressor is stopped due to high-cooling water is determined. If in the high-cooling water-temperature state, the process moves to step S3202. The motor 2102 is driven to operate the second compressor 2012 at a point when the evaporator downstream temperature Te exceeds the maximum allowable temperature T2. Here, the motor 2102 is stopped for a time t2 from the starting point of the compressor stoppage (due to high-cooling water-temperature). In the case of a negative result at step S3102, the process returns to step S3002 to repeat the process.

At step S3302, it is determined whether or not the compressor stoppage has been discontinued as a result of the reduced working load on the engine 9102 (due to the stoppage of the first compressor), which lowers the cooling water temperature Tw to below the minimum allowable cooling water temperature Tw**1**. If the stoppage is discontinued, the process returns to step S**3002** to stop the motor **2102**. If not, the process moves to step S**3402** to keep the motor **2102** operating.

In the prior art, guaranteeing the performance of the cooling unit **1102** (the evaporator downstream temperature Te) during the stoppage due to high-cooling water-temperature has caused an increase in the frequency of operations of the first compressor **1112** as shown by the double-dashed lines in FIG. **35**C. A gradual increase has also occurred in the cooling water temperature Tw of the engine **9102** as shown by the double-dashed lines in FIG. **35**B. In contrast, according to the present embodiment, the operation of the second compressor **2202** by the motor **2102** can guarantee the cooling performance during the stoppage caused by high-cooling water-temperature and reduce the working load on the engine **9102** as, thus stabilizing the cooling water temperature Tw and preventing overheating.

Here, the motor **2102** is kept from activation during the time t**2** before the evaporator downstream temperature Te reaches the maximum allowable temperature T**2**. This reduces the power consumption of the motor **2102**.

Fourteenth Embodiment

FIGS. **36** and **37**A-**37**D show a fourteenth embodiment of the present invention. The fourteenth embodiment is the eleventh embodiment provided with an additional function for driving the motor **2102** to operate the compressor unit **2102** depending on the thermal load on the cooling unit **1102** while the vehicle is under deceleration.

The present embodiment has the same basic configuration as in FIG. **27** (the eleventh embodiment); however, modifications are made in the control of the motor **2102**.

The control flow appears in the flowchart of the seventh embodiment shown in FIG. **21**, to which FIG. **36** is connected. That is, having predicted an engine stoppage during the deceleration of the vehicle at step S**212** of FIG. **21**, the process moves to step S**4002** of FIG. **36** to determine whether or not the thermal load on the cooling unit **1102** is high. Specifically, it is determined if the first compressor **1112** is in full operation and the evaporator downstream temperature Te is higher than or equal to a predetermined value T**12**.

If so, the process moves to step S**4102** in which the motor **2102** is driven to operate the compressor unit **2102**. If it is determined at step S**4002** that the thermal load is low, or that the evaporator downstream temperature Te is higher than or equal to the predetermined value T**12** but the first compressor **1112** is not in full operation (the electromagnetic clutch **1112***b* is engaged/disengaged repeatedly), the process moves to step S**222** to enter the temperature down mode. The temperature down mode may be those described in the seventh through tenth embodiments. For example, the discharge of the first compressor **1112** is increased (the electromagnetic clutch **1112***b* is kept engaged).

When it is determined at steps S**302** and S**402** that the vehicle comes to a temporary halt and the engine **9102** is stopped, the motor **2102** is temporarily stopped at step S**4202** (the compressor unit **2102** is stopped). The process advances to steps S**412** and later so that the motor **2102** is subjected to the control after the stoppage of the engine **9102**.

Consequently, when the vehicle is under deceleration or the stoppage of the engine **9102** is predicted from the deceleration state, the operation of the first compressor **1112** is combined with the operation of the second compressor **2202** by the motor **2102**. Since the flow rate of the refrigerant in the cooling unit **1102** can thus be increased to improve the cooling performance (FIG. **37**D), a cooling effect can be given with a further reduction in the required operation time of the motor **2102**. As a result, it is possible to reduce the power consumption of the motor **2102**, which avoids overtaxing the battery **9122** and extends the life of the motor **2102**. Incidentally, the power of the motor **2102** consumed during the deceleration can be concurrently generated from the deceleration energy (through regeneration) without additional fuel consumption.

Eleventh Embodiment

FIGS. **38** and **39**A-**39**E show an eleventh embodiment of the present invention. The eleventh embodiment differs from the tenth embodiment in that the motor **2102** is also driven depending on the thermal load on the cooling unit **1102** when the engine **9102** is restarted after a stoppage.

Here, the control procedure of the motor **2102** for restarting the engine **9102** after a stoppage is added, as shown in FIG. **38**. That is, at step S**5002**, whether the engine **9102** has been restarted or not is determined. If restarted, the process moves to step S**5102** to determine whether or not the thermal load on the cooling unit **1102** is high. Specifically, it is determined whether the evaporator downstream temperature Te is higher than or equal to the maximum allowable temperature T**2**.

If it is determined at step S**5102** that the thermal load is high, the process moves to step S**5202** so that the motor **2102** is driven to operate the second compressor **2202**. On the other hand, if the result is negative at step S**5102**, the process moves to step S**5302**, in which the motor **2102** is stopped to stop the second compressor **2202**. At step S**5102**, the determination of whether the thermal load is not high is based on a temperature determined by the function $\alpha T(t3)$, which is a increasing monotone increasing function of the stop time period t**3** of the engine **9102** (α is a constant). More specifically, with the evaporator downstream temperature $Te=T2-\alpha T(t3)$ as the criterion, the motor **2102** is stopped when the criterion is not reached (FIGS. **39**D and **39**E).

Consequently, the operation of the first compressor **1112** after the restart of the engine **9102** is combined with the operation of the second compressor **2202** by the motor **2102**. Since the flow rate of the refrigerant in the cooling unit **1102** can be increased to improve the cooling performance, passenger comfort (the expected level of cooling) can be restored in a shorter time than when the minimum necessary cooling has been performed by the second compressor **2202** while the engine **9102** is stopped.

A case where the test for determining whether to stop the motor **2102** is whether the evaporator downstream temperature satisfies the equation $Te=T2-\alpha T(t3)$, which includes a function of the stop time period t**3** of the engine **9102**, has been described. The criterion is not so limited, however. As shown in FIG. **39**D, the motor may be stopped when the vehicle speed increases and when the vehicle accelerates with an increase in the discharge of the first compressor **1112**.

Other Modifications

The embodiments have dealt with cases where the air-conditioning state is determined from the evaporator downstream temperature Te or the passenger compartment temperature Tr. The factors for determining the air-conditioning state are not so limited, however, and the air conditioning state may be determined, among other methods, by the pressure of the refrigerant, or the temperature of the cooling water that is circulating through the heater **1212**.

The seventh through tenth embodiments illustrate cases where the stoppage of the engine **9102** is predicted from the vehicle speed (V1) under deceleration. The prediction may however be based on the engine speed or the braking state or by other operating characteristics. In any case, the stoppage of the engine **9102** can be predicted easily.

Furthermore, the fifth through thirteenth embodiments may be modified as shown in FIG. **40**. That is, the second compressor **2202** may be integrated with the first compressor **1112**, in which case the compressor **1112** is configured as a hybrid compressor **1112***a* to be selectively powered by the engine **9102** and the motor **2102**.

Sixteenth Embodiment

Referring to FIGS. **41**-**45**D, a vehicle cooling system **1003** is used in a so-called idle-stop vehicle which has an engine **9103**, which is stopped when the vehicle comes to a temporary halt while moving. The vehicle cooling system **1003** comprises a refrigeration cycle unit **1103**, a control unit **1303**, and a battery **1403**.

The refrigeration cycle unit **1103**, which performs a known refrigeration cycle, has two compressors **1113** and **1223**. A first compressor **1113** is connected to a condenser **1123**, an expansion valve **1133**, and an evaporator **1143** in series by refrigerant piping **1153**. The first compressor **1113** compresses a refrigerant to a high temperature and high pressure. The condenser **1123** condenses and liquefies the compressed refrigerant. The expansion valve **1133** expands the liquefied refrigerant adiabatically. The evaporator **1143** evaporates the expanded refrigerant so that passing air is cooled by the latent heat of vaporization. The first compressor **1113** is powered by torque from the vehicle engine **9103** through pulleys and a belt.

The condenser **1123** has a fan **1123***a*, which sends air to the condenser **1123** to promote the liquefaction/condensation of the refrigerant.

The evaporator **1143** also has a fan **1143***b*. The fan **1143***b* forces air through the evaporator **1143**, which cools the air, and the conditioned air is sent to the passenger compartment. An evaporator temperature sensor **1143***a* for detecting the temperature of the cooled air (evaporator downstream temperature) is located on the downstream side of the evaporator **1143**.

A second compressor **1223** is arranged in parallel with the first compressor **1113**. Specifically, the second compressor **1223** is connected between the upstream side of the condenser **1123** and the downstream side of the evaporator **1143** by refrigerant piping **1233**. The second compressor **1223** is powered by a motor **1213**, which is powered by the battery **1403**. The motor **1213** and the second compressor **1223** constitute an electric compressor **1203**. The second compressor **1223** is driven when the engine **9103** is stopped and the first compressor **1113** is stopped.

A current sensor **1403***a* for detecting the electric current during the operation of the motor **1213** is located on a lead, which extends from the battery **1403** to the control unit **1303**. A pressure sensor **1163** for detecting a discharge pressure P is arranged on the discharge side of the first and second compressors **1113** and **1223**.

The configuration of the control unit **1303**, which is an essential part of the present invention, will be described with reference to the figures. The control unit **1303** is intended to operate the electric compressor **1203**. The control unit **1303** receives detection signals from the evaporator temperature sensor **1143***a*, pressure sensor **1163**, and current sensor **1403***a*, among other signals from various unillustrated sensors, including a vehicle speed signal, an engine speed signal, an idle-stop determination signal, an inside air temperature signal, an outside air temperature signal, and an A/C request signal. According to the signals, the control unit **1303** controls the motor **1213**, which drives the second compressor **1223**.

For normal operations of the refrigeration unit **1103**, the control unit **1303** performs, as might be expected, on and off control of the compressor **1113** and actuation and air flow rate control of the fans **1123***a* and **1143***b* in accordance with the signals.

In the present embodiment, a control program for operating the motor **1213** such that the battery power is conserved is stored for use by the control unit **1303**.

Initially, a cumulative operating period, during which the motor **1213** is operated while the vehicle is halted and the engine **9103** is stopped (idle-stopped), is determined as a first predetermined time period t**1**. The first predetermined time period t**1** is determined, for example, from the frequency of idling occurrences, which is estimated by simulations of the vehicle moving conditions, and the depth of discharge (use time) obtained from the number of times the battery **1403** is used. In other words, while variations in length are naturally expected of repetitive idling periods, an average value is used as the cumulative operating time (first predetermined time period t**1**) of the motor **1213** per idling in consideration of the battery life time.

As shown in FIG. **42**A, the first predetermined time period t**1** is determined according to the outside air temperature. The relationship of FIG. **42**A is stored in the control unit **1303**. Specifically, the first predetermined time period t**1** decreases when the outside air temperature decreases. That is, since lower outside air temperatures naturally require less work from the second compressor **1223**, the operating time of the motor **1213** is reduced.

The motor **1213** is operated after the lapse of a second predetermined time t**2**, or delay time, from the point when the vehicle comes to a halt and the engine **9103** is stopped. The delay time t**2** is set such that the discharge pressure P of the refrigerant that has been compressed by the first compressor **1113** drops to a predetermined value (discharge pressure Pd) within the second predetermined time t**2** from the time the engine **9103** is stopped (the time the first compressor **1113** is stopped). The motor **1213**, when operated, activates the second compressor **1223** at the reduced discharge pressure Pd.

As shown in FIG. **43**A, the second predetermined time t**2** is determined according to the outside air temperature. The relationship of FIG. **43**A is stored in the control unit **1303**. Specifically, the second predetermined time t**2** increases when the outside air temperature decreases. That is, since lower outside air temperatures naturally have less influence on the cooling performance, even with larger drops in the discharge pressure P, the delay time is increased for lower outside air temperatures.

The first and second predetermined times t**1** and t**2** are incorporated into the stored control program so that the operation of the motor **1213** is controlled accordingly.

When the vehicle is moving, i.e., when the engine **9103** is running, the refrigeration unit **1103** performs normal operations. More specifically, the compressor **1113** is driven by the engine **9103** and compresses refrigerant. The compressed refrigerant is subsequently passed through the condenser **1123**, the expansion valve **1133**, and the evaporator

1143 for condensation, adiabatic expansion, and evaporation in succession, to cool the air passing through the evaporator **1143**.

Since the cooling system is applied to an idle-stop vehicle, the engine **9103** is stopped when the vehicle comes to a temporary halt. The compressor **1113** then quits operating, and the electric compressor **1203**, i.e., the motor **1213** is operated.

The control of the motor **1213** by the control unit **1303** will be described with reference to the flowchart shown in FIG. **44** and the timing charts shown in FIGS. **45**A-**45**D in the following.

Initially, after the engine **9103** is stopped, the motor **1213** is stopped at step S**103**. Next, at step S**203**, the first and second predetermined times t**1** and t**2** are determined. More specifically, as shown in FIGS. **42**A and **43**A, the first and second predetermined time periods t**1** and t**2** are determined from the respective graphs of the predetermined time periods t**1** and t**2** stored with respect to the outside air temperature. Then, the elapse of time is measured for determining whether time period t**2** has elapsed.

Next, at step S**303**, whether the second predetermined time t**2** has elapsed or not is determined. If elapsed, the process moves to step S**403**. If not, step S**303** is repeated.

Next, at step S**403**, the motor **1213** is operated. Here, as mentioned above, the second compressor **1223** operates at the discharge pressure Pd to which the discharge pressure fell during the second predetermined time t**2**. At this point, elapsed time is measured to determine whether the first predetermined time period t**1** has elapsed.

Next, at step S**503**, whether the first predetermined time period t**1** has elapsed or not is determined. If elapsed, the process moves to step S**603** to stop the motor **1213**. If not, step S**503** is repeated.

Having described the configuration and operation, the advantages and effects of the illustrated embodiment will be described. According to the present embodiment, the motor **1213** is precluded from operating beyond the first predetermined time period t**1**, which is established in advance. Overtaxing, or over-draining, the battery **1403** is thus reliably prevented. Since the first predetermined time period t**1** is set at an average value for repetitive idling occurrences, an average cooling performance can be secured during the idling times.

The first predetermined time period t**1** is a function of the outside air temperature. This avoids unnecessary consumption of electric power, which reduces the power consumption of the motor **1213**.

Moreover, the operation of the motor **1213** is preceded by the second predetermined time t**2**, or delay time. The result is that the discharge pressure Pd decreases during the second predetermined time t**2**. Since the motor **1213** of the second compressor **1223** is activated at the reduced discharge pressure Pd, the second compressor **1223** consumes less power than it would if activated to operate at the discharge pressure Pd produced when the engine **9103** is running. The power consumption of the motor **1213** is therefore relatively low. The rush current at the activation of the motor **1213** is reduced accordingly. As a result, it is possible to prevent the rush current from reducing the life of parts and to suppress the voltage drop of the battery **1403**, which will prevent auxiliaries from malfunctioning.

The second predetermined time t**2** is a function of the outside air temperature. This also avoids unnecessary consumption of electric power and reduces motor rush current.

The first predetermined time period t**1** may be set as shown in FIG. **42**B, i.e., such that t**1** increases when the outside air temperature decreases below a predetermined value. The second predetermined time t**2** may be set as shown in FIG. **43**B, i.e., such that t**2** decreases when the outside air temperature decreases below a predetermined value. This provides improved windshield defogging during wintertime.

The first and second predetermined times t**1** and t**2** may be functions of variables corresponding to the cooling load on the refrigeration cycle unit **1103**, rather than the outside air temperature.

Seventeenth Embodiment

FIGS. **46**-**48**F show a seventeenth embodiment of the present invention. The basic configuration appears in FIG. **46**. As compared to the sixteenth embodiment, the control unit **1303** is provided with an engine start request function for requesting the starting of the engine **9103** depending on the battery capacity C.

More specifically, the battery capacity C is calculated according to the signal from the current sensor **1403***a*. If the battery capacity C falls below a predetermined capacity C**1**, the motor **1213** is stopped. An engine start request signal is sent to an engine control unit **9113** for controlling the operation of the engine **9103**. In response to this signal, the engine **9103** is started.

FIGS. **47** and **48**A-**48**F are a flowchart and timing charts during the control on the motor **1213**, respectively. As in the sixteenth embodiment, the basic controls are performed at steps S**103** to S**603**. The battery capacity C is checked at step S**703**. More specifically, when the motor **1213** is operated while the engine **9103** stopped, the battery capacity C of the battery **1403** decreases. When the battery capacity falls below the predetermined capacity C**1**, which is established in advance, the motor **1213** is stopped at step S**603**, even if the elapsed operating time of the motor **1213** is less than the first predetermined time period t**1**. At step S**803**, the engine start request signal is sent to the engine control unit **9113** to start the engine **9103**. The first compressor **1113** is thus driven with the engine **9103** as the driving source, which reliably prevents over-draining the battery while continuing the air conditioning. The battery **1403** is charged after the engine **9103** is started.

Eighteenth Embodiment

FIGS. **49**A-**49**E show an eighteenth embodiment of the present invention. The eighteenth embodiment differs from the sixteenth embodiment in that after the motor **1213** is stopped, the engine **9103** is started depending on a cooling temperature. As in the seventeenth embodiment, the control unit **1303** has the engine start request function.

Initially, the evaporator **1143** serves as a location (predetermined location) to detect the representative cooling temperature in the cooling system **1003**. Among the evaporator downstream air temperatures (hereinafter, evaporator temperatures) Te obtained by the evaporator temperature sensor **1143***a*, shown in FIG. **41**, a maximum allowable temperature in terms of cooling performance is previously set as a first predetermined temperature T**1**.

Then, as shown in FIGS. **49**A-**49**E, the engine **9103** is stopped, and the motor **1213** is operated after a lapse of the second predetermined time t**2**. The motor **1213** is stopped when its operating time reaches the first predetermined time period t**1**. Subsequently, if the engine **9103** is kept stopped for a relatively long time, the evaporator temperature Te goes up. When the evaporator temperature Te exceeds the first predetermined temperature T**1**, the engine **9103** is started as in the seventeenth embodiment.

Consequently, even if the stop time of the engine **9103** is long with respect to the first predetermined time period t**1**, the motor **1213** can be stopped after the first predetermined time period t**1** to prevent a dead battery, while the first compressor **1113** is powered by the engine **9103** for cooling performance when necessary.

Nineteenth Embodiment

FIGS. **50**A-**52**E show a nineteenth embodiment of the present invention. In the nineteenth embodiment, the motor **1213** is turned on and off and the engine is started depending on the evaporator temperature Te, which represents the degree of cooling.

Initially, first, second, and third predetermined temperatures T**1**, T**2**, and T**3** of the evaporator are established in advance on a characteristic chart with respect to the outside air temperature as shown in FIG. **50**A. The first predetermined temperature T**1** is a maximum allowable temperature, in terms of cooling performance, the second predetermined temperature T**2** is a minimum allowable temperature, and the third predetermined temperature T**3** an intermediate value. The predetermined temperatures T**1**, T**2**, and T**3** are stored in the control unit **1303**.

As shown in FIGS. **51**-**52**E, the motor **1213** is turned on and off and the engine **9103** is started with the predetermined temperatures T**1**, T**2**, and T**3** as criteria.

FIG. **51** shows the same control flowchart as that of the sixteenth embodiment of FIG. **44**, except in that step S**203** is replaced with step S**213** and except that steps S**313**, S**513**, and S**613** are added. The control procedure will be described with particular emphasis on the added steps.

Initially, at step S**213**, the first to third predetermined temperatures T**1**, T**2**, and T**3** and the first and second times t**1** and t**2** are determined from the graphs, which are established in advance.

Next, after the lapse of the second predetermined time t**2** at step S**303**, the process moves to step S**313** to determine whether the evaporator temperature Te exceeds the third predetermined temperature T**3**. If not, this step is repeated, suh that time elapses beyond the second predetermined time t**2**, and the evaporator temperature Te increases. When the evaporator temperature Te exceeds the third predetermined temperature T**3**, the motor **1213** is operated at step S**403**. That is, if a waiting time that elapses before the evaporator temperature Te exceeds the third predetermined temperature T**3** is longer than the second predetermined time t**2**, the waiting time dominates the operation of the motor **1213**.

The operation of the motor **1213** reduces the evaporator temperature Te. Before a lapse of the first predetermined time period t**1**, the process enters step S**513** to determine whether or not the evaporator temperature Te has fallen below the second predetermined temperature T**2**. If the evaporator temperature Te has fallen below the second predetermined temperature T**2**, the motor **1213** is stopped at step S**603** even if the operating time of the motor **1213** has not reached the first predetermined time period t**1**.

Then, the evaporator temperature Te increases again. At step S**613**, whether or not the evaporator temperature Te has exceeded the first predetermined temperature T**1** is determined. If so, the engine **9103** is started.

Consequently, after the engine **9103** is stopped, the cooling performance up to the third predetermined temperature T**3** can be guaranteed, while the motor **1213** is prevented from operating beyond the second predetermined time t**2**. This allows further conservation of energy.

The timing at which the motor **1213** is operated can be determined from the third predetermined temperature T**3**, which is easier than using the second predetermined time t**2** for determining when to start the motor **1213**.

In addition, if the operation of the motor **1213** produces a sufficient drop in the evaporator temperature Te at an earlier point, the motor **1213** is stopped accordingly. Thus, the power of the battery **1403** can be further conserved.

After the engine **9103** is started, the cooling performance is provided by the first compressor **1113**.

In addition, the relationship of the first to third predetermined temperatures T**1**, T**2**, and T**3** with respect to the outside air temperature may be set as shown in FIG. **50**B, i.e., such that the predetermined temperatures decrease when the outside air temperature decreases below a predetermined value. This provides improved windshield defogging during wintertime.

Moreover, the first to third predetermined temperatures T**1**, T**2**, and T**3** may be associated with variables corresponding to the cooling load on the refrigeration cycle unit **1103**, rather than the outside air temperature.

Twentieth Embodiment

FIGS. **53**-**54**F show a twentieth embodiment of the present invention. The twentieth embodiment is one in which a third predetermined time period t**3** is established, such that the starting of the engine **9103** is delayed by the third predetermined time t**3** after the motor **1213** is stopped.

This delay time from the stoppage of the motor **1213** to the start of the engine **9103**, or the third predetermined time t**3**, is stored in the control unit **1303**. The third predetermined time t**3** is set at or below 0.5 seconds. The control procedure is performed as shown in FIGS. **53**-**54**F. (In FIG. **53**, steps S**103** to S**503** are the same as in the sixteenth embodiment, and a detailed description of these steps is omitted.)

After the lapse of the second predetermined time period t**2**, the motor **1213** is operated. While the operating time is measured, whether a start signal for the engine **9103** has been issued is determined at step S**523**. That is, whether the engine control unit **9113**, shown in FIG. **46**, has issued an operation signal to an unillustrated starter, for starting the engine **9103**, is determined.

If it is determined that the start signal of the engine **9103** has been generated before the lapse of the first predetermined time period t**1**, the motor **1213** is stopped at step S**603**. A start request for the engine **9103** is issued at step S**813**. In response to the start request, the starter is operated after a lapse of the third predetermined time t**3**, which is measured from the stoppage of the motor **1213**, and the engine **9103** is started.

Consequently, the starter for starting the engine **9103** and the motor **1213** are prevented from concurrent operation. This reduces the voltage drop of the battery **1403**, which prevents auxiliaries from malfunctioning.

Since the third predetermined time t**3** is short (0.5 seconds or less), the engine **9103** can be started without an excessive time lag and occupants can start driving smoothly from a halt.

Twenty-first Embodiment

FIGS. **55**A-**55**C show a twenty-first embodiment of the present invention. In the twenty-first embodiment, the motor **1213** is operated depending on the discharge pressure P of the first compressor **1113**.

Here, the timing for operating the motor **1213** after the stoppage of the engine **9103** is determined from the discharge pressure P of the first compressor **1113**. More specifically, a pressure value that is lower than a normal discharge pressure P of the first compressor **1113** and acceptable in terms of cooling performance is previously established as a first predetermined pressure P**1**. The motor **1213** is operated if the discharge pressure is less than the first predetermined pressure P**1**.

Consequently, the motor **1213** of the compressor **1223** can be activated at the reduced first predetermined pressure P**1** so that the cooling performance is satisfactory while the compressor **1223**, as in the sixteenth embodiment, consumes less power than when activated at the discharge pressure P produced when the engine **9103** is running. The motor **1213** therefore consumes relatively less energy. The rush current at the activation of the motor **1213** is reduced accordingly. As a result, it is possible to prevent the rush current from reducing the life of relevant parts and to limit the voltage drop of the battery **1403**, which prevents auxiliaries from malfunctioning.

Twenty-Second Embodiment

FIGS. **56**A-**56**D show a twenty-second embodiment of the present invention. In the twenty-second embodiment, the motor **1213** is operated when the discharge pressure P of the first compressor **1113** is controlled to a smaller value before the stoppage of the engine **9103**, or in the present case, before the first compressor **1113** is turned off, when the engine **9103** is stopped.

Consequently, the second compressor **1223** can be activated at the dropped discharge pressure P, so that the second compressor **1223**, as in the sixteenth embodiment, consumes less power than it would if activated at the discharge pressure P produced when the engine **9103** is running. The motor **1213** thus consumes less energy. The rush current at the activation of the motor **1213** can be reduced accordingly. As a result, it is possible to prevent the rush current from reducing the life of relevant parts and to suppress the voltage drop of the battery **1403**, which prevents auxiliaries from malfunctioning.

The first compressor **1113** is not limited to a compressor that is controlled by being switched on and off but may be a variable displacement type compressor.

Twenty-Third Embodiment

FIGS. **57**A-**57**D show an twenty-third embodiment of the present invention. In the twenty-third embodiment, the discharge pressure of the first compressor **1113** is lowered while the vehicle is decelerating and not yet stopped (for example, when the vehicle speed has fallen below a predetermined vehicle speed V**1**). The motor **1213** is operated after the engine **9103** is stopped. Specifically, the discharge pressure P is lowered by turning the first compressor **1113** off. This has the same effects as the twenty-second embodiment.

Twenty-Fourth Embodiment

FIGS. **58**A-**58**D show a twenty-fourth embodiment of the present invention. The twenty-fourth embodiment differs from the twenty-third embodiment in that the discharge pressure P is lowered by increasing the air flow rate of the fan **1123***a* of the condenser **1123** shown in FIG. **41**.

Since the forced cooling of the refrigerant in the condenser **1123** lowers the discharge pressure P before the operation of the motor **1213**, the effects are the same as those of the twenty-second and twenty-third embodiments.

Twenty-Fifth Embodiment

FIGS. **59**A-**60**D show a twenty-fifth embodiment of the present invention. In the twenty-fifth embodiment, the operation of the motor **1213** is precluded as an emergency measure when the cooling load on the refrigeration cycle unit **1103** before a stoppage of the engine **9103** is higher than a predetermined load.

Here, the discharge pressure P of the first compressor **1113** is used as a variable representative of the cooling load on the refrigeration cycle unit **1103**. For a criterion, a second predetermined pressure P**2** is established above the first predetermined pressure P**1** of the twenty-first embodiment, shown in FIG. **55**B.

Specifically, as shown in FIGS. **59**A-**59**D, when the discharge pressure P before the stoppage of the engine **9103** exceeds the second predetermined pressure P**2**, the engine start request signal is input to the engine control unit **1303** to prevent stoppage of the engine **9103** even if the vehicle comes to a halt. The first compressor **1113** is kept operating (the motor **1213** does not operate). Moreover, as shown in FIGS. **60**A-**60**D, even if the engine **9103** is stopped, the motor **1213** is kept from operating.

Consequently, when the discharge pressure P exceeds the second predetermined pressure P**2** and the cooling load is extremely high, the motor **1213** is entirely precluded from operation as a safety measure. This prevents extreme power consumption by the motor **1213**, which avoids overtaxing the battery.

As shown in FIGS. **59**A-**59**D, when the engine **9103** is running, the cooling can be performed by the first compressor **1113**.

The cooling load on the refrigeration cycle unit **1103** may be represented by other factors such as the passenger compartment temperature and the evaporator temperature Te.

Other Embodiments

The illustrated embodiments have shown cases where the compressors consist of the first compressor **1113** and the second compressor **1223**, which are driven by the engine **9103** and the motor **1213**, respectively. However, the compressors are not so limited. As shown in FIG. **61**, a so-called hybrid compressor **1113***a*, which is selectively powered by the engine **9103** and the motor **1213**, may be used instead.

The invention claimed is:

1. A vehicle air-conditioning apparatus for use in a vehicle in which an engine for driving the vehicle is stopped when the vehicle comes to a temporary halt from a running state, the apparatus comprising:

a cooling unit for cooling air by a refrigerating cycle comprising an engine driven compressor, a condenser, an expanding device and an evaporator;

a heating unit for heating the air by using, as a heat source, cooling water for the engine circulated by an engine driven mechanical pump;

an electric compressor-pump including an electric motor capable of switching rotating directions, a compression unit operatively connected with the motor for compressing a refrigerant in the refrigerating cycle as a substitute for the compressor, and a pump unit operatively connected with the motor for circulating the cooling water as a substitute for the mechanical pump; and a control unit for controlling the rotating directions of the motor so that the motor rotates in the first direction when the engine is stopped during operating the cooling unit, and rotates in the second direction when the engine is stopped during operating the heating unit, wherein

the electric compressor-pump includes:

means, provided as a mechanical structure to enable or disable the compression unit and the pump in response to the rotating direction of the motor alone, for enabling the compression unit to compress the refrigerant when the motor rotates in the first direction, for disenabling the compression unit when the motor rotates in the second direction, and for enabling the pump unit to circulate the cooling water at least when the motor rotates in the second direction.

2. The vehicle air-conditioning apparatus according to claim **1**, wherein

the compression unit is connected with the refrigerating cycle in parallel with the engine driven compressor, and the pump unit is connected with a circuit of the cooling water in series with a heat exchanger to heat the air.

3. The vehicle air-conditioning apparatus according to claim **2**, further comprising:

a circuit for switching current direction supplied to the motor in response to the control unit.

4. The vehicle air-conditioning apparatus according to claim **3**, wherein

the means enables the pump unit to circulate the cooling water when the motor rotates in the second direction, and disenables the pump unit when the motor rotates in the first direction.

5. The vehicle air-conditioning apparatus according to claim **3**, wherein

the means enables the pump unit to circulate the cooling water when the motor rotates in either the first direction or the second direction.

6. The vehicle air-conditioning apparatus according to claim **3**, wherein

the means is provided by a unidirectional clutch provided between the compression unit and the motor.

7. The vehicle air-conditioning apparatus according to claim **6**, wherein

the electric compressor-pump further includes:

a shaft sealing unit, provided at a position closer to the compression unit, for preventing leakage of the refrigerant.

8. The vehicle air-conditioning apparatus according to claim **3**, wherein

the means is provided by a rotary compression unit provided as the compression unit that performs compression only during rotation in the first direction.

9. The vehicle air-conditioning apparatus according to claim **3**, wherein

the means is provided by a releasing mechanism provided in the compression unit, the releasing mechanism opening a compression chamber during rotation in the second direction.

10. The vehicle air-conditioning apparatus according to claim **9**, wherein

the compression unit includes a scroll compressor, and

the releasing mechanism is a radius compensating mechanism.

11. The vehicle air-conditioning apparatus according to claim **9**, wherein

the compression unit includes a scroll compressor having a fixed scroll and a movable scroll, and

the releasing mechanism is provided by at least one of the fixed scroll and the movable scroll which is made of resin.

12. The vehicle air-conditioning apparatus according to claim **1**, wherein

the compression unit and the pump unit are located at opposite ends of a rotating shaft of the motor.

13. The vehicle air-conditioning apparatus according to claim **1**, wherein

the electric compressor-pump farther includes:

a shaft sealing unit for preventing leakage of the refrigerant and being located between the compression unit and the motor; and

a magnetic coupling to couple the pump unit with the motor.

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